

Pulsation and Vibration Control for Positive Displacement Machinery Systems for Petroleum, Chemical, and Natural Gas Industry Services

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Introduction

Users of this standard should be aware that further or differing requirements may be needed for individual applications. This standard is not intended to inhibit a vendor from offering, or the purchaser from accepting, alternative equipment or engineering solutions for the individual application. This may be particularly appropriate where there is innovative or developing technology. Where an alternative is offered, the vendor should identify any variations from this standard and provide details.

Annex A contains informative descriptions of the work process for acoustical and mechanical analyses. It is focused on reciprocating compressors but is somewhat applicable to all positive displacement (PD) machinery.

Annex B contains information concerning stepless capacity control for reciprocating compressor cylinders.

Annex C contains the Design Approach 2 flowchart for API 618 machines.

Annex D contains the Design Approach 3 flowchart for API 618 machines.

Annex E contains information for forced mechanical response analyses of compressor systems and piping.

Annex F contains information for small-bore piping analysis and design.

Annex G contains pulsation considerations for flow metering equipment.

Annex H contains information for Vi and pressure ratio considerations for API 619 machines.

Annex I contains the design approach flowchart for PD pumps.

Annex J contains information concerning cavitation considerations for PD pump systems. In this standard, U.S. customary (USC) units are included in brackets for information.

Pulsation and Vibration Control for Positive Displacement Machinery Systems for Petroleum, Chemical, and Natural Gas Industry Services

1 Scope

This standard covers the minimum requirements for pulsation and vibration control for positive displacement (PD) machinery systems used in the petroleum, chemical, and natural gas industry services. The specific machinery addressed includes:

- reciprocating compressors (ref. API 618);
- rotary-type PD compressors (ref. API 619);
- PD pumps—reciprocating (ref. API 674);
- PD pumps—controlled volume (ref. API 675);
- PD pumps—rotary (ref. API 676).

2 Normative References

2.1 The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

API Standard 618, *Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services*

API Standard 619, *Rotary-type Positive-displacement Compressors for Petroleum, Chemical, and Gas Industry Services*

API Standard 674, *Positive Displacement Pumps—Reciprocating*

API Standard 675, *Positive Displacement Pumps—Controlled Volume for Petroleum, Chemical, and Gas Industry Services*

API Standard 676, *Positive Displacement Pumps—Rotary*

ASME B31.3 ¹, *Process Piping*

ASME B31.8, *Gas Transmission and Distribution Piping Systems*

ASME *Boiler and Pressure Vessel Code (BPVC)*, Section III: Rules for Construction of Nuclear Power Plant Components; Division 2: Code for Concrete Reactor Vessels and Containments

ASME *Boiler and Pressure Vessel Code (BPVC)*, Section III, Appendix 1, 2010 Edition

ASME *Boiler and Pressure Vessel Code (BPVC)*, Section VIII: Rules for Construction of Pressure Vessels; Division 1: Rules for Construction of Pressure Vessels

ASME *Boiler and Pressure Vessel Code (BPVC)*, Section VIII: Rules for Construction of Pressure Vessels; Division 2: Alternative Rules

¹ ASME International, 2 Park Avenue, New York, NY 10016-5990, www.asme.org.

2.2 The hierarchy of documents shall be specified.

NOTE Typical documents submitted as a user inquiry or order are user specifications, industry specifications (such as API and ISO specifications), datasheets, meeting notes, and supplemental agreements.

3 Terms, Definitions, Acronyms, and Abbreviations

3.1 Terms and Definitions

For the purposes of this document, the following definitions apply.

3.1.1

acoustic simulation

Process whereby the one-dimensional (1D) acoustic characteristics of fluids, and the compressor or pump dynamic flow influence on these characteristics, are modeled, taking into account the fluid properties, the compressor model and the connected vessels and piping, and other equipment. The model is based upon the governing mathematical equations (motion, continuity, etc.). The simulation should allow for determination of pressure/flow modulations at any point in the piping model resulting from any generalized compressor excitation.

3.1.2

active analysis

Portion of the acoustic simulation in which the pressure pulsation amplitudes due to imposed compressor(s)/pump(s) operation for the anticipated loading, speed range, and state conditions are predicted. (Refer to 3.1.1.)

3.1.3

amplification factor

AF

Measure of damping or pulsation/vibration sensitivity to excitation when the frequency of the excitation source is coincident with or near an acoustic or mechanical natural frequency. A high amplification factor ($AF > 10$) indicates that vibration during operation near a natural frequency could be excessive. A low amplification factor (e.g. $AF < 5$) indicates that the system is not as sensitive to excitation when operating in the vicinity of the associated acoustic or mechanical natural frequency.

3.1.4

analog simulation

Method using electrical components (inductances, capacitances, resistances, and current supply devices) to achieve acoustic simulation. This technique was used to pioneer the techniques for pulsation control but has been superseded by modern digital pulsation simulation software.

3.1.5

coefficient of restitution

Ratio of the relative velocity of two particles after impact to their relative velocity before impact. This term is used to help describe the dynamic motion of the moving components of valves in PD compressors and pumps.

3.1.6

combined rod load

Algebraic sum of gas load and inertia force on the crosshead pin. Gas load is the force resulting from differential gas pressure acting on the piston differential area. Inertia force is the force resulting from the acceleration of the total reciprocating mass. The inertia force acting on the crosshead pin is the summation of the products of all reciprocating masses (piston and rod assembly and crosshead assembly including pin) and their acceleration.

3.1.7**compression ratio**

Ratio of internal compressor discharge pressure to suction pressure, expressed as p_i . Compression ratio for a screw compressor is dependent on volume ratio and the ratio of specific heats of the gas (k).

3.1.8**damping**

Inherent property of a dynamic system by which mechanical energy is removed. Examples of energy dissipation in machinery and piping systems are internal material damping and friction forces due to relative motion of the piping and supports.

3.1.9**design**

Manufacturer's calculated parameter.

NOTE A term used by the equipment manufacturer to describe various parameters such as design power, design pressure, design temperature, or design speed. It is not intended for the purchaser to use this term.

3.1.10**designer**

Refers to analyst performing acoustical/mechanical studies. May be the original equipment manufacturer (OEM), engineering, procurement, and construction (EPC), or third party.

3.1.11**digital acoustic simulation**

Method using various mathematical techniques on digital computers to achieve the acoustic simulation.

3.1.12**discharge dampener**

A term commonly used to refer to the pulsation suppression device on the discharge side of a PD pump.

3.1.13**frequency**

Number of cycles of a repetitive motion within a unit of time. Frequency is calculated as the reciprocal of the period of the repeating motion. Frequency is generally expressed as cycles per second (cps or Hertz) or cycles per minute (cpm).

3.1.14**infinite length line (boundary condition)**

Term used for an acoustically nonreflective boundary condition. In this case, the waves move away from the source and are not reflected at any point in the line. This is a method used to create an acoustically nonresonant piping system for the purpose of evaluating preliminary designs of pulsation suppression devices or in the case that the piping for the model is not available.

3.1.15**line-side connections**

Term used to identify the connections of a pulsation suppression device, which are the inlet connection on suction and the outlet connection on discharge.

3.1.16**maximum continuous speed**

Highest rotational speed [revolutions per minute (RPM)] at which the machine, as built and tested, is capable of continuous operation.

3.1.17**mode shape of an acoustic resonance**

Description of the relative pulsation amplitudes and phase angle relationship at various points in the piping system. Knowledge of the mode shape allows the analyst to understand the pulsation patterns in the piping system.

3.1.18**natural frequency**

The frequency at which a system tends to oscillate in the absence of any driving force.

3.1.19**normal operating condition**

Point at which usual operation is expected and optimum efficiency is desired, usually the certified point.

3.1.20**oil stiction**

Condition where the moving sealing element of a compressor or pump valve “sticks” to the stationary sealing element because of the adhesive effect of oil that is present. When stiction occurs, it increases the force required to open the valve.

3.1.21**oil-flooded screw compressor**

Helical-lobe rotary compressor with a lubricant (compatible with the process gas) injected into the rotor area after the closed thread position of the rotor.

NOTE The lubricant helps seal rotor clearances and establishes an oil film between the rotors. One rotor drives the other in the absence of a timing gear.

3.1.22**oil-free screw compressor**

Helical-lobe rotary compressor that uses no liquid for sealing the rotor clearances and driving the non-coupled rotor.

NOTE 1 The rotor-to-rotor relationship is maintained by timing gears on each rotor, and the non-coupled rotor is driven by the coupled rotor through the timing gears.

NOTE 2 No rotor-to-rotor contact occurs in the oil-free screw compressor.

3.1.23**passive analysis**

Portion of the acoustic simulation in which a constant flow amplitude modulation over an arbitrary frequency range is imposed on the system, normally at the cylinder valve locations. The resulting transfer function defines the acoustic natural frequencies and the mode shapes over the frequency range of interest.

3.1.24**pocket passing frequency****PPF**

Frequency at which the gas is discharged from the rotor lobes into the discharge port (for a screw compressor).

NOTE Pocket passing frequency (PPF), expressed in hertz, is calculated by multiplying the rotor rotational speed, expressed in RPM, by the number of lobes on that rotor and dividing the product by 60.

3.1.25**pressure design code**

Recognized pressure vessel standard specified or agreed by the purchaser.

3.1.26

pulsation suppression device

Typically includes volume bottles, baffles, choke tubes, and/or orifice plates configured to control pulsation levels. The terms “suction stabilizer” and “discharge dampener” are commonly used for pulsation suppression devices on PD pumps.

3.1.27

relief-valve set pressure

Pressure at which a relief valve starts to lift.

3.1.28

resonance

Term applied to either acoustic or vibration forced response. Resonance occurs when a forcing function frequency coincides with, or is near, an acoustic or mechanical natural frequency, resulting in an amplified response limited only by the amount of dampening in the system.

3.1.29

rotor

Rotating male or female assembly (for screw compressors), including rotor body, shaft, and shrunk-on sleeves (if furnished).

3.1.30

separator

Pressure-containing device, usually a vessel, used to separate a fluid from the process gas.

3.1.31

silencer

The term commonly used to refer to pulsation suppression devices for screw compressors. Silencers can be “absorptive” or “reactive” designs. Absorptive silencers utilize some type of “fill” material (steel wool, foam, etc.), whereas reactive designs do not.

3.1.32

stepless capacity control

Achieved by holding the suction valves completely open for a calculated duration of time during the compression stroke to vary compressor cylinder capacity.

3.1.33

stiffness

Equivalent spring rate in Newtons/millimeter (N/mm) or pounds/inch (lb/in.) of mechanical system elements. Examples of components for which stiffness is important are piping supports and cylinder supports.

3.1.34

suction stabilizer

A term commonly used to refer to the pulsation suppression device on the suction side of a PD pump.

3.1.35

suction system

The equipment on the suction side of the compressor, including silencer, valve/s, knockout drums, and any other inline equipment.

3.1.36

support

Refers to a six-degree-of-freedom boundary condition for components such as pipe clamps, cylinder supports, bottle supports, etc. Also commonly referred to as a restraint. This can refer to a variety of actual support types such as clamps, guides, stops, vertical supports only, etc.

3.1.37**volume ratio**

Ratio of internal compressor suction volume to discharge volume, expressed as V_i . Volume ratio is not dependent on gas composition. It is adjustable on some oil-flooded screw compressors but not on oil-free screw compressors.

3.2 Acronyms, Abbreviations, and Symbols

<i>AF</i>	amplification factor
bara	bar absolute
barg	bar gauge
BDC	bottom dead center
CE	crank end
CPN	cavitation potential number
DN	Nominal Diameter, mm
EPC	engineering, procurement, and construction
FAM	fundamental analytical method
FEA	finite element analysis
FEM	finite element method
GVF	gas void fraction
HE	head end
IDC	inner dead center
LWN	long weld neck
<i>MACCRL</i>	maximum allowable continuous combined rod load
MACGL	maximum allowable continuous gas load
MNF	mechanical natural frequency
NPIP	net positive inlet pressure
NPIPA	net positive inlet pressure available
NPIPR	net positive inlet pressure required
NPS	Nominal Pipe Size, inches
NPSH	net positive suction head
NPSHR	net positive suction head required
ODC	outer dead center
OEM	original equipment manufacturer

PD	positive displacement
<i>PD</i>	piston displacement (bore × stroke)
P&ID	piping and instrument diagrams
PPF	pocket passing frequency
PSV	pressure safety valve
PV	pressure-volume
RPM	revolutions per minute
<i>SM</i>	separation margin
TDC	top dead center
VDPA	valve dynamic performance analysis
1D	one-dimensional
3D	three-dimensional

4 General

4.1 Dimensions and Units

Drawings and maintenance dimensions shall be in SI units or U.S. customary (USC) units. Use of an SI datasheet indicates that SI units shall be used. Use of an USC datasheet indicates that USC units shall be used.

4.2 Responsibility

The supplier shall assume responsibility and shall assure that all sub-vendors comply with the requirements of this standard and all referenced documents.

5 Requirements

5.1 Reciprocating Compressors (Ref. API 618)

5.1.1 General

The focus of this standard is on reciprocating compressors that are API 618 compliant (typically low-to-medium speed compressors, 300 RPM to 900 RPM). However, the vibration and pulsation control techniques are often applied to higher speed compressors and non-API equipment. The current design approaches have been applied with good success for higher speed compressors (>900 RPM). At some point (e.g. 1800 RPM machines), the basic approach of avoiding resonance [i.e. minimum mechanical natural frequency (MNF) > 2.4× running speed] is not practical and alternate design techniques (not covered in this standard) must be used. Robust pulsation control is the primary design goal in all cases.

