EFRC Report

EFRC

EUROPEAN FORUM
for RECIPROCATING
COMPRESSORS

Review of the sixth edition of the API 618 and the second edition of the API 688

Part 2: Second edition of the API 688

First edition, May 2025

Contents

1 Introduction to part 2 of this EFRC report

1.1 General information on the second edition of the API 688

The first edition of the API *Recommended Practice* (RP) 688 “Pulsation and Vibration Control in Positive Displacement Machinery Systems for Petroleum, Petrochemical, and Natural Gas Industry Services”, has been released in 2012. This first edition of the API RP 688 was intended to describe, discuss and clarify the design of pulsation and vibration control for positive displacement machinery systems used for services in the petroleum, petrochemical and natural gas industries. The original focus was to provide insight on the significant changes to the PVC sections in Clause 7.9 of the fifth edition of API 618 for reciprocating compressors. Furthermore, additional information has been given on pulsations and vibrations of reciprocating compressor and positive displacement reciprocating pumps as stipulated in the API 674/675/676and API 619.

An important change of the second edition of the API 688 is that it is now a standard instead of a recommended practice (RP) and includes now also specific industry requirements. The concept behind the second edition was to group the similar requirements for PVC of distinct types of machinery into one document. Another advantage is that future adjustments on pulsation and vibration control need only to be done in one document, the API 688, and no longer in the aforementioned standards. The PVC sections of the next editions of the particular standards would then no longer be part of that standard and instead there will be a reference to the particular section of the second (and higher) edition of the API 688.
For those reasons, the PVC clauses of the API 618 (reciprocating compressors), the API 619 (Rotary type PD compressors (screw compressors), the API 674 (PD pumps-reciprocating), API 675 (PD pumps-controlled volume) and the API 676 (PD pumps-rotary) have been included in the second edition of the API 688 which is a significant improvement.

1.2 Organisation of part 2 of this EFRC report

The sections of the second edition of the API 688 have been compared to the latest editions of the applicable standards for the different type of positive displacement machinery and are as follows: sixth edition of the API 618, fifth edition of the API 619, third edition of the API 674, third edition of the API 675, and fourth edition of the API 676.
It is noted that the sections with *general* items of the first part of the second edition of the API 688 have been compared to those of the first edition of the API 688. The section numbers of this EFRC report (not the API 688 section numbers!) are summarised in Table 1.1

Table 1.1 EFRC report section numbers for the general parts of the second edition of the API 688

EFRC report chapter number	Title
2.1	Special Notes
2.2	Forword
2.3	Important Information Concerning Use of Asbestos or Alternative Materials (New)
2.4	Introduction of the 2nd edition of the API 688
2.5	Scope
2.6	Normative references
2.7	Terms, Definitions, Acronyms, and Abbreviations
2.8	General

There are sections in every standard that have been moved, removed, added, and so on, which makes a review between two editions challenging. For this reason, the cross-reference number of the edition with which the latest version for the same topic has been compared is indicated in parentheses in each section. An example is section 5.1.7.7.1 (Permissible cyclic stress) of the second edition of the API 688, which is section 7.11.7.6.1 of the sixth edition of the API 618. In some cases, the title of the section has also been changed, and for those cases, the title of the section is also listed in parentheses.

The following distinctions have been made in review status:

1. If the requirement description has not been adjusted the review status is *Unchanged.*
2. If the requirement description has been partly or completely adjusted without a change of content the review status is *Requirement description has been adjusted with preservation of content.*
3. If the requirement description has been partly or completely adjusted with a change of content a short or complete description has been given of the adjustment(s).
4. The review status is *New* for a requirement which has been added and for those situations the new requirement is given in its entirety.

For all API sections with a review status of 3 (partial or complete adjustment) or 4 (new), a detailed description has been provided because these adjustments may have an effect on the design of (parts of) the system or its operation in practice, and it is the user's responsibility to determine this. To keep the document readable, concise and clear, the detailed descriptions are briefly or completely summarized in tabular form. In this way, the adjustments that are important to the user are quickly accessible.

> Identical to part 1 of this EFRC report, additional explanations or comments on certain topics have been provided which are highlighted in blue with indentation.

An overview of the chapter numbers for part 2 of this EFRC report is shown in Table 1.2.

Table 1.2 Chapter numbers for Part 2 of this EFRC report (API 688)

EFRC report chapter number	**Title**
1	Introduction to part 2 of this EFRC report
2	Summary of the adjustments of the general parts of the 2nd edition compared to the 1st edition of the API 688.
3	Summary of the adjustments of the requirements of the 2nd edition of the API 688 compared to the 6th edition of the API 618
4	Summary of the adjustments of the requirements of the 2nd edition of the API 688 compared to the 5th edition of the API 619.
5	Summary of the adjustments of the requirements of the 2nd edition of the API 688 compared to the 3rd edition of the API 674/675
6	Summary of the adjustments of the requirements of the 2nd edition of the API 688 compared to the 4th edition of the API 676
7	Annexes of the second edition of the API 688
8	References
Annex A	Summary of detailed descriptions of adjustments of API 688 Section 5.1 (API 618)
Annex B	Summary of detailed descriptions of adjustments of API 688 Section 5.2 (API 619)

2 Summary of adjustments of the general parts of the 2nd edition compared to the 1st edition of the API 688

Chapter 2.1 through 4 of this EFRC report contains the *general* parts of the second edition of the API 688 which have been compared to those of the first edition of the API 688.

2.1 Special Notes (New)

This is a now a new part and contains some general information and it is recommended to read them carefully.
A possible reason that this part was missing in the first edition of API 688 could be that the first edition is a Recommended Practice instead of a Standard.