Refer to Annex A for guidance on the application of pulsation and vibration control techniques.

NOTE Annex A contains information on the design of pulsation suppression, piping and support layout, sizing of acoustic filters, and shaking forces that were part of API 688, First Edition.

5.1.1.1 The recommended techniques used for control of detrimental pulsation and vibration are the following:

- a) system design based on analysis of the interactive effects of pulsation and the attenuation requirements for satisfactory levels of piping vibration, compressor performance, valve life, and operation of equipment sensitive to flow pulsation;
- b) utilization of pulsation suppression devices such as: pulsation filters; volume bottles, often with internals; choke tubes; orifice systems; and selected piping configurations (refer to Annex A for a discussion of bottle internals, weep holes, etc.).
- c) mechanical restraint design; specifically including such things as: type, location, and number of supports.

5.1.1.2 If specified for preliminary sizing, pulsation suppression devices shall have suction surge volume and discharge surge volume (not taking into account liquid collection chambers), as determined from Equations (1), (2), and (3), but in no case shall either volume be less than 0.03 m³ (1 ft³).

Proprietary methods for preliminary bottle sizing may be used and the resultant volumes supplied for consideration in addition to those determined by Equations (1) to (3).

In SI units:

$$V_s = 8.1 \times PD \left(\frac{kT_s}{M} \right)^{1/4} \quad (1)$$

$$V_d = 1.6 \times \left(\frac{V_s}{R^{1/k}} \right) \quad (2)$$

$$V_s \text{ and } V_d \geq 0.03 \quad (3)$$

In USC units:

$$V_s = 7 \times PD \left(\frac{kT_s}{M} \right)^{1/4} \quad (1)$$

$$V_d = 1.6 \times \left(\frac{V_s}{R^{1/k}} \right) \quad (2)$$

$$V_s \text{ and } V_d \geq 1.0 \quad (3)$$

where

V_s is the minimum required suction surge volume in m³ (ft³);

V_d is the minimum required discharge surge volume in m³ (ft³);

k is the isentropic compression exponent at average operating gas pressure and temperature;

R is the stage pressure ratio at cylinder flanges (absolute discharge pressure divided by absolute suction pressure);

T_s is the absolute suction temperature in K (°R);

M is the molecular weight;

PD is the total net displaced volume per revolution of all compressor cylinders to be connected to the surge volume in m^3 (ft^3).

The internal diameter of the surge volume shall be based on the minimum surge volume overall length required to manifold the compressor cylinders. For a single-cylinder surge volume, the ratio of surge volume length to internal diameter shall not exceed 4.0. The inside diameter of spherical volumes shall be calculated directly from the volumes determined by Equations (1), (2), and (3).

NOTE 1 Equations (1), (2), and (3) can be used for preliminary sizing of the pulsation suppression devices that are included with the compressor vendor's proposal. The sizes are typically changed according to the simulation analysis employed by Design Approaches 2 and 3. Sizing requirements can be substantially influenced by operating parameters, interaction among elements of the overall system, and mechanical characteristics of the compressor system. The magnitude of the effects of these factors cannot be accurately predicted at the outset.

NOTE 2 Some compressor applications require the use of properly designed low-pass acoustic filters, which can affect final sizing.

5.1.1.3 When acceptance criteria as defined in 5.1.7 cannot be met, the purchaser and vendor shall agree on acceptable limits and control methods.

5.1.1.4 The purchaser shall specify if the analysis is to be performed by the vendor or a third party. If a third party is selected to perform the analysis, the compressor vendor shall provide the necessary information required for the third party to complete the analysis.

5.1.2 Alternate Operating Conditions

5.1.2.1 Analyses shall be performed for all specified alternative gases, operating conditions, loading steps, or stepless capacity control. Stepless unloading shall be evaluated at least at minimum flow, 50 %, 75 %, and 100 %. When stepless capacity control is utilized, the design procedures in Annex B shall apply.

5.1.2.2 For compressor systems with extensive alternate operating conditions, the extent of the analysis shall be defined in the proposal.

NOTE It may be impractical or unnecessary to analyze every combination of operating conditions. The number of analysis points may be based on experience and purchaser/vendor agreement.

5.1.2.3 When a compressor is to be operated on two or more gases of dissimilar molecular weights (e.g. hydrogen compressor with an occasional low-pressure nitrogen condition), pulsation levels shall be optimized for the gas on which the unit will operate for the greater length of time. Verification of mechanical integrity at the alternate condition is required.

5.1.3 Multiple Unit Additive Effects

The scope of the analysis shall be based on agreement between the purchaser and vendor.

5.1.3.1 The purchaser shall specify when multiple compressors are to be operated in parallel.

5.1.3.2 The purchaser shall specify when the compressor is to be operated in conjunction with existing compressor units and their associated piping systems. Modifications to an existing system to obtain acceptable pulsation levels shall be mutually agreed upon.

NOTE In some cases, it may be necessary to impose tighter limits for each new compressor than those defined in 5.1.7.4 in order for the combined system to achieve acceptable pulsation levels.

5.1.4 Design Approach Selection

Table 1 shall be utilized to determine the design approach.

Table 1—Design Approach Selection

Discharge Pressure	Rated Power per Cylinder		
	kW/cyl < 55 (hp/cyl < 75)	55 < kW/cyl < 220 (75 < hp/cyl < 300)	220 < kW/cyl (300 < hp/cyl)
$P < 35$ barg ($P < 500$ psig)	2	2	2
35 barg < $P < 70$ barg (500 psig < $P < 1000$ psig)	2	2	3
70 barg < $P < 200$ barg 1000 psig < $P < 3000$ psig)	2	3	3
$P > 200$ barg ($P > 3000$ psig)	3	3	3

NOTE 1 Design Approach 1 has been removed and the equations used for initial bottle sizing are defined in 5.1.1.2.

NOTE 2 In some cases, a partial system analysis is performed, referred to as an “on-skid” analysis. The acoustical model is terminated with reflection-free boundary conditions at the skid edge. The mechanical model is also terminated at the skid edge. An “on-skid” vs “off-skid” analysis approach is typically agreed upon between the purchaser and the vendor. This is independent of Design Approach 2 vs Design Approach 3 procedures. The “off-skid” piping can be considered once the piping system is finalized.

5.1.5 Design Approach 2—Acoustic Simulation and Piping System Mechanical Review

5.1.5.1 General

Design Approach 2 is pulsation control through the use of pulsation suppression devices and proven acoustic techniques in conjunction with mechanical review of pipe spans and support systems to achieve control of vibrational response. This approach includes the evaluation of acoustic interaction between the compressor, pulsation suppression devices, and associated piping, including pulsation effects on compressor performance and an evaluation of acoustic shaking forces in the pulsation suppression devices. The evaluation is accomplished by modeling the compressor system and the piping and then performing an acoustic simulation to determine the response. Refer to Annex C for the Design Approach 2 flowchart.

5.1.5.2 Compressor System Acoustical Model

Pulsation suppression devices are analyzed using acoustic simulation. The compressor system model shall include piston and valve motion, cylinder passages, pulsation suppression device(s), and terminates at the line-side nozzle flanges. When stepless capacity control is used, the frequency range shall be sufficient to cover the higher harmonics as generated by the stepless control system. This model is only used for the acoustic simulation. There is no mechanical modeling of the compressor system included in Design Approach 2.

5.1.5.3 Piping System Model

When the layout and sizing of the piping system is completed, an acoustic simulation of the complete system shall be performed to confirm compliance with the requirements of 5.1.7.4 or to identify changes necessary to achieve compliance. In some cases, a downstream volume can be introduced to terminate the scope of the acoustic model. Refer to Annex A for a discussion of the extents of the acoustic model.

5.1.5.4 Pre-study

For applications where the piping system is not defined, the acoustic simulation can be performed with the piping system initially modeled with an infinite length, acoustically nonreflective line. This is referred to as a “pre-study.” When a pre-study is performed, the maximum allowable pressure pulsation level at the pulsation suppression device line-side nozzle flange shall be 70 % of the allowable value defined in 5.1.7.4. Pressure-drop requirements in 5.1.8.1 and shaking force requirements in 5.1.7.5 shall apply.

NOTE 1 The 70 % limit may not ensure that the pulsation allowable levels in 5.1.7.4 will be met in the final design without use of other acoustical measures (orifices, increased bottle diameters, piping modifications, etc.).

NOTE 2 In order to meet contract delivery, all parties should cooperate to schedule the design of the pulsation suppression device, the pulsation analysis, and piping design. Ordering components after the pre-study can facilitate the procurement of long delivery components of the pulsation suppression devices such as end caps, nozzles, and cylindrical sections. However, the final length, nozzle orientation, and need for vessel internals cannot be optimized until the piping system is added to the acoustic model.

NOTE 3 If the pulsation suppression devices are fabricated prior to finalizing the piping configuration, the only remaining system design optimization methods available to the designer are the installation of orifices, or piping modifications.

5.1.5.5 Mechanical Review and Piping Restraint Analysis

A mechanical review shall be performed using span and basic vessel MNF calculations to avoid mechanical resonance. This review shall result in a table of various pipe sizes that indicates the maximum allowable span (based on the maximum compressor operating speed) between piping supports as a function of pipe diameter and the separation margin requirements of 5.1.8.2.

5.1.6 Design Approach 3—Acoustic Simulation and Piping Restraint Analysis Plus Mechanical Analysis

5.1.6.1 General

This acoustic simulation approach is identical to Design Approach 2, with the addition of a mechanical analysis of the compressor system (defined in 5.1.6.2.2) and associated piping systems including interaction between acoustic and mechanical system responses. Forced mechanical response is included when necessary (i.e. when separation margins or shaking force guidelines are not met). Refer to Annex D for the Design Approach 3 flowchart.

5.1.6.1.1 If specified, an analysis of the stresses found in the pulsation suppression device internals that result from pulsation-induced shaking forces and pressure-induced static forces shall be analyzed to confirm compliance with 5.1.7.7 stress criteria.

5.1.6.1.2 If required by the specific pressure vessel code, such as ASME *BPVC* Section VIII, Division 1 and Division 2, a low-cycle fatigue analysis shall be performed to predict stresses from thermal gradients, thermal transients, and pressure cycles on the pulsation suppression devices and internal components.

5.1.6.2 Step 3a—MNF Analysis of the Compressor System and Piping System

5.1.6.2.1 An analysis of the compressor system and piping system shall be done to predict the mechanical natural frequencies. The mechanical and acoustic system shall be designed to meet the separation margin criteria of 5.1.8.2 and the shaking forces shall not exceed the limits defined in 5.1.7.5. The purpose of calculating piping system mechanical natural frequencies is to determine whether or not the separation margin guidelines are met. These calculations are usually made using one of two methods. The first method involves the use of fundamental beam theory and exact closed form solutions. This method is referred to as the fundamental analytical method (FAM). The second method involves the use of numerical solutions or finite element method (FEM). The accuracy of either method is greatly influenced by the assumed boundary conditions. When properly applied, the two methods should give similar results. If basic design concepts for piping layout are followed (e.g. minimize the number of bends, place a clamp near each bend and near all

concentrated masses, etc.), the minimum natural frequency of most piping systems can readily be evaluated using the FAM approach. The FEM approach for calculating natural frequencies is only required for unusual layouts [long flexible spans with complex three-dimensional (3D) bends between supports, etc.]. However, if separation margin and shaking force guidelines are not met and the calculation of vibration amplitudes and dynamic stresses is required, the FEM approach is then employed.

NOTE The FEM model of the piping is best done concurrently with the thermal flexibility analysis.

5.1.6.2.2 The starting point of the compressor system mechanical model shall be the crosshead guide-to-crankcase interface. This location shall be relatively stiff when compared to the rest of the compressor mechanical model, and it shall be accurately described by a six-degree-of-freedom spring, or a 3D model of the compressor frame can be utilized. The compressor system mechanical model end point shall be a location on the suction and discharge piping moving away from the line side nozzles of the pulsation suppression devices that has sufficient stiffness to enforce a vibration node. This location is typically the second pipe support.

NOTE The intent is to avoid mechanical resonance of the compressor system and piping system at frequencies where high shaking forces also exist.

5.1.6.3 Step 3b1—Forced Mechanical Response Analysis of the Compressor System

5.1.6.3.1 When the excitation frequency separation margins (5.1.8.2) or the shaking force guidelines for pulsation suppression devices (5.1.7.5.3) cannot be met, a forced mechanical response analysis of the compressor system to the pulsation-induced forces and cylinder gas rod load forces shall be performed. The allowable vibration criteria in 5.1.7.6 and the allowable cyclic stress criteria in 5.1.7.7 shall apply for the piping. Refer to Annex E for additional information regarding the forced response analysis.

NOTE It is not intended that analysis of the cyclic stresses in the compressor components be included in this design approach. The compressor components are included in the model only for the purpose of enabling the analysis of the effects of their flexibility and dynamic movement on the pulsation suppression devices.

5.1.6.3.2 The allowable vibration limits for the compressor cylinders and frame shall be supplied by the compressor OEM.

NOTE The allowable compressor vibration levels are generally the limiting design criteria.