2.2 Foreword (Adjusted)

The original focus of the first edition of the API 688 was to provide insight on the many changes to the pulsation and vibration material in the Clause 7.9 of the 5th Edition of API 618 for reciprocating compressors only. This paragraph is no longer included in the second edition of the API 688 and has been replaced by information about verbal forms (shall, should, ay, can) and some general information about reviewed and revised, reaffirmed, or withdrawn versions of API standards.

2.3 Important Information Concerning Use of Asbestos or Alternative Materials (New)

This new section has been added to the second edition of the API 688, and it is recommended to read this section carefully.

2.4 Introduction (New)

This is now a new section describing that the users of this standard should be aware that further or differing requirements may be needed for in dividual 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.

Furthermore, it gives a summary of the different API 688 Annexes for the different positive displacement machinery for which the standard has been developed.

2.5 1 Scope (1)

The scope has been adjusted and gives now an overview of machines which are covered in this standard:

- 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).

2.6 2 Normative references (New)

The following normative references have been added. 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

2.7 3 Terms, Definitions, Acronyms, and Abbreviations (section 2 of the first edition of the API 688: Definitions of Terms)

Definitions no longer included:

- 2.4 analog simulation
- 2.5 coefficient of restitution
- 2.6 combined rod load
- 2.7 damping
- 2.8 design
- 2.9 digital acoustic simulation
- 2.10 frequency
- 2.18 pulsation spectral frequency distribution
- 3.1.1 acoustic simulation (2.1)
  Parts of the definition has been adjusted with preservation of content.

3.1.2 active analysis (2.2)
Unchanged

3.1.3. amplification factor (2.3)
Unchanged

3.1.14 infinite length line (boundary condition) (2.11)

3.1.15 line-side connections (New)
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 (2.12)
Some parts of the definitions are no longer included with preservation of content.

3.1.17 mode shape of an acoustic resonance (2.13)
Unchanged

3.1.18 natural frequency (2.14)
The definition has been adjusted to: “The frequency at which a system tends to oscillate in the absence of any driving force.”

3.1.19 normal operating condition (New)
Point at which usual operation is expected, and optimum efficiency is desired, usually the certified point.

3.1.20 oil stiction (2.15)
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 (New)
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 (New)
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 (2.16)
Unchanged

3.1.24 pocket passing frequency PPF (New)
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 (New)
Recognized pressure vessel standard specified or agreed by the purchaser.

3.1.26 pulsation suppression device (New)
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 (New)
Pressure at which a relief valve starts to lift.

3.1.28 resonance (2.17)
Unchanged

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

3.1.30 separator (New)
Pressure-containing device, usually a vessel, used to separate a fluid from the process gas.

3.1.31 silencer (New)
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 (New)

3.1.33 stiffness (2.19)
Achieved by holding the suction valves completely open for a calculated duration of time during the compression stroke to vary compressor cylinder capacity.

3.1.34 suction stabilizer (New)
A term commonly used to refer to the pulsation suppression device on the suction side of a PD pump.

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

3.1.36 support (New)
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 (New)
Ratio of internal compressor suction volume to discharge volume, expressed as Vi. 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 (New)
See the standard for an overview.

2.8 4 General

4.1 Dimensions and Units (New)
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 (New)
The supplier shall assume responsibility and shall assure that all sub-vendors comply with the requirements of this standard and all referenced documents.

Supplier is not defined in this section. A supplier can be the supplier of the compressors system or the API pulsation study. In the API 618 a distinction is made between vendor and purchaser which are better definitions to use.

3 Review summary of reciprocating compressors (API 618)

3.1 5 Requirements

3.2 5.1 Reciprocating Compressors (Ref. API 618)

The pulsation and vibration control section 5.1 of the second edition of the API 688 has been compared to section 7.11 of the sixth edition of the API 618.
Several sections on pulsation and vibration control of the second edition of the API 688 has been adjusted compared to the sixth edition of the API 618 (section 7.11). It mainly concerns sections on stepless flow control (new), shaking force control (adjusted), piping system model (adjusted), low cycle fatigue analysis (new), gas rod load induced forces (new), allowable pulsations at compressor valves (new), allowable piping cyclic stress (adjusted).
A number of important Annexes have also been added that provide a more detailed explanation of important topics such as:

- Annex B (*normative*) Stepless Capacity Control for Reciprocating Compressor Cylinders.
- Annex E (informative) Mechanical Forced Response Analysis.
- Annex F (informative) Small-bore Piping Design and Analysis.

Table 3.1 provides a summary of the most important changes and a summary of the detailed descriptions of the adjustments have been given in Appendix A of this report.

Table 3.1 Summary of adjustments of the 2nd edition of API 688 compared to the 6th edition of API 618

Section nr 6th edition API 618	**Section nr 2nd edition API 688**	**Subject**	**Description/adjustments of the second edition of the API 688**
7.11.1	5.1.1.	General	This is a completely new description of section 7.11.1 of the sixth edition of the API 618. This session describes now the different speeds of the compressors for which the pulsation and vibration control may be applied. The detailed requirement can be found in section 5.1.1
7.11.2.1	5.1.2.1	Loading steps for stepless flow control	The loading steps for stepless capacity control has been added and the number of steps for this unloading device has been specified as follows: “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”
7.11.2.1	5.1.2.2	Alternate operation conditions	This section contains now the last requirement of section 7.11.2.1 of the sixth edition of the API 618: “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”
7.11.5.2	5.1.5.2	Compressor system acoustical model	The following requirements has been added: “When stepless capacity control is used, the frequency range shall be sufficient to cover the higher harmonics as generated by the stepless control system”

7.11.5.3	5.1.5.3	Piping system model	The following requirements has been added: “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.” Identical to section 7.11.5.3 (Piping System Model) of the sixth edition of the API 618 as summarised in Part 1 of this EFRC report, the reference of this section to the design limits of section 5.1.7.4 may be confusing, see detailed description **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A of this report.**
7.11.5.4	5.1.5.4	Pre-study	**Unchanged but it is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A.**
7.11.6	5.1.6	Design Approach 3	**Unchanged but it is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A.**
7.11.6.1.2	5.1.6.1.1	Stresses in pulsation suppression devices	The following requirement has been added to the existing requirement: “… that result from pulsation-induced shaking forces and pressure-induced static force.”
-	5.1.6.1.1 (New)	Low cycle fatigue analysis	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.
7.11.6.2	5.1.6.2	Step 3a MNF analysis	**Unchanged but it is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A.**
7.11.6.2	5.1.6.2.1	MNF Analysis Compressor system and piping	A detailed description has been added about the FAM and FEM method. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A of this report.**
5.1.6.3.1	5.1.6.3.1	Forced mechanical response analysis of the compressor system	It has been indicated that 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. This is not correct because the allowable vibration level may be exceeded but, in that case, a cyclic stress calculation shall be conducted. The piping in this context consists of the piping and the pulsation damper that are attached to the compressor cylinder
7.11.6.3.2	5.1.6.3.2	Vibration limits of compressor cylinder and frame	The reference to the European Forum for Reciprocating Compressors (EFRC) vibration guidelines is no longer included and this is replaced by the following requirement: “The allowable vibration limits for the compressor cylinders and frame shall be supplied by the OEM”
7.11.6.4	5.1.6.4.1	Exceeding separation or shaking force criteria	**Unchanged but it is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A.**
-	5.1.7.3 (New)	Allowable pulsations at compressor 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.

7.11.7.4.2	5.1.7.5.2	Allowable piping system acoustic shaking force	Compared to the sixth edition of the API 618, a further distinction has been made between ground and rack supported systems and potentially resonant (separation margin smaller (SM) than 20%) and potentially non-resonant (separation margin smaller than 20%). A distinction has now been made between a pipe diameter indicated in NPS and in DN. It is common to identify pipes by inches using NPS or " Nominal Pipe Size". The metric equivalent is DN or "Diametre Nominel". The metric designations conform to International Standards Organization ISO (ISO 6708) usage and apply to all plumbing, natural gas, heating oil, and miscellaneous piping used in buildings. An overview of the NPS and DN outer diameters is shown in Table 3.2 in the detailed description. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A of this report.**
7.11.7.4.3	5.1.7.5.3	Allowable cylinder mounted pulsation suppression device shaking force.	The value of SF_{dmax} has been adjusted from 45,000 (N) of equation (12) of the sixth edition of the API 618 to equation (8): $SF_{dmax} = 5\sqrt{MACCRLx\ Number\ of\ Cylinders}$ (8) – SF_{dmax} maximum pulsation suppression device non-resonant peak-to-peak force in N (lbf). – MACCRL is the maximum allowable continuous combined rod load defined in API 618 in N (lbf). – Number of Cylinders to which the pulsation suppression device is attached. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A of this report.**
-	5.1.7.5.4 (New)	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.
7.11.7.5	5.1.7.6	Piping design vibration criteria	All values of vibration in this standard are peak-to-peak values and the user shall be aware of the fact that in b) the value is now given in 18 mm/s 0-p which may be overlooked.
7.11.7.6.1	5.1.7.7.1	Allowable cyclic stress	The allowable peak-to-peak cyclic stress has now been reduced from 180 N/mm^2 to 96 N/mm^2 considering all stress concentration factors present and with all other stresses within applicable code limits. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A of this report.**
• 7.11.7.6.2	5.1.7.7.2	Piping design standards	This section is no longer preceded by a bullet. The specified piping code providing the design criteria, has now been added (e.g. ASME B31.3, ASME B31.8, etc.).
-	5.1.10	Summary of design steps (New)	Historical descriptions (the M options of the 4thedition of the API 618) of the commonly used analysis methods for pulsation and vibration control have now been added. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex A of this report.**

4 Review summary of rotary type PD compressors (API 619)

4.1 5.2 Rotary Type PD Compressors (Ref. API 619) (6.9 Pulsation suppressors/silencers for dry screw compressors)

The pulsation and vibration control section 5.2 of the second edition of the API 688 has been compared to section 6.9 of the fifth edition of the API 619 "Pulsation suppressors/silencers for dry screw compressors".
Due to the fact that the flows, forces, pressures, and speeds have been increased the last decades, there is a greater chance of pulsation and vibration problems, especially on the discharge side. In addition, compared to other types of positive displacement machines, the excitation frequencies are higher and more difficult to control.
For these reasons, the section on pulsation and vibration control of rotary compressors has been adjusted and expanded with new requirements, mainly on the number of harmonics to be analysed (new), mechanical analysis of silencers (new), separation margin criterion and pocket passing frequency (PPF) of discharge piping (new), vendor supply of suction and discharge silencers (new), silencer efficiency (new), orifice plates (new), small bore connections (new), reactive-type silencers and material for absorptive-type silencers (new).
It also includes explanations and a new informative Appendix H "Vi and Pressure Ratio Considerations for Rotary Screw Compressors" which deals with over- and under compression which in general leads to higher amplitudes of pulsations and vibrations with higher frequencies that in general more difficult to mitigate. Several additional images on over- and under compression have been provided in Appendix H of this EFRC report to provide a better understanding of the effect on pulsations.

Table 4.1 provides a summary of the most important changes and a summary of the detailed descriptions of the adjustments have been given in Appendix B of this report.

Table 4.1 Summary of adjustments of the 2nd edition of API 688 compared to the 5th edition of API 619

Section nr 5th edition API 619	Section nr 2nd edition API 688	Subject	Description/adjustments of the second edition of the API 688
6.9.1	5.2.1	Decisions to perform an analysis related to power	The following requirement has been added: “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.” Two important Notes regarding have been added, and it is recommended reading these notes carefully.
-	5.2.2 (New)	Agreement between purchaser and vendor on scope of analysis	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.
6.9.1	5.2.3	Purchaser’s responsibility for providing process condition	The requirement regarding the scope of a pulsation and noise analysis has been adjusted to: “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.” Two important notes (NOTE 2 and 4) have been added regarding gas composition and high frequency pulsations, and it is recommended to read these notes carefully.