5.1.6.4 Step 3b2—Forced Mechanical Response of the Piping System Model

5.1.6.4.1 When the separation margins (5.1.8.2) or the shaking force guidelines (5.1.7.5.2) for the piping system cannot be met, a forced-mechanical response analysis of the piping system to pulsation-induced shaking forces shall be performed. The piping vibration design criteria in 5.1.7.6 shall apply. If the vibration limits are not met, the cyclic stress limits in 5.1.7.7 shall apply. The piping system model end points shall be defined by the analyst in agreement with the purchaser. The piping system model shall include all of the piping that was included in the acoustic simulation. Refer to Annex E for additional information regarding the forced response analysis.

5.1.6.4.2 When a forced mechanical response analysis of the piping system is performed without doing a forced mechanical response analysis of the compressor mechanical model, the starting point of the piping system is at the compressor cylinder flanges, which are assumed to be rigid.

NOTE As with Step 3b1, the vibration is generally the limiting design consideration. This is because when the vibration levels are within the recommended allowable limits, the allowable stress levels are usually not approached. The exception is where high stress concentrations occur at large-diameter reductions such as nozzle connections and small-bore piping on significantly larger piping. Refer to Annex F for additional discussion of small-bore piping.

5.1.7 Design Criteria

5.1.7.1 General

Pulsation suppression devices and techniques applied in accordance with Design Approaches 2 and 3 shall satisfy the basic criteria defined in 5.1.7 and the other criteria defined in 5.1.8.

5.1.7.2 Maximum Allowable Pressure Pulsation at Compressor Cylinder Flanges

Unless other criteria (such as loss of compressor efficiency) are specified, the unfiltered peak-to-peak pulsation level at the compressor cylinder flange, as a percentage of mean absolute line pressure, shall be limited to the lesser of 7 % or the value computed from Equation (4).

$$P_{cf} = 3R \% \quad (4)$$

where

P_{cf} is the maximum allowable unfiltered peak-to-peak pulsation level, as a percentage of mean absolute line pressure at the compressor cylinder flange;

R is the stage pressure ratio.

After calculating the percentage from Equation (4), this percentage shall be multiplied by the mean absolute line pressure or 3.5 bara (50 psia), whichever is greater, to establish the allowable pulsation in peak-to-peak units.

5.1.7.3 Maximum Allowable Pressure Pulsation at Compressor Cylinder Valves

The unfiltered peak-to-peak pulsation level at the compressor cylinder valves shall be limited to double the allowable pressure pulsation at the cylinder flanges as defined in 5.1.7.2.

5.1.7.4 Allowable Pulsation Levels at and Beyond Line-side Connections of Pulsation Suppression Devices

Based on normal operating conditions, the peak-to-peak pulsation levels in the initial suction, interstage, and final discharge piping systems beyond pulsation suppression devices shall satisfy the requirements specified in 5.1.7.4.1 and 5.1.7.4.2.

5.1.7.4.1 For systems operating at absolute line pressures above 3.5 bara (50 psia), the peak-to-peak pulsation level of each individual frequency component shall be limited to that calculated by Equation (5a).

In SI units:

$$P_l = \left[\frac{400}{(P_L \times D_l \times f)^{0.5}} \right] \sqrt{\frac{a}{350}} \quad (5a)$$

In USC units:

$$P_l = \left[\frac{300}{(P_L \times D_l \times f)^{0.5}} \right] \sqrt{\frac{a}{1150}} \quad (5a)$$

where

P_l is the maximum allowable peak-to-peak level of each individual frequency component expressed as a percentage of mean absolute line pressure;

- a is the speed of sound for the gas in m/s (ft/s);
- P_L is the mean absolute line pressure in bar (psia);
- D_I is the inside diameter of line pipe in mm (in.);
- f is the pulsation frequency in Hz.

The pulsation frequency f is derived from Equation (6).

$$f = \frac{zN}{60} \quad (6)$$

where

- N is the shaft speed in RPM;
- z is 1, 2, 3..., corresponding to the fundamental frequency and integer harmonics.

5.1.7.4.2 For absolute pressures less than 3.5 bara (50 psia), the allowable pulsation from Equation (5a) shall be multiplied by Equation (5b).

In SI units:

$$\sqrt{\frac{3.5}{P_L}} \quad (5b)$$

In USC units:

$$\sqrt{\frac{50}{P_L}} \quad (5b)$$

5.1.7.4.3 If pulsation levels exceed the limits defined by 5.1.7.4.1 or 5.1.7.4.2, the allowable shaking force (5.1.7.5), or vibration design criteria (5.1.7.6), or cyclic stress levels (5.1.7.7) shall be met. Maximum pulsation levels shall be approved by the purchaser.

5.1.7.4.4 Flow pulsations in elements sensitive to such phenomena (e.g. check valves, cyclone separators) shall be limited to mutually agreed criteria.

5.1.7.5 Allowable Acoustic Shaking Force

5.1.7.5.1 General

The allowable nonresonant peak-to-peak piping shaking force based on the design vibration guideline shall be determined from Equation (7).

$$SF_k = k_{\text{eff}} \times V \quad (7)$$

where

- SF_k is the nonresonant peak-to-peak force guideline relative to static structural stiffness in N (lbf);
- k_{eff} is the effective static stiffness along the piping or pulsation suppression device axis where the shaking force acts in N/mm (lbf/in.). See Annex A for a detailed discussion of k_{eff} ;

V is the design peak-to-peak vibration guideline (see Figure 1) in displacement units (mm or in.).

NOTE The shaking force guideline (SF_k) applies to nonresonant vibration. Shaking forces near resonance shall be reduced well below the above shaking force guideline. This guideline is simplified from a complex analysis, contains many inherent assumptions, and should be applied with care.

Various support types provide ranges of support stiffness as follows.

Configuration	Typical Stiffness (N/mm)	Typical Stiffness (lbf/in.)
Elevated unbraced rack supports	900 to 2,700	5,000 to 15,000
Grade level typical supports and clamps	2,700 to 27,000	15,000 to 150,000
Grade level heavy-duty supports and clamps	27,000 to 45,000	150,000 to 250,000

5.1.7.5.2 Allowable Piping System Acoustic Shaking Force

The allowable piping peak-to-peak shaking forces shall be the lower of the values calculated from Equation (7) or from Table 2.

Table 2—Allowable Shaking Force Guideline

Configuration	Allowable Shaking Force per Elbow Pair			
	Nonresonant ($SM > 20\%$)		Potentially Resonant ($SM < 20\%$)	
	USC Units (lbf, p-p)	SI Units (N, p-p)	USC Units (lbf, p-p)	SI Units (N, p-p)
Ground supported	$250 \times \text{NPS}$	$44.5 \times \text{DN}$	$25 \times \text{NPS}$	$4.45 \times \text{DN}$
Rack supported	$20 \times \text{NPS}$	$3.5 \times \text{DN}$	$2 \times \text{NPS}$	$0.35 \times \text{DN}$

5.1.7.5.3 Allowable Cylinder Mounted Pulsation Suppression Device Shaking Force

The allowable peak-to-peak shaking forces for cylinder mounted pulsation suppression devices shall be the lower of the values calculated from Equation (7) or from Equation (8). For Design Approach 2, since the shaking force levels are not evaluated using Equation (7), the maximum allowable level shall be 10 % of Equation (8); however, for frequencies within $\pm 20\%$ of the calculated pulsation suppression device MNF, the allowable level shall be 1 % of Equation (8).

$$SF_{\text{dmax}} = 5\sqrt{MACCRL \times \text{Number of Cylinders}} \quad (8)$$

where

SF_{dmax} is the maximum pulsation suppression device nonresonant peak-to-peak force guideline in N (lbf);

$MACCRL$ is the maximum allowable continuous combined rod load defined in API 618 in N (lbf);

$\text{Number of Cylinders}$ is number of cylinders to which the pulsation suppression device is attached.

NOTE The shaking force criteria are intended as design criteria for shaking forces that act along the pulsation suppression device axis.

5.1.7.5.4 Gas Rod Load Induced Forces

Gas rod loads can generate cylinder shaking forces that are independent of the pulsation control design. A forced response analysis of the compressor system shall be performed if the gas rod loads at the higher harmonics (typically 3× to 8×) are greater than 10 % of the rated gas rod load [maximum allowable continuous gas load (MACGL)]. Refer to Annex E for additional details on forced response analysis.

5.1.7.6 Piping Design Vibration Criteria

The predicted piping vibration magnitude for each discrete frequency shall be limited to the design level illustrated in Figure 1. The diagram in Figure 1 is based on the following:

- a constant allowable vibration amplitude of 0.57 mm p-p (22 mils p-p) for frequencies below 10 Hz;
- a constant allowable vibration velocity of approximately 18 mm/s 0-p (0.71 in./s 0-p) for frequencies between 10 Hz and 200 Hz.

The limits in Figure 1 are intended as a design trigger level for analysis in accordance with 5.1.6.

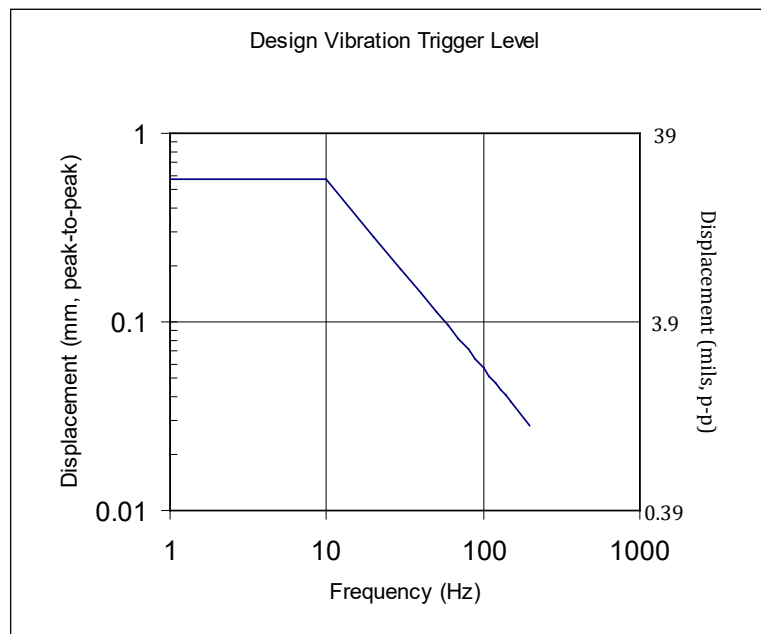


Figure 1—Piping Vibration Design Trigger Level at Discrete Frequencies

NOTE 1 Field acceptance criteria may differ from the design levels. Refer to other standards (such as OEM guidelines, user experience, as agreed between purchaser and vendor) for field acceptance criteria.

NOTE 2 The requirements in this subsection are considered to be conservative. There are, however, situations in which high-stress risers and unbraced small-diameter attached piping can pose a problem even though the main pipe exhibits acceptable vibration limits. There are no criteria conservative enough to be used without a significant understanding of vibrational mechanics.

5.1.7.7 Allowable Cyclic Stress

5.1.7.7.1 For Design Approach 3, Steps 3b1 and 3b2, pulsation and/or mechanically induced vibration shall not cause a cyclic stress level in the piping and pulsation suppression devices in excess of the endurance limits of materials used for components subject to these cyclic loads. For carbon steel with an operating temperature below 370 °C (700 °F), the peak-to-peak cyclic stress shall be less than 96 MPa (14,000 psi) considering

stress concentration factors present and other stresses within applicable code limits. Endurance limits for materials other than carbon steel shall be as defined in the specified pressure vessel code.

5.1.7.7.2 If specified, a piping system flexibility analysis that predicts forces, moments, and stresses resulting from thermal gradients, thermal transients, pipe and fitting weights, static pressure, and bolt-up strains shall be performed. The specified piping code (e.g. ASME B31.3, ASME B31.8, etc.) shall provide the design criteria.

5.1.8 Other Criteria

5.1.8.1 Allowable Pressure Drop

Allowable pressure drop through the pulsation control devices for each suction and discharge system of each compression stage is defined below.

- a) Steady flow pressure drop shall not exceed 0.5 % of mean absolute line pressure at pressure ratios less than or equal to 1.4. At pressure ratios greater than 1.4, the value shall be determined by Equation (9). The allowable limits shall be increased by a factor of two when the pressure drop is calculated using the total flow, where total flow is the sum of the steady state and dynamic flow components, provided that the steady flow component still meets the above criteria determined by Equation (9).

$$\Delta P = 1.67 \left(\frac{R-1}{R} \right) \% \quad (9)$$

where

ΔP is the allowable pressure drop based on steady flow through devices installed for pulsation control, expressed as a percentage of mean absolute line pressure;

R is the stage pressure ratio.

- b) When a moisture separator is an integral part of the pulsation suppression device, steady flow pressure drop shall not exceed 0.6 % of mean absolute line pressure at pressure ratios less than or equal to 1.4. At pressure ratios greater than 1.4, the value shall be determined by Equation (10). The allowable limits shall be increased by a factor of two when the pressure drop is calculated using the total flow, where total flow is the sum of the steady state and dynamic flow components, provided that the steady flow component still meets the criteria determined by Equation (10).

$$\Delta P = 2.17 \left(\frac{R-1}{R} \right) \% \quad (10)$$

NOTE 1 The allowable pressure drop specified in this section can be exceeded by mutual agreement between purchaser and vendor, when this is the consequence of the preferred pulsation control.