-	5.2.4 (New)	Number of harmonics to be analysed	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 (New)	Mechanical analysis of silencers	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. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex B of this report.**
-	5.2.6 (New)	Separation margin MNF's and PPF of discharge piping	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. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex B of this report.**
-	5.2.7 (New)	Vendor supply of suction and discharge silencers	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.
6.9.3	5.2.8.1	Allowable pressure drop statement in data sheet	The requirement of section 6.9.3 of the fifth edition of the API 619 regarding the allowable pressure drop stated in the datasheet has now been moved to this section.
6.9.3	5.2.8.2	Allowable pressure drop for low pressures.	The requirement of section 6.9.3 of the fifth edition of the API 619 regarding the allowable pressure drop for low pressure applications has now been moved to this section. Low pressure has now been defined as a suction pressure < 1 barg
6.9.3	5.2.8.3	Varying operating conditions	The remark of paragraph 6.9.3 of the fifth edition of API 619 regarding varying operating conditions has now been moved to this section and the following requirement has been added: "For machines with highly variable operating conditions (e.g., speed, molecular weight of the gas, pressure, etc.), the above limits may not be achievable in all cases. Unless otherwise stated, the above limits apply to the normal operating condition on the data sheets of the API 619 compressor."

-	5.2.10 (New)	Silencer efficiency	The NOTE of section 6.9.6 of the fifth edition of the API 619 regarding the silencer efficiency has been moved to this section. In addition, the following requirement has been added: "In the case of oil-flooded screw compressors, the oil separator should be as close to the discharge flange as practical." **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex B of this report.**
-	5.2.11 (New)	Orifice plates	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. **It is recommended to read the additional explanation(s) on this subject in the applicable API 688 section in Annex B of this report.**
-	5.2.13 (New)	Small bore connections	Small-bore connections (DN 40 or NPS 11/2 in.) and smaller shall be addressed per the discussion in Annex F.
	5.2.14 (New)	Reactive-type silencers	Only reactive-type silencers shall be applied for oil-flooded-screw compressors.
-	5.2.15 (New)	Absorptive-type silencers materials	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.
6.9.5 through 6.9.7 and 6.9.7 through 6.9.18	-	Mechanical design pulsation suppression devices and silencers	These sections contain requirements regarding the mechanical design of the pulsation suppression devices and silencers and are no longer included in section 5.2 on pulsation and vibration control of the second edition of API 688.

5 Review summary of reciprocating and controlled volume pumps (API 674/API 675)

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

In the third edition of the API 674 (Positive Displacement Pumps— Reciprocating) and fourth edition of the API 675 (Positive Displacement Pumps— Controlled Volume for Petroleum, Chemical, and Gas Industry Services), the requirements for the pulsation and vibration control sections are given in respectively informative Annex C of the API 674 and informative Annex F of the API 675. This is in contrast to the API 618 and API 619 where the pulsation and vibration control requirements are part of the “main sections” of the standard and only additional information is given in the Annexes.

The following sections are the only sections on pulsation and vibration control of the “main sections” of the third edition of the API 674 and API 675:

API 674
7.7.1.3 Selection and Scope of Design Analysis Methods—The Purchaser shall specify if design analysis for pulsation and vibration control is required, and, if so, which method to follow (see Annex C). The Purchaser shall also indicate whether existing pumps and their associated piping are to be included in the analysis.

API 675
7.8.1 If specified, the Vendor shall furnish pulsation suppression devices to be located at the pump suction and/or discharge connections. The following are three basic types of pulsation suppression devices:

7.8.2 The Purchaser and the Vendor shall mutually agree on the type and arrangement of pulsation suppression devices to be used for each pumping system.

The pulsation and vibration control sections of the third edition of the API 674 and API 675 are identical and for that reason the API has decided to combine them into one section of the second edition of the API 688. The pulsation and vibration control sections of the API 674 and API 675 have been adjusted, improved, and extended considerably and are now in line with the philosophies and design approaches of the API 618. This makes sense because it involves similar pulsations and vibrations of piston compressor pipe systems. The adjustments and extensions mainly concerns an analysis selection chart (new), alternate operating conditions (new), multiple unit additive effects (new), analysis design selection criteria (new), valve closure delay (new), mechanical analysis (new), inlet pressure vs liquid vapor pressure (adjusted), new design criteria such as allowable piping system acoustic shaking force (new), piping design vibration criteria (new), cyclic stress (new), allowable pressure drop (new), flow measurement error (new) and documentation (new).

It is also common praxis nowadays to carry out a preliminary study of the pulsation dampers for a PD pump identical to that laid down in the API 618. This is done to optimize the damper layout at an early stage of the project without all the data of the pipes being known and definitive. If the damper optimisation were to be carried out at the time when all the necessary piping data is available and completed, the planning of the project could be unnecessarily delayed. This may lead to additional costs because the production time of the pulsation dampers could be considerable. Unfortunately, the API has decided not to include this pre-study in the second edition of the API 688.

The pulse shape of the pulsations of plunger pumps are different from those of piston compressors and are mainly due to the fact that a liquid is much less compressible than a gas. The pulsation dampers of liquid systems are different from those of a gas system often have a gas filling for a more efficient dampening of pulsations. Another relevant aspect of the pulsation control of a liquid system is that cavitation at the suction side may occur caused by pulsations which may lead to fatigue failure of pump parts, opening of relief valves, hammering of check valves, and metering errors.
The mechanical vibrations are identical to those of a gas system because the design and layout of the pipe system and pipe supports are identical to those of a gas system.