NOTE 2 Dynamic pressure drop can be a concern for restrictions located near compressor cylinders (i.e. cylinder flange orifice plates). Generally, these orifice plates (when properly sized) improve performance due to pulsation control. However, exercise caution to avoid excessive pressure drop and potential driver overload.

5.1.8.2 Separation Margins

The guidelines described below are to be used to avoid coincidence of excitation frequencies with mechanical natural frequencies of the compressor, pulsation suppression devices, scrubbers, and piping systems:

- a) the minimum MNF of any compressor or piping system element shall be designed to be greater than 2.4× the fundamental frequency at maximum rated speed, and

- b) the predicted mechanical natural frequencies shall be designed to be separated from significant excitation frequencies by at least 20 %.

NOTE 1 The intent is to be above twice running speed, because there is generally sufficient excitation energy at the first and second orders to excite resonances to an unacceptable level.

NOTE 2 Significant excitation sources include cylinder gas loads (stretch) and can also include pulsation shaking forces at frequencies above the second order.

5.1.8.3 Flow Measurement Error

For flow meters located in the piping system, the maximum flow measurement error caused by flow pulsations shall not exceed the following.

- a) For non-custody transfer meters: 1.00 % error.
- b) For custody transfer meters: 0.125 % error.

Refer to Annex G regarding allowable pulsation and/or velocity fluctuations at the flow meter that must be maintained such that the above noted meter errors will not be exceeded.

5.1.9 Documentation Requirements

A written report on the control of pulsation and vibration shall be furnished to the purchaser. Compliance with the requirements of 5.1 for the specified design approach shall be documented. The report shall define the analysis scope, including analysis guidelines, compressor configuration, load steps, gas composition, and extent of the piping system analyzed. The report shall include the recommendations resulting from the analysis. The documentation shall also present results applicable to each type of analysis performed. Acoustic simulation results include cylinder nozzle and piping pulsation, acoustic shaking forces, and flow pulsation at equipment sensitive to this in spectrum form. Separation margin analysis results include natural frequencies and mode shapes. Forced mechanical response results include vibration and cyclic stress. The presentation of the analysis results should allow easy comparison with applicable criteria.

5.1.10 Summary of Design Steps

Historical descriptions of the commonly used analysis methods for pulsation and vibration control include the following.

- M.1—Basic design of pulsation suppression devices using proprietary and/or empirical techniques (possibly including a simplified pulsation simulation), combined with a simplified mechanical analysis of the piping system.
- M.2—Pulsation simulation, control of pulsation, and shaking force levels.
- M.3—Pressure drop and performance evaluation.
- M.4—Mechanical analysis of piping system to determine mechanical natural frequencies (FAM and/or FEM techniques) to maintain adequate separation margins.
- M.5—Mechanical analysis of compressor system to determine mechanical natural frequencies (FEM techniques) to maintain adequate separation margins.
- M.6—Forced response analysis of compressor system.
- M.7—Forced response analysis of piping system.
- M.8—Static and dynamic stress analysis of pulsation bottle internals.

- M.9—Valve dynamics analysis including attached piping.
- M.10—Low-cycle fatigue analysis of pulsation suppression devices due to thermal and pressure cycles.
- M.11—Thermal flexibility analysis of piping system.

The typical design approaches utilize various combinations of these methods as follows.

- Design Approach 2 (<100 hp, <75 kW): M.1.
- Design Approach 2: M.2–M.4.
- Design Approach 3 (Default Approach—Design Step 3a): M.2–M.5.
- Design Step 3b1: M.6.
- Design Step 3b2: M.7.
- Optional Analysis: M.8–M.11 (guidance on when to employ these is included in Annex A).

5.2 Rotary Type PD Compressors (Ref. API 619)

5.2.1 The requirement for, and the scope of, an analysis of pulsation, vibration, and noise suppression shall be agreed between the purchaser and the vendor. The decision to perform an analysis is usually based on experience of the purchaser and criticality of the application. Compressors operating at less than 500 hp (375 kW) are not addressed since they typically do not have pulsation control issues.

NOTE 1 Pulsation and vibration issues are more common on the compressor discharge; however, the suction system is also included for some applications (particularly oil-free screw compressors).

NOTE 2 Preference should be given to (a) system designs with proven success and (b) system designs that can be easily adjusted in the field.

5.2.2 Before order placement, purchaser and vendor shall agree upon the scope of information to be provided by the vendor as part of the pulsation analysis. Some methods used by vendors to dampen pulsations are proprietary in nature. This can include the sizing and internal design of silencers, layout of piping, connection types, etc.

5.2.3 When an acoustic analysis is required by the purchaser, the purchaser is responsible to provide the gas conditions at the inlet of the vendor's scope of supply, including gas composition, pressure, and temperature at all operating conditions. The purchaser is also responsible to define the system boundaries for the analysis.

NOTE 1 When designing the compressor and piping system, consider the entire operating range, including the entire speed range in variable-speed applications, range of temperatures, pressures, and variation of the gas conditions, as well as intermittent operating conditions with purge gas.

NOTE 2 Gas composition has a significant impact on acoustic behavior. Therefore, accurate gas composition is critical for acoustic analysis and design. (See Annex H for discussion of V_1 and compression ratio.)

NOTE 3 In screw compressor systems, the flow of gas is not steady but moves through the piping in a series of flow pulses that are superimposed upon the steady (average) flow. The characteristics of the flow pulses are determined by the size and the operating conditions of the compressor (displacement, speed, rotors, pressures, etc.). The mechanical and acoustical response from the piping system is a function of the amplitude and frequencies of the pulses, the thermo-physical properties of the gas, and the piping system's characteristics (layout, supports, natural frequencies, etc.).

NOTE 4 Screw compressors generate pulsation at high frequencies. This, combined with large-diameter vessels or piping, makes 3D modes more important to consider.

5.2.4 Unless otherwise specified, a pulsation analysis shall be conducted up to at least the 5th harmonic of the PPF. The pulsation source strength shall be provided by the OEM in order to evaluate the pulsation levels. The source strength is given as either an equation or a graph of flow vs time and is considered at the nozzles of the compressor. This analysis shall demonstrate that the silencer(s) is/are capable of reducing gas pulsations to the levels specified in 5.2.8.4.

5.2.5 A mechanical analysis of the silencer(s) shall be conducted to evaluate the effect of the excitation forces on vibration levels. The vendor shall demonstrate that the natural frequency of the silencer(s) shell is not within 20 % of the integer PPF harmonics up to the 5th harmonic of PPF. Since this is impractical in many applications (i.e. variable speed), the silencer shall be designed using structural damping means, e.g. to withstand operation in resonance where it cannot be avoided.

NOTE For oil-flooded screw compressors, the oil separator is effectively considered the discharge silencer.

5.2.6 Unless otherwise specified, the vendor(s) with piping design responsibility shall demonstrate that the natural frequencies of the discharge piping (scope of discharge piping limits for analysis should be agreed upon) and its main branches are not within 10 % of PPF. Pipe clamps/supports may be necessary to maintain this 10 % separation margin. For piping and vessel shell wall natural frequencies, pipe supports are not effective. Wall thickness may be chosen based on shell wall natural frequencies instead of static pressure requirements.

5.2.7 Unless shown to be unnecessary by the acoustic analysis, suction and discharge silencers for each casing shall be supplied by the vendor. Their primary function shall be to provide the maximum practical reduction of pulsations up to at least the 5th harmonic of the PPF without exceeding the pressure-drop limit specified in 5.2.8.

5.2.8 Unless otherwise agreed, the mean line pressure drop through the silencers shall not exceed the following values:

- a) for suction silencers: 1 % of the absolute pressure at the silencer inlet;
- b) for discharge silencers: 2.5 % of the absolute pressure at the silencer discharge.

5.2.8.1 The pressure drop shall be stated in the datasheets and shall be accounted for in the calculation of the power required.

5.2.8.2 In the case of low-pressure [suction pressure < 1 barg (15 psig)] and vacuum applications, the suction pressure drop may exceed the 1 % limit to achieve the necessary pulsation attenuation.

5.2.8.3 For machines with widely varying operating conditions (e.g. speed, molecular weight of the gas, pressure, etc.), the above-mentioned limits might not be achievable in all cases. Unless otherwise specified, the limits above will apply to the normal operating condition on the API 619 compressor datasheets.

5.2.8.4 The peak-to-peak pulsation levels, p_{pp} , expressed as a percentage of the mean line-side absolute pressure on the process piping side of the inlet and discharge silencers, shall not exceed 2 % or the value calculated from Equation (11), expressed in SI units, and Equation (12), expressed in USC units, whichever is the smaller:

$$p_{pp} = \frac{28.6}{P_{AM}^{1/3}} \quad (11)$$

$$p_{pp} = \frac{15}{P_{AM}^{1/3}} \quad (12)$$

where

P_{AM} is the absolute mean line-side pressure, expressed in kilopascals (pounds per square inch).

Below approximately 3000 kPa (435 psia) absolute line pressure, the 2 % limit governs. Above that, Equations (11) and (12) govern. These limits are shown graphically in Figure 2.

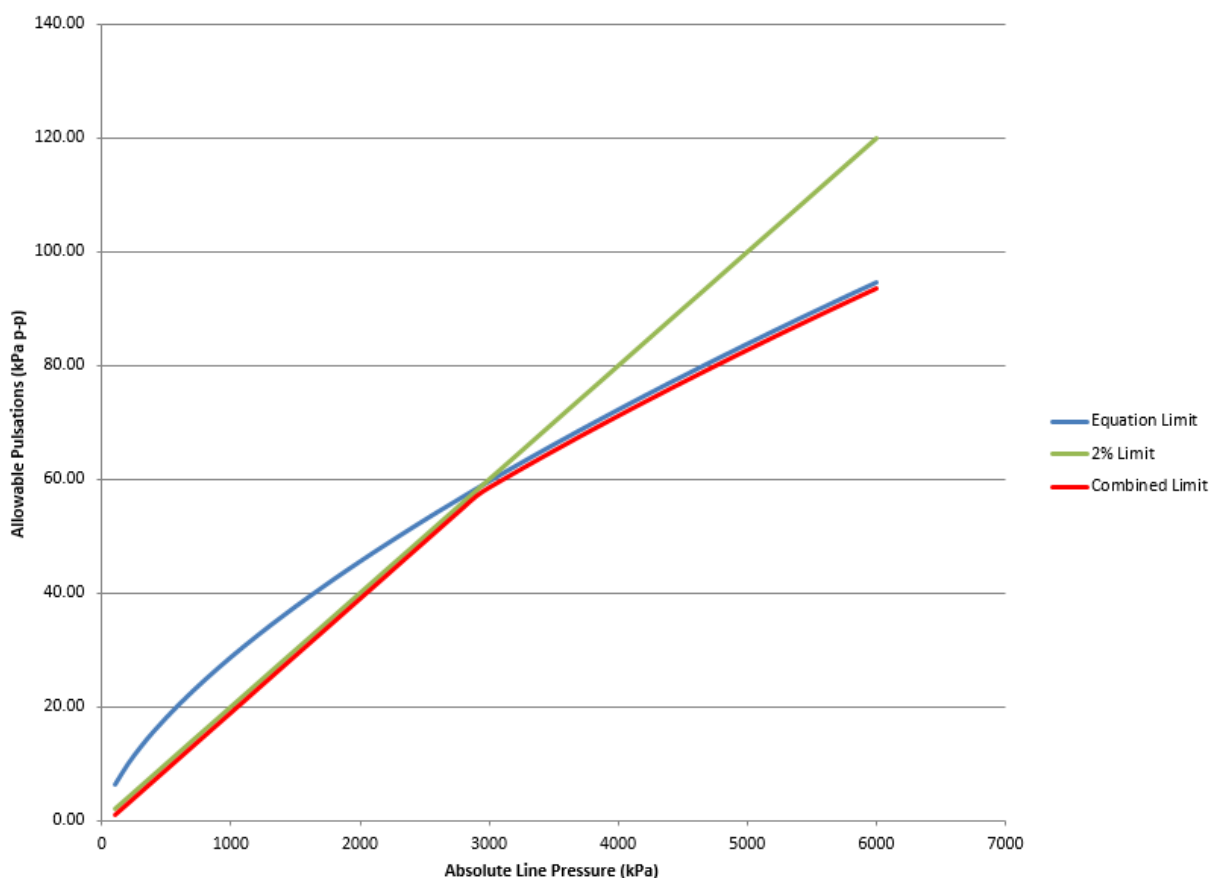


Figure 2—Allowable Pulsation vs Absolute Line Pressure

5.2.9 Silencers shall be located with respect to the compressor flanges as agreed by the purchaser and vendor.

5.2.10 Maximum silencer efficiency results from mounting the silencers directly on the compressor flanges. In the case of oil-flooded screw compressors, the oil separator should be as close to the discharge flange as practical.

5.2.11 If orifice plates are used, the analysis shall provide the calculated pressure drop and corresponding power losses for each orifice plate proposed. If used, such devices shall be easily accessible for inspection, cleaning, removal, and/or replacement.

5.2.12 If specified, the silencer vendor shall supply detailed drawings to permit an independent study of the acoustical characteristics of the silencers together with the purchaser's piping system.

5.2.13 Small-bore connections (DN 40 or NPS 1½ in.) and smaller shall be addressed per the discussion in Annex F.

5.2.14 Only reactive-type silencers shall be applied for oil-flooded-screw compressors.

5.2.15 Reactive-type silencers shall be applied for oil-free screw compressors. The use of absorptive-type silencers/materials may be used on dry-screw applications, subject to review by the purchaser.

5.3 PD Pumps—Reciprocating and Controlled Volume (Ref. API 674 and API 675)

5.3.1 General

Refer to Annex A and Annex I, the informative part of reciprocating pumps according to the API 674 and API 675, for guidance on the application of pulsation control requirements, which contains general information of liquid pump systems and information on the design of pulsation suppression devices. For the mechanical vibration control, refer to Annex A, which contains general information on mechanical control, piping, and support layout.

The primary pulsation excitation frequencies (Hz) are defined in Equation (13).

$$\frac{N}{60} \times P \times n \quad (13)$$

where

- N is pump shaft speed (RPM);
- P is number of plungers;
- n is 1 for single acting, 2 for double acting.

Piping system pulsations and vibrations typically occur at multiples of the frequencies defined by Equation (13).

5.3.1.1 The basic techniques used for control of detrimental pulsations and vibrations are the following:

- a) system design based on analysis of the interactive effects of pulsations and the attenuation requirements for satisfactory levels of piping vibration, avoiding cavitation, avoiding seat leakage and/or premature opening of relief valves, and operation of equipment sensitive to flow pulsation;
- b) utilization of pulsation suppression devices such as pulsation filters and attenuators; volume bottles, with or without internals such as bladders; choke tubes; orifice systems; and selected piping configurations;
- c) mechanical restraint design, specifically including such things as type, location, and number of pipe and equipment clamps and supports.

5.3.1.2 When acceptance criteria defined in 5.3.7 cannot be met, the purchaser and vendor shall agree on acceptable limits and control methods.

5.3.1.3 The purchaser and the vendor shall agree on who shall carry out the analysis. If a third party is selected to perform the analysis, the pump vendor shall provide the necessary information required for the third party to complete the analysis.

5.3.2 Alternate Operating Conditions

5.3.2.1 The analysis shall be performed for all specified operating conditions. For pump systems with extensive alternate operating conditions, the extent of the analysis shall be defined in the proposal.

NOTE 1 It may be impractical or unnecessary to analyze every combination of operating conditions. The number of analysis points may be reduced based on experience.

NOTE 2 A pump with healthy valves will exhibit characteristic pulsation and vibration behaviors based on its excitation frequency. When a valve fails, the characteristic pulsation excitation frequency of the pump is effectively changed based

on the number of plungers that become inactive or partially active due to the failure. As such, the resulting change in vibration behavior of the piping system can be dramatic. Awareness of valve health during operation of reciprocating pumps is of necessary importance. Valve health can be determined with online monitoring comparisons of expected flow with actual flow. The acoustic design will not typically account for pulsations associated with a system that has failed valves.

5.3.3 Multiple Unit Additive Effects

The scope of the analysis shall be based on agreement between the purchaser and vendor.

5.3.3.1 The purchaser shall specify when multiple pumps are to be operated in parallel. The acoustical simulation shall include all units in the system.

5.3.3.2 The purchaser shall specify when the pump is to be operated in conjunction with existing pump units and their associated piping systems. Modifications to an existing system to obtain acceptable pulsation levels shall be agreed upon.

NOTE In some cases, it may be necessary to impose tighter limits for each new pump than those defined in 5.3.7 in order for the combined system to achieve acceptable levels.

5.3.4 Analysis Design Selection Criteria

5.3.4.1 Table 3 shall be utilized to determine when the specified analyses shall be carried out for API-674-type pumps as defined in 5.3.5 and 5.3.6, unless otherwise agreed upon between the purchaser and vendor.

5.3.4.2 If specified, analyses for API-675-type pumps shall be performed according to Table 3 criteria.

NOTE API-675-type pumps typically will not require acoustic and/or mechanical analyses; however, some applications warrant these types of analyses due to criticality, toxicity, integrity, etc.

Table 3—Analysis Selection Chart

Speed (Maximum Operating Speed as a % of the Maximum Allowable Speed in Table 4)	Rated Power per Pump		
	hp < 100 (kW < 75)	100 ≤ hp ≤ 200 (75 ≤ kW ≤ 150)	hp > 200 (kW > 150)
Speed < 50 % of max allowable	No acoustic or mechanical analysis	Piping system mechanical review	Acoustic simulation and piping system mechanical review
50 % ≤ Speed ≤ 75 %	Piping system mechanical review	Acoustic simulation and piping system mechanical review	Acoustic simulation and mechanical analysis
75 % < Speed	Acoustic simulation and piping system mechanical review	Acoustic simulation and mechanical analysis	Acoustic simulation and mechanical analysis

NOTE A piping system mechanical review as defined in 5.3.6.2 is not necessary if a detailed finite element analysis (FEA) of the piping system is conducted to determine the MNFs and mode shapes.

Table 4—Maximum Allowable Speed Table

Stroke Length		RPM for a <i>Single-acting</i> Plunger Pump	RPM for a <i>Double-acting</i> Piston-type Pump
mm	in.		
50	2	450	140
75	3	400	127
100	4	350	116
125	5	310	108
150	6	270	100
175	7	240	94
200	8	210	88
250	10	168	83
300	12	140	78
350	14	120	74
400	16	105	70

5.3.5 Acoustic Simulation

5.3.5.1 General

The acoustic simulation shall be carried out through the use of pulsation suppression devices and proven acoustic techniques. The evaluation is accomplished by modeling the pump piping system and then performing an acoustic simulation to determine the response. Refer to Annex I for the design analysis flowchart.

5.3.5.2 Acoustic Simulation Model

The system shall be analyzed using proven acoustic simulation models. The pump system model shall include the piston or plunger and the effect of valve motion, the clearance volume, pump manifold, pulsation suppression devices, and the pipe system up to suitable acoustic boundary conditions.

NOTE This model is only used for the acoustic simulation, and there is no mechanical modeling of the pump system to evaluate mechanical resonances.

The following requirements shall apply:

- the pulsation level at the line connection as specified in 5.3.7.2;
- inlet pressure vs liquid vapor pressure as defined in 5.3.7.3;
- pressure-limiting valve protection in case a pressure safety valve (PSV) is mounted on the pump, the pulsation suppression device, or pipe system as specified in 5.3.7.4;
- pressure drop as specified in 5.3.8.1.

NOTE Suitable acoustic boundary conditions are large-volume, large-diameter transitions, closed valves, valves with a significant pressure loss, and long undisturbed line. The acoustic simulation must always be used to evaluate whether or not a sufficient boundary exists.

5.3.5.2.1 A valve closure delay of at least 7 degrees after top dead center (TDC) shall be included in the acoustic simulation model(s) (refer to Annex J for additional information).

5.3.6 Mechanical Review and Mechanical Analysis

5.3.6.1 General

The type of mechanical analysis to be performed is specified by Table 3. Forced mechanical response is included when necessary.

5.3.6.2 Piping System Mechanical Review

A mechanical review shall be performed using span and basic vessel MNF calculations to avoid mechanical resonance. This review shall result in a table of various pipe sizes that indicates the maximum allowable span (based on the maximum pump operating speed) between piping supports as a function of pipe diameter, and the separation margin requirements of 5.3.8.2.

NOTE Descriptions of good practices regarding small-bore piping and support layout, vents and drains restraints, and small-bore piping restraints for PD pumps can be found in Annex F.

5.3.6.3 Mechanical Analysis

5.3.6.3.1 Step 1—Piping System MNF Analysis

An analysis of the piping system shall be done to predict the mechanical natural frequencies. The mechanical and acoustic system shall be designed to meet the separation margin criteria of 5.3.8.2, and the shaking forces shall not exceed the limits found in 5.3.7.5. The purpose of calculating piping system mechanical natural frequencies is to determine whether or not the separation margin guidelines are met. These calculations are usually made using one of two methods. The first method involves the use of fundamental beam theory and exact closed form solutions. This method is referred to as the fundamental analytical method (FAM). The second method involves the use of numerical solutions or finite element method (FEM). The accuracy of either method is greatly influenced by the assumed boundary conditions. When properly applied, the two methods should give similar results. If basic design concepts for piping layout are followed (e.g. minimize the number of bends, place a clamp near each bend and near all concentrated masses, etc.), the minimum natural frequency of most piping systems can readily be evaluated using the FAM approach. The FEM approach for calculating natural frequencies is only required for unusual layouts (long flexible spans with complex 3D bends between supports, etc.). However, if separation margin and shaking force guidelines are not met and the calculation of vibration amplitudes and dynamic stresses is required, the FEM approach is then employed.

NOTE The FEM model of the piping is best done concurrently with the thermal flexibility analysis.

The starting point of the mechanical model is the pump flange. The piping system model end points shall be defined by the designer in agreement with the purchaser. The piping system model shall (as a minimum) include all of the piping that was included in the acoustic simulation, including the piping up to the 2nd pipe support upstream/downstream of closed valves (e.g. PSV or block-valve) or a valve with more than 5 % pressure drop. An exception can be made for parts of the piping with low pulsation-induced forces (<10 % of allowable).

NOTE The intent is to avoid mechanical resonance of pulsation suppression devices and piping system at frequencies where significant shaking forces exist.

5.3.6.3.2 Step 2—Piping System Forced Mechanical Response

When the excitation frequency separation margins or the shaking force amplitude guidelines for the piping system cannot be met, a forced-mechanical response analysis of the piping system to acoustic shaking forces shall be performed. The allowable vibration and cyclic stress limits in 5.3.7.6 and 5.3.7.7, respectively, shall apply.

NOTE The vibration is generally the limiting design consideration, because when the vibration levels are within the recommended limits, the allowable stress levels are usually not approached. The exception is where high stress

concentrations occur at large-diameter reductions such as nozzle connections and weldolets for small-bore piping attached to significantly larger piping.

5.3.7 Design Criteria

5.3.7.1 General

Pulsation suppression devices and techniques applied in accordance with the Analysis Design Selection Criteria (5.3.4) shall satisfy the basic criteria in 5.3.7 and the other criteria in 5.3.8.

5.3.7.2 Allowable Pulsation Levels at and Beyond Line-side Connections of Pulsation Suppression Devices

The peak-to-peak pulsation levels in the suction and discharge piping systems beyond the pulsation control devices shall not exceed the levels calculated by Equation (14), which specifies the allowable peak-to-peak pulsation level of each individual pulsation frequency component.

In SI units:

$$P_1 = 3500 / (ID \times f)^{1/2} \quad (14)$$

In USC units:

$$P_1 = 100 / (ID \times f)^{1/2}$$

where

P_1 is the maximum allowable peak-to-peak pulsation level of the individual pulsation frequency components, expressed in kPa (psi), (suction and discharge pulsation levels also shall be limited to values that will not cause cavitation or relief valve lifting);

ID is the inside diameter of line pipe, in mm (in.);

f is the pulsation frequency, in Hertz, derived from the following equation:

$$f = (N)z / 60 \quad (15)$$

where

N is shaft speed (RPM);

z an integer that corresponds to the fundamental frequency ($z = 1$) and harmonics of the pump speed ($z = 2, 3$, etc.).

NOTE For multiple units in parallel, the purchaser and vendor shall consider and agree upon the additive effects of pulsation due to simultaneous operation of all pumps or the level of pulsation at a particular test point.

5.3.7.2.1 If pulsation levels exceed the limits defined by 5.3.7.2, then the shaking forces (5.3.7.5), vibration levels (5.3.7.6), or cyclic stress levels (5.3.7.7) shall be met. Maximum pulsation levels shall be approved by the purchaser.

5.3.7.2.2 Flow pulsations in elements sensitive to such phenomena (e.g. check valves and flow meters) shall be limited to agreed criteria.

5.3.7.3 Inlet Pressure vs Liquid Vapor Pressure

- 1) The time average NPIPA (net positive inlet pressure available) at the reference point shall be higher than the NPIPR (net positive inlet pressure required) as specified by the vendor in the pump datasheet. The NPIPA shall consider the pressure drop across the pulsation suppression device and associated piping between it and the pump flange.

NOTE The reference point is typically the vendor skid battery limit (pump inlet flange or skid tie in point).

- 2) Unless otherwise specified, the minimum value of the suction complex pressure wave $P_{\min}$ shall not be lower than the value as specified in Equation (16). This shall be applicable for the entire suction pipe system up to the pump valves.

NOTE An example plot for visualizing the above noted criteria is provided in Figure 3.

In SI units:

$$P_{\min} = P_{\text{vap}} + (0.03 \times P_{\text{mean}}) + 0.1 \quad (16)$$

where

P_{vap} is the highest liquid vapor pressure or the gas dissolution pressure, expressed in kPa (psia);

$P_{\min}$ is the minimum value of the pressure wave expressed in kPa (psia);

P_{mean} is the time average pressure expressed in kPa (psia).

NOTE The theoretical maximum amplitude of the suction pulsation occurs when the negative peak of the pulsation complex wave equals the average suction pressure minus the vapor pressure. Equation (16) provides for a margin of safety between the negative peak of pulsation and vapor pressure. See also Figure 3.