Identical to other sections of this report, explanations and remarks have been given in this EFRC report for those sections where it may be relevant.

Because most of the sections are new, it was decided not to summarize the adjustments in a table as has been done for the other machines.

5.3.1. General (New)
This section gives some general information and refers to Annex A (Annex A (informative) Description of Work Processes for Reciprocating Compressors and Plunger Pumps) and Annex I (Annex I (informative) Design Approach Flowchart for PD Pumps).
Annex J must be added to this section because this is an informative Annex on “Cavitation Considerations for PD Pump Systems.”

5.3.3. Alternate Operating Conditions (New)
The requirements (are more or less identical to the requirements defined by the sixth edition of the API 618.

5.3.3. Multiple Unit Additive Effects (New)
The requirements are more or less identical to the requirements defined by the sixth edition of the API 618.

5.3.4 Analysis Design Selection Criteria (New)
This new requirement is a significant improvement compared to the penultimate editions of the API 674 and API 675 and is used to select which analyses steps are required for a specific pump system depending on the maximum allowable pump speed and rated power of the pump. This can be compared to Table 6 (Design Approach Selection) of section 7.11.4 of the sixth edition the API 618. The different analysis steps have also been summarised in the flowchart of informative Annex I (Design Approach Flowchart for PD Pumps).

5.3.5. Acoustic Simulation (C1.3)
5.3.5.2 Acoustic Simulation Model (C1.3)
This was already described in penultimate editions of the API 674 and API 675, but the description has now been improved considerably.

5.3.5.2.1 (New)
A further improvement which is new is the following requirement of “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)”. This gives a better description of the flow pulse related to the valve opening time which is related to the compressibility of the liquid.

5.3.6. Mechanical Review and Mechanical Analysis (C.1.4 API 674)
5.3.6.1 General (C.1.4 API 674)
5.3.6.2 Piping System Mechanical Review (C.1.4 API 674)
This section already existed in penultimate editions and is the pipe support span calculation and is identical to that of the sixth edition of the API 618.

5.3.6.3 Mechanical Analysis (New)

The requirements of this section are completely new and are (more or less) identical to the requirements defined by the sixth edition of the API 618.

- Step 1 —Piping System MNF Analysis of section 5.3.6.3.1 is identical to step 3A of the sixth edition of the API 618 with the exception that this is only applicable for the piping in this standard.
- Step 2—Piping System Forced Mechanical Response of section 5.3.6.3.2 is identical to step 3B2 of the sixth edition of the API 618.

5.3.7 Design Criteria (Partly new)

5.3.7.2 Allowable Pulsation Levels at and Beyond Line-side Connections of Pulsation Suppression Devices (C1.4 API 674)

Unchanged

5.3.7.3 Inlet Pressure vs Liquid Vapor Pressure (C1.6 API 674)

The requirements have been adjusted to:

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:

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

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).

5.3.7.4 Pressure-limiting Valve Protection (C1.7 API 674)

1. The smaller sign (<) has now been adjusted to an equal sign (=) in equation (17) which suggests that the positive peak P_p must always meet the value of equation (17) which should not be the case because smaller values are also correct. It is assumed that this is a typo.
2. The relief valve setting P_{rv} is indicated as *gauge* pressure expressed in kPa (psia). Gauge is not correct in this case because it concerns an absolute pressure in kPa.

5.3.7.5 Allowable Piping System Acoustic Shaking Force (New)

The requirements are identical to the requirements defined by the sixth edition of the API 618.

5.3.7.6 Piping Design Vibration Criteria (New)

The requirements are identical to the requirements defined by the sixth edition of the API 618.

5.3.7.7 Maximum Allowable Cyclic Stress (New)
The requirements are identical to the requirements defined by the sixth edition of the API 618.

5.3.8 Other criteria
5.3.8.1 Allowable Pressure Drop (New)
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.

The conversion from 15 kPa to 2 psi is not correct and must be 2.18 psi.

5.3.8.2 Separation Margins (C1.8 API 674)
This criterion has been adjusted to the requirement below and are identical to those of the sixth edition of the API 618

a) the minimum MNF of any pump 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 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 (New)
For flow meters located in the piping system, the maximum flow measurement error caused by flow pulsations shall not exceed the following and is identical to that of the sixth edition of the API 618.

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 (New)
This new section describes which information shall be included in the pulsation and vibration study report and is identical to that of the sixth edition of the API 618.

6 Review summary of PD rotary pumps (API 676)

6.1 5.4 Pulsation and Vibration Control Techniques for PD Pumps—Rotary

The pulsation and vibration control section 5.4 of the second edition of the API 688 has been compared to section 7.8 of the fourth edition of the API 676 " Pulsation and Vibration Control Requirements for Multiphase Skids”.
The fourth edition of the API 676 has only one requirement in section 7.8.1: “The interaction of the dynamic flow generated by the multiphase pump (MPP) with acoustical resonance in piping systems can result in excitation in the pump and piping. This energy can result in pump and piping failures. Refer to API RP 688 for guidance on pulsation and vibration”.
Generally, PD pumps have a near-constant volumetric flow, and pulsations do not normally occur or are negligible for 100% liquid operation, assuming that sufficient NPIP (Net Positive Inlet Pressure) is provided to the pump inlet to prevent cavitation. There is a greater potential for vibration issues when the process conditions are a combination of high gas void fraction (GVF), high differential pressure, and/or high viscosity of the liquid phase. For those cases, further discussions with the supplier may be warranted to review the possibility for pulsation and vibration control.
Previous editions of the API 676 did not pay much attention to pulsation and vibration control of the piping, and one of the reasons is that the source of pulsations is difficult and inefficient to determine. In the second edition of the API 688 some more attention has been paid to pulsation and vibration control in section 5.4.3 (Analysis approach).

5.4.1 General (New)

5.4.3.1 (General) and 5.4.3.2 (Compressible fluids)
These new sections give only some general information when pulsations may become important and do not include any requirement.