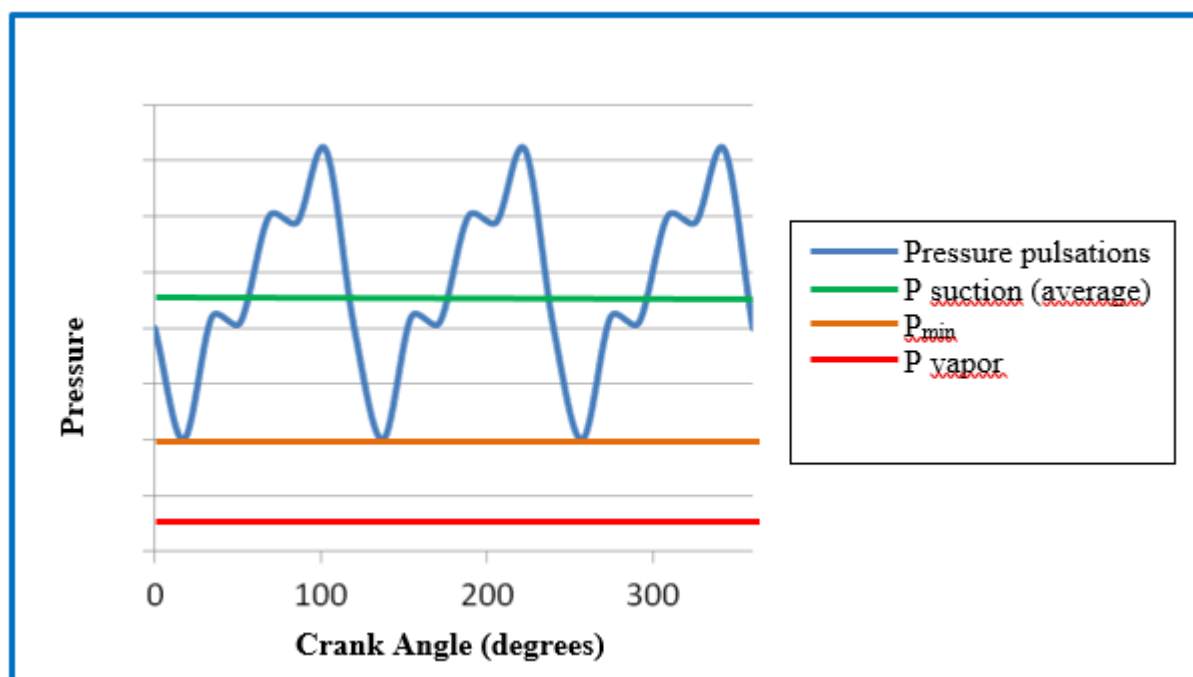


Figure 3—Suction Complex Wave for Triplex Pump Relative to Vapor Pressure

NOTE Entrained and/or dissolved gases can also significantly alter the cavitation characteristics of liquids due to a potential change in the speed of sound (see Annex J for additional information).

5.3.7.4 Pressure-limiting Valve Protection

Unless otherwise specified, the margin of separation between the positive peak of the pulsation complex wave at the relief valve and the relief valve setting shall be 5 % of the maximum working pressure or 165 kPa (25 psi), whichever is greater (Figure 4). See Equation (17).

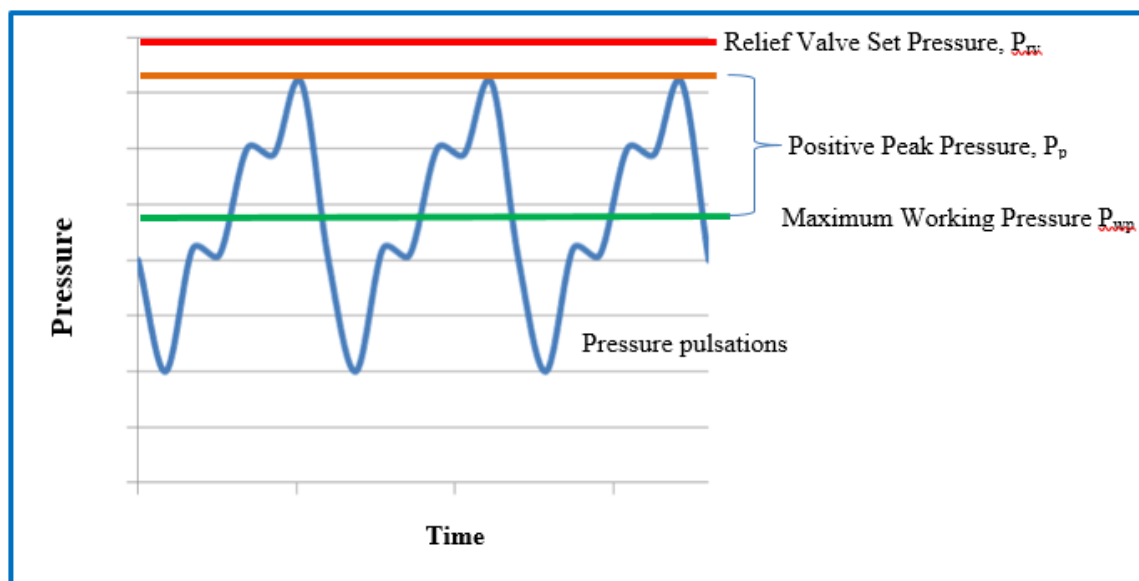


Figure 4—Pressure Pulsation at Relief Valve for a Non-cavitating Pump

$$P_p = P_{RV} - P_{WP} - (0.05 \times P_{WP}) \quad *$$
(17)

* Or use a pressure of 165 kPa (25 psi), whichever is greater.

where

P_{WP} is the maximum specified working pressure, expressed in kPa (psia);

P_p is the positive peak of the pulsation complex wave, expressed in kPa (psia);

P_{RV} is the required relief valve setting in gauge pressure, expressed in kPa (psia).

5.3.7.5 Allowable Piping System Acoustic Shaking Force

Table 5 shall be utilized as the criteria for acoustic shaking forces allowed in a given piping span.

Table 5—Allowable Shaking Force Guideline

Configuration	Allowable Shaking Force per Elbow Pair			
	Nonresonant ($SM > 20\%$)		Potentially Resonant ($SM < 20\%$)	
	USC Units (lbf, p-p)	SI Units (N, p-p)	USC Units (lbf, p-p)	SI Units (N, p-p)
Ground supported	250 × NPS	44.5 × DN	25 × NPS	4.45 × DN
Rack supported	20 × NPS	3.5 × DN	2 × NPS	0.35 × DN

5.3.7.6 Piping Design Vibration Criteria

For the forced response analysis of 5.3.6.3.2, the predicted piping vibration magnitude for each discrete frequency shall be limited to the design level in Figure 5. The diagram in Figure 5 is based on the following:

- a) a constant allowable vibration amplitude of 0.57 mm p-p (2.5 mils p-p) for frequencies below 10 Hz;
- b) a constant allowable vibration velocity of 36 mm/s p-p (1.41 in./s p-p) for frequencies between 10 Hz and 200 Hz.

The limits in Figure 5 are intended as a design trigger level for analysis in accordance with 5.3.6.3.2.

NOTE 1 Field acceptance criteria may differ from the design levels.

NOTE 2 The requirements in this section are considered to be conservative. There are, however, situations in which high-stress risers and unbraced small-diameter attached piping can pose a problem even though the main pipe exhibits acceptable vibration limits.

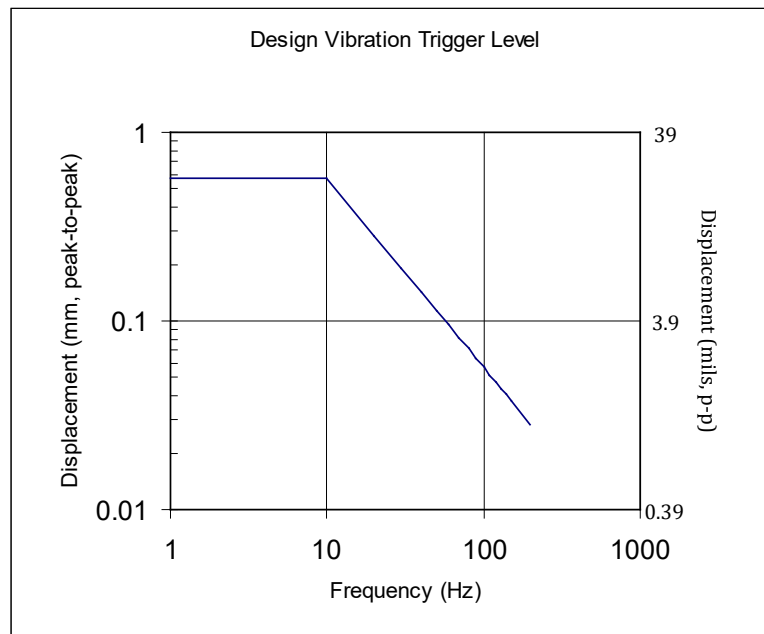


Figure 5—Piping Design Trigger Level at Discrete Frequencies

5.3.7.7 Maximum Allowable Cyclic Stress

For the forced response analysis of 5.3.6.3.2, pulsation-induced vibration shall not cause a cyclic stress level in the piping and pulsation suppression device nozzles in excess of the endurance limits of materials used for components subject to these cyclic loads. For carbon steel with an operating temperature below 370 °C (700 °F), the peak-to-peak cyclic stress shall be less than 96 MPa (14,000 psi) considering all stress concentration factors present and with all other stresses within applicable code limits. Endurance limits for materials other than carbon steel shall be as defined in the specified pressure vessel code.

NOTE The above carbon steel cyclic stress value is from 2010 ASME *BPVC* Section III, Appendix 1.

5.3.8 Other Criteria

5.3.8.1 Allowable Pressure Drop

Allowable pressure drop through the pulsation suppression devices for the suction and discharge system is defined as follows.

- Unless otherwise specified, the steady flow pressure drop through the suction pulsation suppression device shall not exceed 2 % of mean absolute line pressure or 15 kPa (2 psi), whichever is greater.
- Unless otherwise specified, the steady flow pressure drop through the discharge pulsation suppression device shall not exceed 2 % of mean absolute line pressure.

NOTE The allowable pressure drop specified in this section can be exceeded by agreement between purchaser and vendor when this is the consequence of the preferred pulsation control design.

5.3.8.2 Separation Margins

The guidelines described below are to be used to avoid coincidence of excitation frequencies with mechanical natural frequencies of the pump, pulsation suppression devices, and piping system:

- a) the minimum MNF of any pump or piping system element shall be designed to be greater than $2.4\times$ the fundamental frequency at maximum rated speed, and
- b) the predicted mechanical natural frequencies shall be designed to be separated from significant excitation frequencies by at least 20 %.

NOTE 1 The intent is to be above twice running speed, because there is generally sufficient excitation energy at the first and second orders due to mechanical unbalance loads to excite resonances to an unacceptable level.

NOTE 2 Significant excitation sources include pulsation shaking forces at multiples of the primary pulsation excitation frequency [see Equation (13)].

5.3.8.3 Flow Measurement Error

For flow meters located in the piping system, the maximum flow measurement error caused by flow pulsations shall not exceed the following.

- a) For non-custody transfer meters: 1.00 % error.
- b) For custody transfer meters: 0.125 % error.

Refer to Annex G regarding allowable pulsation and/or velocity fluctuations at the flow meter that must be maintained such that the above noted meter errors will not be exceeded.

5.3.8.4 Documentation Requirements

A written report on the control of pulsation and vibration shall be furnished to the purchaser. Compliance with the requirements of 5.3 for the specified design approach shall be documented. The report shall define the analysis scope, including analysis guidelines, compressor configuration, operating conditions, gas composition, and extent of the piping system analyzed. The report shall include the recommendations resulting from the analysis. The documentation shall also present results applicable to each type of analysis performed. Acoustic simulation results include piping pulsation and flow pulsation at equipment sensitive to this in spectrum form. MNF analysis results include natural frequencies and mode shapes. Forced mechanical response results include vibration and cyclic stress. The format of the results presentation should permit easy comparison with the analysis guidelines.

5.4 Pulsation and Vibration Control Techniques for PD Pumps—Rotary

5.4.1 General

Due to the fundamental operating principle of rotary PD pumps (continuous rotary pumping action, near-constant volumetric flow with limited internal slip without the need for separate inlet and outlet valves), pulsations do not normally occur or are negligible for 100 % liquid operation, assuming that sufficient NPIP is provided to the pump inlet to prevent cavitation. In instances where extreme pulsation-induced vibration has been observed when handling 100 % liquid, it has generally been linked to a resonance within the piping system. In cases where some percentage of compressible medium is anticipated within the incoming process stream on a continuous or intermittent basis, the possibility for pulsation-induced vibration of greater amplitudes may be more substantial. In all cases, the acceptable pump vibration limits shall be in accordance with API 676.

5.4.2 Compressible Fluids

When volumetric flow is delivered by a rotary pump into the process system, it creates a resistance in the piping system. This resistance creates a measureable differential pressure between the outlet and the inlet of the pump, which provides the drive mechanism to force a portion of the pumped liquid (identified as slip) back through the internal pump clearances. Compressible fluids entering the enclosed pumping cavities will experience a reduction in volume, and this volume is filled up by the internal fluid slip within the pump. Depending on the volume of compressible fluid present as a function of the total inlet volume and the absolute pressure ratio between the inlet and outlet pressures, pulsation-induced vibration may occur in excess of the acceptable vibration limits mentioned above. It is critical at this stage that the purchaser differentiate between expected transient operating conditions and the expected steady state conditions when assessing potential excitation forces. When the process conditions are a combination of high gas void fraction (GVF), high differential pressure, and/or high viscosity of the liquid phase, there is a greater potential for vibration issues. In these cases, further discussions with the supplier may be warranted to review the possibility for pulsations and resultant vibrations.