5.4.3 Analysis approach (New)

5.4.3.1 General (New)
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.

1. In general, the pulsation excitation sources are not known, and it is noted that cavitation does not only depend on the temperature, but also on other process parameters such as for example the local pressures in the pipe system and the pump housing.
2. The analytical approach to be considered and listed in 5.4.3.2 is the mechanical pipeline assessment that requires the frequencies of the dominant pulsation forces. However, the standard does not specify how to determine the anticipated vibrations and refers to only the bearing housing vibration levels that exceed the API 676 limits as defined in section 6.11 and it suggests that the frequencies of measured vibrations are required. These can be obtained during a factory acceptance test (FAT) or when the system is installed at the plant. It is noted that the vibrations measured on the pump and piping during a FAT may differ significantly from those from the field because the foundation structure and pipe system of the test rig differ from the actual field situation in many cases.
3. The analytical approach to be considered and listed in 5.4.3.3 is the calculation of the shaking forces of the pipe system. These forces are difficult to measure in the field and

require for that reason a detailed pulsation analysis. As explained earlier, this is a major challenge because in many cases the source of excitation is unknown and therefore difficult to determine. This means that this requirement is difficult to fulfil.

The general conclusion is that it is challenging and not efficient to fulfill both requirements of section 5.4.3.2 and 5.4.3.3.

5.4.3.2 Mechanical Piping Review (New)
The requirements are identical to the requirements defined by the sixth edition of the API 618.

5.4.3.3 Allowable Shaking Forces (New)
The requirements are identical to the requirements defined by the sixth edition of the API 618.

7 Annexes of the second edition of the API 688

The Annexes have been compared to the Annexes of the 1st edition of the API 688.

7.1.1 Annex A (informative) Description of Work Processes for Reciprocating Compressors and Plunger Pumps (3)

1. The content of most sections is unchanged.
2. This Annex is identical section 3 of the first edition of the API RP 688 and was part of the main section.
3. Several parts are difficult to understand which is also caused by the fact that only USC units have been used in many equations, especially section A.2.6.6 through A.2.6.12.
4. A few sections are also obsolete such as section A.4.3.2 (Electrical Circuit Analog), section A.4.3.5 (Spectral or Frequency Domain Methods) which are only interesting from a historical point of view.
5. Section 3.2.5 (Non-Resonant Shaking Force Limits for Piping Systems) is no longer included.

7.1.2 Annex B (normative) Stepless Capacity Control for Reciprocating Compressor Cylinders (New)

This an Annex with general information about stepless capacity control for reciprocating compressor cylinders. This capacity control is becoming more popular because it is a very efficient control method compared to other methods.
This capacity control requires more attention regarding pulsations and vibration because the amplitudes and frequencies may become higher especially at the suction side. For those reasons, the task force group has developed a new Annex.

7.1.3 Annex C (informative) Design Approach Flowchart

1. Header title “Design Approach” must be “Design Approach 2”.
2. The flowchart has now been adjusted from normative to informative.
3. The pulsation criteria at the line side nozzle shall not exceed the limits of section 5.1.7.3 (Maximum Allowable Pressure Pulsation at Compressor Cylinder Valves) as shown in the two purple boxes in the picture below. Section number 5.1.7.3 is incorrect and must be 5.1.7.4 (Allowable Pulsation Levels at and Beyond Line-side Connections of Pulsation Suppression Devices).

   When a pre-study is performed according to section 5.1.5.4, the maximum allowable pressure pulsation level at the pulsation suppression device line-side nozzle flange shall be 70 % of the allowable value defined in section 5.1.7.4, the pressure-drop requirements in section 5.1.8.1, and the shaking force requirements of the *pulsation suppression device* in section 5.1.7.5.3 shall apply. The pressure-drop requirements and the shaking force requirements of the *pulsation suppression device* are not indicated in the flowchart, see purple box on the right side of Figure C.1

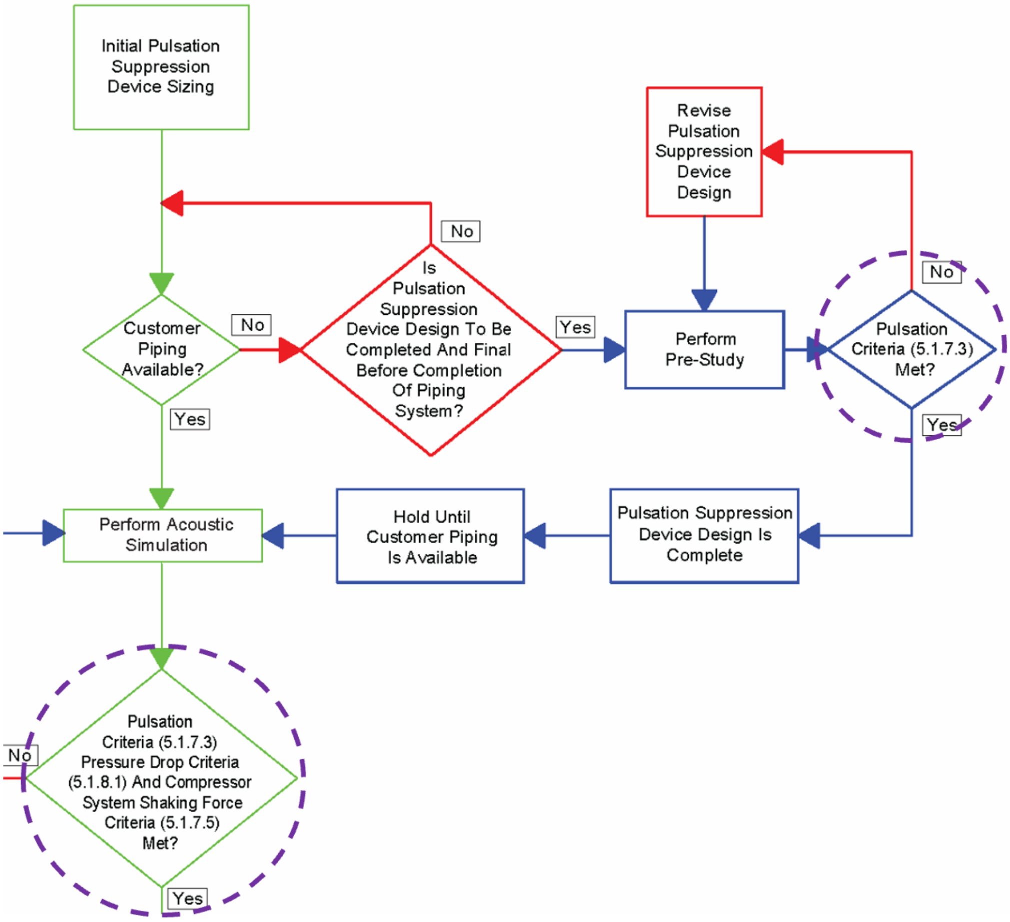


Figure C.1 Part of API 618 flowchart DA2

7.1.4 Annex D (informative) Design Approach 3 Flowchart

1. The flowchart has now been adjusted from normative to informative which is remarkable because the API 688 has been adjusted from a Recommended Practice to a Standard.
2. The same as discussed in 3. for flowchart C, it also applies to flowchart D as shown in Figure D.1

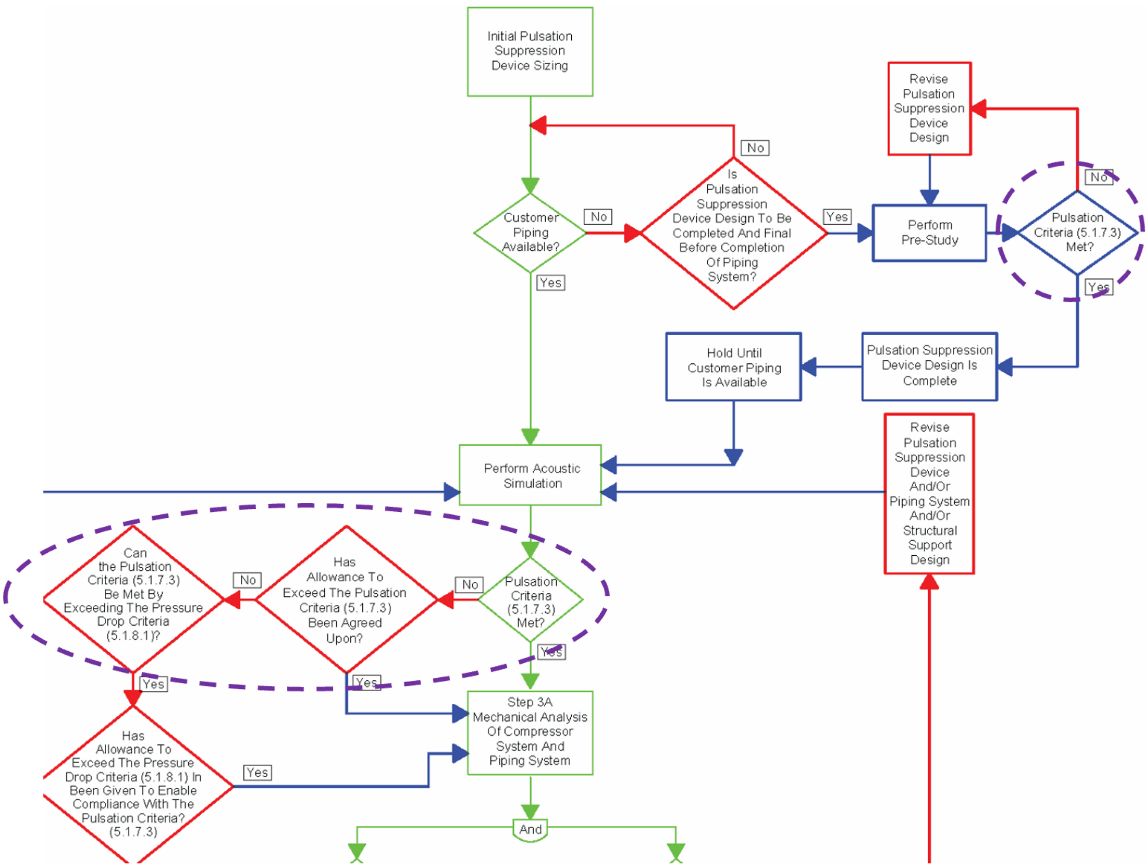


Figure D.1 Part of API 618 flowchart DA3

7.1.5 *Annex E (informative) Mechanical Forced Response Analysis (New)*

This is a very comprehensive and easy to understand new Annex describing the details of the important items in the steps of a mechanical response analysis of the piping and compressor manifold of the DA3 analysis.

E.3 General Mechanical Forced Response Analysis

"Due to uncertainties in the prediction of mechanical natural frequencies including fabrication tolerances, the vibration response should be predicted assuming mechanical resonance when the predicted MNF is within ±10 % of the frequency of the excitation source. As shown in Figure E.1, the resonant response should be utilized when the 10 % separation margin is not met".

It strongly depends on the system how large the margin shall be, but a margin of ±20 % is required in many internation standards e.g. ISO and API. The separation margin criterion as defined in section 5.1.8.2 of this standard also defines a margin of ±20% and it is recommended to use the same margin. The margin shall even be larger for higher frequencies because experience has shown that pipe support clamps may become in resonance resulting in deviations of MNF's, which is often observed for screw compressor systems.