5.4.3 Analysis Approach

5.4.3.1 General

Should the anticipated vibration levels exceed the API 676 limits under steady state operation, the analytical approaches listed in 5.4.3.2 and 5.4.3.3 should be considered by the purchaser accepting the fact that they may not yield sufficient accuracy if the following conditions apply.

- Inlet flow velocities below 0.3 m/s (1 ft/s) or above 3 m/s (10 ft/s).
- The inlet liquid temperature is high enough that cavitation can be anticipated.

When deciding which approach should be used, the purchaser should consider such things as the flow characteristics as well as the GVF range and variability, the required input power and operating speed range, the absolute pressure ratio between pump inlet and outlet, the piping layout, unit reliability, and experience with similar pumps and installations. Should any one of these variables fall outside the experience of the purchaser, additional analysis should be considered.

5.4.3.2 Mechanical Piping Review

The piping review shall include a simplified analysis of the piping system with excitation frequency data from the pump vendor to determine critical piping lengths that may be in resonance with the acoustical excitation frequencies. This review results in a table of various pipe sizes that indicates the maximum allowable span (based on the maximum pump operating speed) between piping supports as a function of pipe diameter.

5.4.3.3 Allowable Shaking Forces

The following table (Table 6) defines the allowable shaking forces for the piping system. The supports/restraints shall be properly designed to withstand these force levels.

Table 6—Allowable Shaking Force Guideline

Configuration	Allowable Shaking Force per Elbow Pair			
	Nonresonant ($SM > 20\%$)		Potentially Resonant ($SM < 20\%$)	
	USC Units (lbf, p-p)	SI Units (N, p-p)	USC Units (lbf, p-p)	SI Units (N, p-p)
Ground supported	$250 \times \text{NPS}$	$44.5 \times \text{DN}$	$25 \times \text{NPS}$	$4.45 \times \text{DN}$
Rack supported	$20 \times \text{NPS}$	$3.5 \times \text{DN}$	$2 \times \text{NPS}$	$0.35 \times \text{DN}$

In the piping design, when clamps are used to avoid mechanical resonance, the thermal flexibility effects and static stresses should also be considered. To accurately predict and avoid piping resonance, the supports and clamps must rigidly restrain the piping. The piping restraint is not considered to be rigid unless the restraints have either enough mass or stiffness sufficient to enforce a vibration node at the restraint and the pipe is attached to the restraint using clamps. This requirement is difficult to attain with overhead piping and/or the use of simple supports, hangers, and guides (refer to Annex A for additional discussion of pipe support details).

Annex A

(informative)

Description of Work Processes for Reciprocating Compressors and Plunger Pumps

A.1 Scope

This annex is intended to provide insight on important concepts involved in the design of pulsation and vibration control for PD machinery systems. The content of Annex A was originally published as API 688, First Edition.

The purpose of this document is to provide guidance on the application of pulsation and vibration control requirements now found in the normative sections of API 688.

This document is not intended to be an all-inclusive source of information for this complex subject. Rather, it is offered as an introduction to the major aspects of pulsation and vibration control for PD machinery addressed during a typical system design. A significant amount of the material has been extracted from documents previously published by the contributors. The different design philosophies of the various contributors are consolidated in this document to help users understand the choices available and make informed decisions about what is appropriate for their application. While the theory is generally applicable to all types of PD machinery, the text in this annex will frequently refer specifically to reciprocating compressors.

A.2 Fundamentals of Pulsation and Mechanical Vibration Theory

A.2.1 General

In order to understand how to control pulsation and vibration in PD machinery systems, it is imperative that one understands the differences between acoustical and mechanical concepts. In A.2.4, the acoustic issues, along with acoustic control techniques, will be addressed. In A.2.5, the elements of the mechanical system will be explained along with the concept of acoustic-mechanical coupling and the mechanical techniques for controlling vibration.

A.2.2 Overview of Pulsation Concepts

Pressure variations that result from oscillatory flow of PD machinery are the subject of this section. These variations in pressure are referred to as pulsation. Pulsation occurs in systems handling both gases and liquids. High vibration, support degradation, and fatigue failures caused by dynamic forces induced by the pulsation are the most common problems resulting from pulsation. In order to reduce the possibility of detrimental pulsation and vibration at the design stage, it is necessary to understand several technical concepts.

Excitation mechanisms are first addressed in A.2.3. Acoustic response is then explained in A.2.4. In A.2.5, the most common results of excessive pulsation are reviewed and the concept of acoustic-mechanical coupling is explained. Closing the discussion about pulsation concepts is a review of the techniques used for pulsation control in A.2.8.

A.2.3 Excitation Sources

In systems utilizing PD machinery, the flow of gas or liquid is not steady. Instead, the fluid moves through the piping in a series of flow pulses (dynamic or time varying), which are superimposed upon the steady (average) flow.

As an example, the magnitude and shape of the flow pulses through the compressor valves in a reciprocating compressor cylinder are determined by physical, geometrical, and mechanical characteristics of the compressor (rotational speed, bore, stroke, loading, compression ratio, etc.). These flow pulses act as excitations that create

pressure and flow modulations (acoustic waves) that move through the process fluid as it moves through the piping system. Generally, the predominant pressure and flow modulations generated by a reciprocating compressor are at frequencies that can be modeled as 1D waves.

An important part of the acoustic analysis is the development of a compressor model that accurately predicts the dynamic flow excitation (flow vs time) delivered by the compressor. Some simplified examples are shown in Figure A.1, Figure A.2, Figure A.3, and Figure A.4.

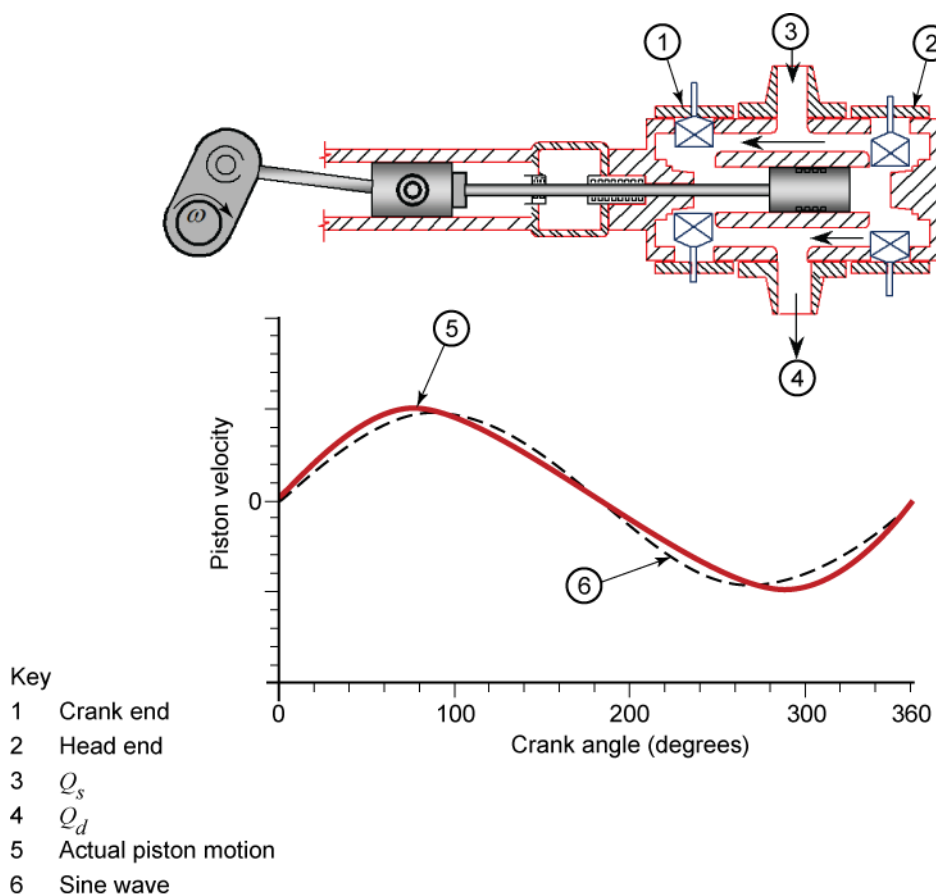


Figure A.1—Piston Motion and Velocity for a Slider Crank Mechanism

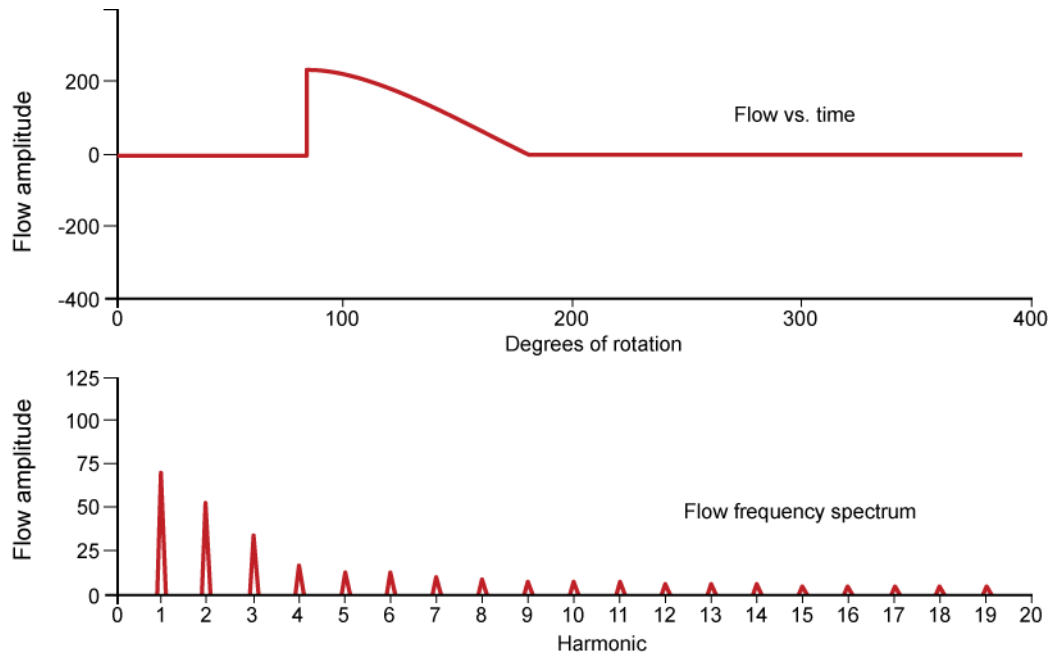


Figure A.2—Single-acting Compressor Cylinder with Rod Length/Stroke = ∞ and No Valve Losses

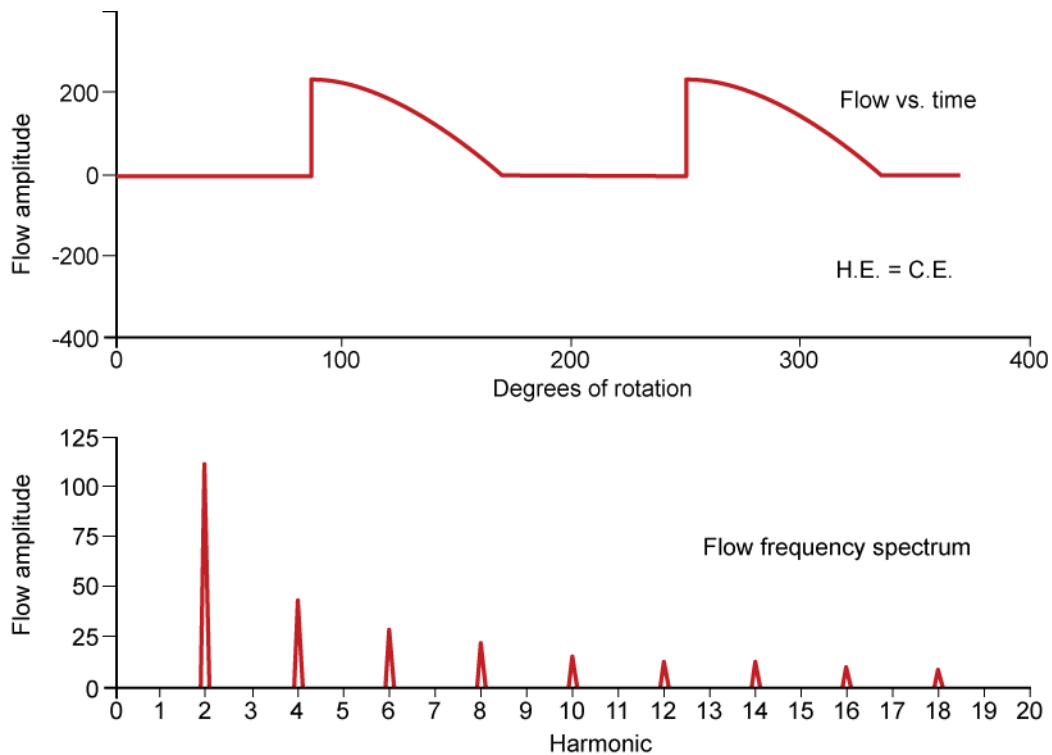


**Figure A.3—Symmetrical, Double-acting Compressor Cylinder
with Rod Length/Stroke = ∞ and No Valve Losses**

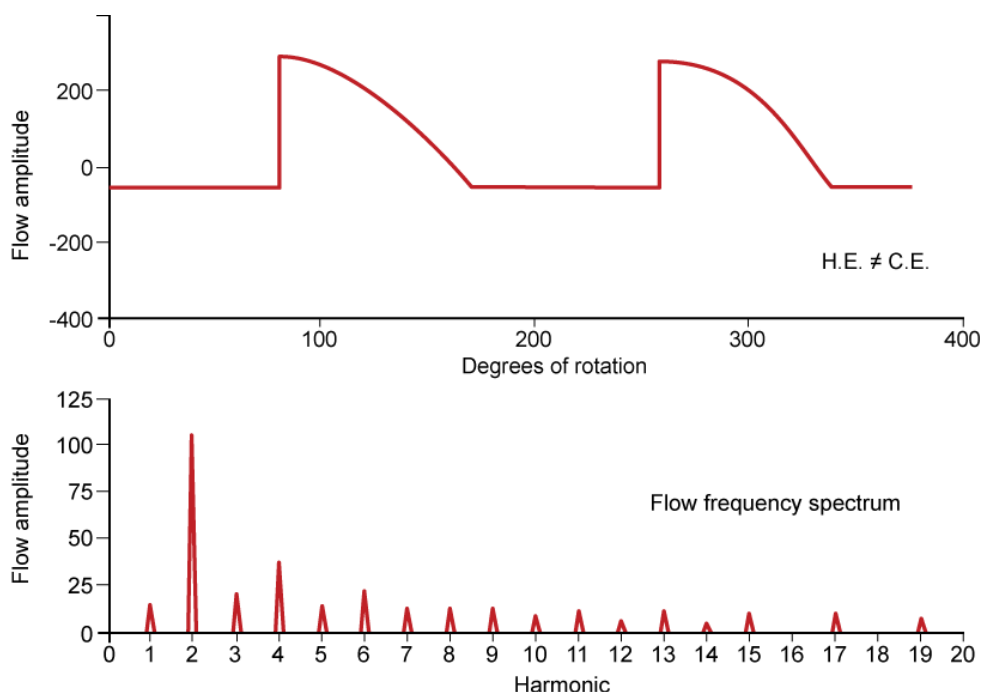
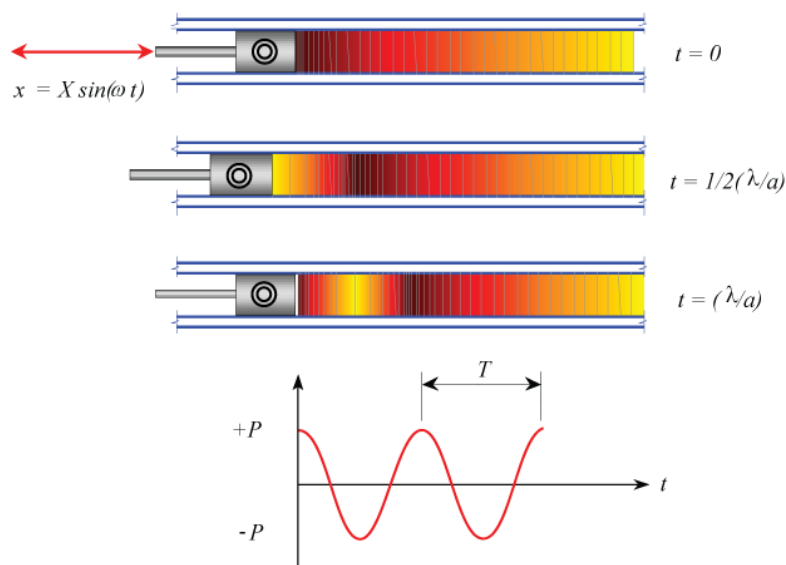


Figure A.4—Unsymmetrical, Double-acting Compressor Cylinder with Rod Length/Stroke = 5 and No Valve Losses

A.2.4 Acoustic Response and Resonance

The flow pulses caused by the reciprocating action of the compressor or pump create pressure pulses or waves that move through the piping system as shown in Figure A.5.



where

λ is the wavelength = a/f ;

a is the acoustic velocity;

f is the frequency = $1/t$;

T is the time period of one cycle;

t is the time.

Figure A.5—Traveling Wave in Infinite Length Pipe

While the flow pulse frequencies generated by the compressor are a function of the mechanical properties of the compressor, the acoustical response in the piping is a function of the mechanical properties of the compressor, the thermophysical properties of the gas, and the acoustical network defined by the attached piping.

When a particular harmonic of running speed is near or coincident with an acoustical natural frequency, the acoustic response (dynamic pressure amplitude) is amplified. These resonances that occur when an excitation frequency coincides with a natural frequency can be simple organ-pipe type resonances or complex modes involving all of the piping.

For simple constant diameter lines with open and/or closed boundary conditions, specific pipe lengths determine acoustical natural frequencies. If a line length coincides with integer multiples of one half or one quarter of the wavelength, depending on the combination of open or closed end conditions, an acoustical resonance can be excited. End conditions are defined as either open or closed. For half-wave resonances, both end conditions must be the same, i.e. open-open or closed-closed. For quarter-wave resonances, the end conditions must be opposite, i.e. one open end and one closed end. Examples of these configurations are shown in Figure A.6 and Figure A.7 and are defined by Equation (A.1) and Equation (A.2).

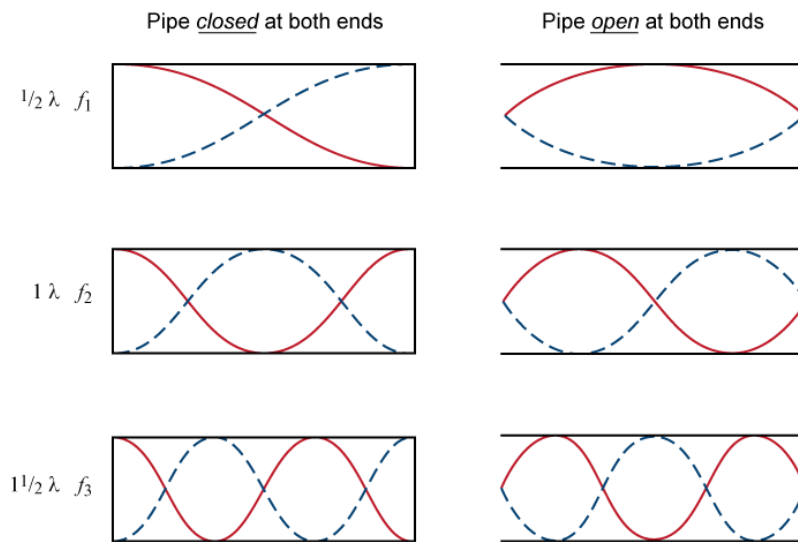


Figure A.6—Mode Shapes of Half-wave Responses

Pipe open at one end and closed at the other end

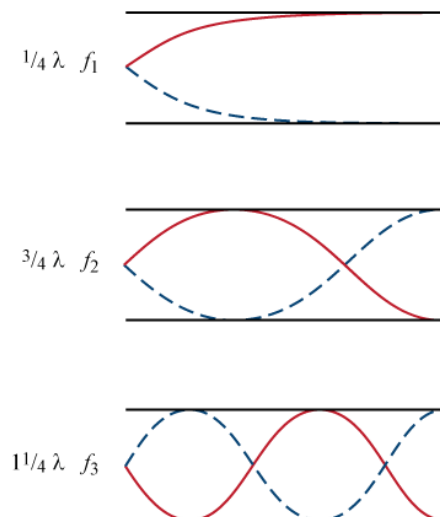


Figure A.7—Mode Shapes of Quarter-wave Responses

Formula for half wave (closed-closed and open-open acoustic response frequency):

$$f = \frac{na}{2L} \quad (\text{A.1})$$

where

f is the acoustical natural frequency (subscript indicates order);

n is 1, 2, 3,...;

a is the speed of sound;

L is the length of pipe.

Formula for quarter wave (open-closed acoustic response frequency):

$$f = \frac{na}{4L} \quad (\text{A.2})$$

where

f is the acoustical natural frequency (subscript indicates order);

n is 1, 3, 5,...(odd integers);

a is the speed of sound;

L is the length of pipe.

A.2.5 Effects of Pulsation

A.2.5.1 General

Pulsation, which is produced by a moving element such as the piston of a PD pump or compressor, propagates through the suction and discharge valves, the internal gas passages, the pulsation suppression devices, and the attached suction and discharge piping systems. The types of problems that can result depend upon the magnitude and frequency of the pulsation. Vibration, fatigue failures, performance degradation, driver overload, and flow metering inaccuracy are some of the problems that result from excessive pulsation. Each of these will be discussed in more detail in the following sections.

A.2.5.2 Shaking Force

Pulsation itself will not produce vibration of the piping system; points of acoustical-mechanical coupling are necessary to develop a dynamic force, which in turn produces the vibration. Geometric discontinuities such as elbows, reducers, tees, and capped ends are common force-coupling points in compressor piping systems.

Elbows or bends have differential areas due to outside and inside radii differences, which result in dynamic forces in the plane of the elbow when pulsation is present. Therefore, when possible, the number of elbows in a piping system should be minimized to reduce the number of force coupling points. Location of dynamic restraints (clamps) near elbows is desirable since dynamic forces will occur at these locations.

Force coupling points occur at reducers, tees, and other points in the piping having unbalance areas. Probably the most easily understood case of a force coupling unbalanced area is that of a capped end. This condition

occurs at the ends of all vessels, the compressor pulsation suppression device being one such vessel. The forces acting on the closed ends of the volume bottles can produce high vibration of the bottles themselves.

Equipment, vessels, and pipe runs can typically be considered rigid along their axis, thus the effective shaking force is the sum of pulsation multiplied by unbalanced area at each geometric discontinuity. The pulsation magnitude and phase angle and the force direction at each unbalanced area must be considered as shown in Figure A.8, Figure A.9, Figure A.10, Figure A.11, Figure A.12, Figure A.13, and Figure A.14. The figures illustrate examples of dynamic shaking forces at points of acoustic-mechanical coupling.

Equations used in Figure A.8:

$$F = P_{\text{dyn}} \left[\frac{\pi}{4} D^2 \right] \quad (\text{A.3})$$

$$F_{\text{Result}} = 2P_{\text{dyn}} \frac{\pi}{4} D^2 \cos\left(\frac{\varphi}{2}\right) \quad (\text{A.4})$$

where

- F is the shaking force;
- P_{dyn} is the dynamic pressure;
- D is the internal diameter;
- F_{Result} is the resultant shaking force;
- φ is the elbow angle.

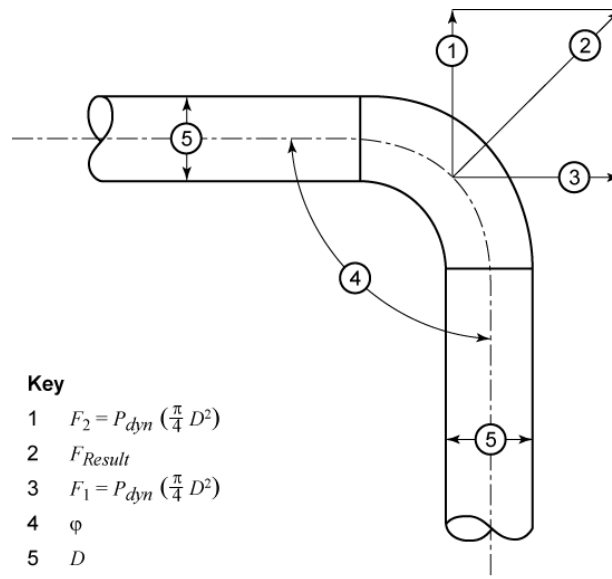


Figure A.8—Elbow with Dynamic Forces

Equation used in Figure A.9:

$$F = P_{\text{dyn}} \times \frac{\pi}{4} (D_1^2 - D_2^2) \quad (\text{A.5})$$

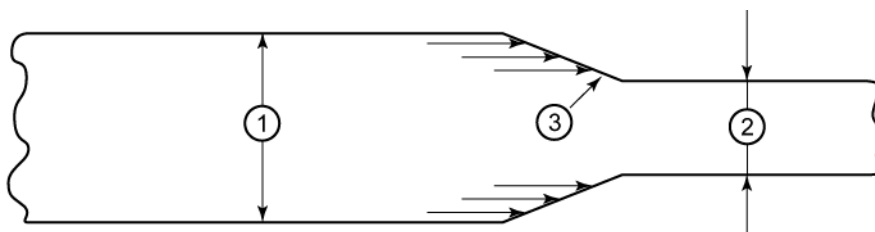
where

F is shaking force;

P_{dyn} is dynamic pressure;

D_1 is internal diameter of larger pipe;

D_2 is internal diameter of smaller pipe.



Key

1 D_1

2 D_2

3 P_{dynamic}

Figure A.9—Reducer with Dynamic Forces

Equation used in Figure A.10:

$$F = P_{\text{dyn}} \left[\frac{\pi}{4} D^2 \right] \quad (\text{A.6})$$

where

F is the shaking force;

P_{dyn} is the dynamic pressure;

D is the internal diameter of pipe.