E.4 Dynamic Analysis for Step 3b1, Compressor System

E.4.1 "The forced response study should generally consider frequencies up to at least 6× compressor running speed. Forces to be included in this analysis are gas rod loads (acting on cylinders causing "stretch"), compressor system pulsation-induced shaking forces, and vertical crosshead forces from gas and inertia rod load."

The 6x compressor speed is sometimes not large enough and this value can be determined by studying the frequency of the dominant pulsation forces. This can also be concluded from picture Figure E.3 which are also showing high frequency vibrations.

7.1.6 Annex F (informative) Small-bore Piping Design and Analysis (New)

This is an especially important new Annex describing key details which are required to avoid vibration and fatigue failure problems of small-bore connections. These are often experienced in systems with a pulsating flow and dynamic mechanical loads. Although the vibration of the mainline pipe or vessel to which the small-bore connections are mounted may be acceptable, fatigue failure can occur.

The definition of a small-bore connection as given in this Annex of the API 688 is:
“A small-bore connection is defined as a branched connection on mainline piping, vessels, or equipment that is NPS 2 in. (DN 50) and smaller. This includes connections that have a branch pipe to mainline pipe ratio (“branch ratio”) of less than 10 % and excluding connections that have a branch ratio greater than 25%. These are shown in figure F.5 of the API 688 standard below.

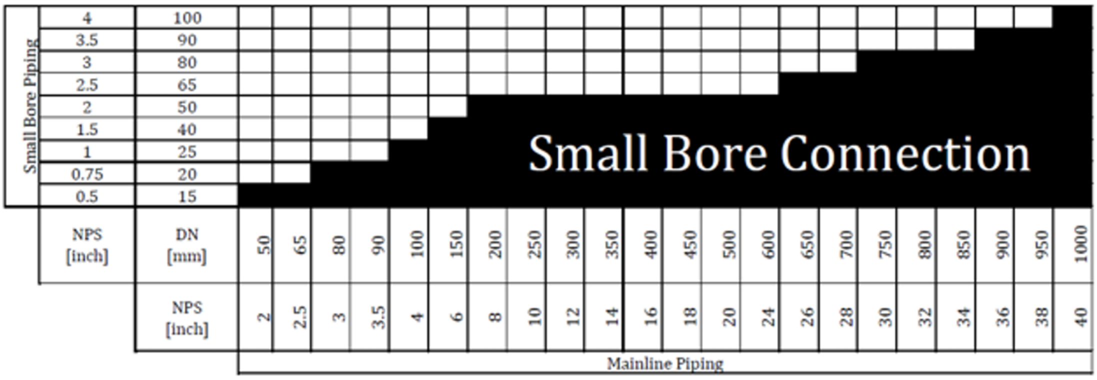


Figure F.5—Scope of Small-bore Piping

Figure F.1 Table F.5 from the standard

The ISO 20816-8 has a different definition which is as follows:
“A small-bore connection is a branch connection on mainline piping, vessels or equipment that has an actual outer diameter of 60,3 mm or smaller (DN 50, NPS 2), or that has an actual outer diameter larger than 60,3 mm with a branch ratio (3.8) of less than or equal to 12%”. The branch ratio is defined as the ratio of the small-bore connection actual outer diameter to mainline piping actual outer diameter.

Note 1: All connections that have a branch ratio greater than 36 % are excluded.
Note 2: The small-bore connection piping extends until the effect of the mainline piping vibration is negligible, which is typically the first support.

Small-bore connections according to ISO 20816-8 are shown in Table E.2 of the ISO 20816-8 and it can be concluded that the number of small-bore connections according to the API 688 definition is 111 and 116 according to the ISO 20816-8.

The general conclusion is that ISO 20816-8 contains more small-bore connections for the larger main pipe diameters.

Table E.1 — Diameters of small bore connection

				Mainline piping																							
	Outer diameter, mm			60,3	73,0	88,9	102	114	168	219	273	324	356	406	457	508	610	660	711	762	813	864	914	965	1 016
		NPS, inch		2	2,5	3	3,5	4	6	8	10	12	14	16	18	20	24	26	28	30	32	34	36	38	40
			DN, mm	50	65	80	90	100	150	200	250	300	350	400	450	500	600	650	700	750	800	850	900	950	1 000
Small bore piping	21,3	0,5	15																						
	26,7	0,75	20																						
	33,4	1	25																						
	48,3	1,5	40																						
	60,3	2	50																						
	73,0	2,5	65																						
	88,9	3	80																						
	102	3,5	90																						
	114	4	100																						

NOTE 1 Grey shading indicates small bore connection.

NOTE 2 NPS is the nominal pipe size (see e.g. ASME B31.3), DN is the nominal diameter (see e.g. ISO 6708).

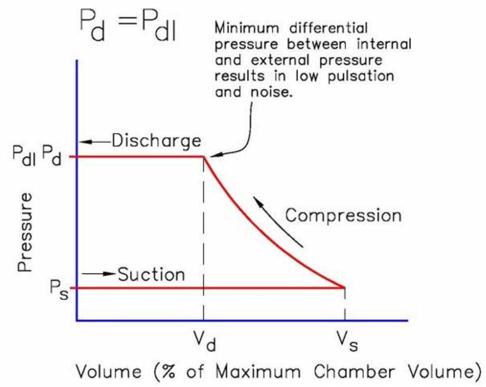
Figure F.2 Table E.1 of the ISO 29816-8

7.1.7 *Annex G (informative) Pulsation Considerations for Flow Metering Equipment*

Unchanged

7.1.8 *Annex H (informative) Vi and Pressure Ratio Considerations for Rotary Screw Compressors (New)*

Pictures H.1 through H.3 give a better understanding on the shapes of the PV chart with the effect on lost power for systems with over- and under compression. A frequency spectrum of the discharge flow pulse in over- and under- compression will show the higher frequency components. Note that at the suction side, the process gas is taken in at a constant suction pressure. Therefore, at the suction side, under- or over compression will not occur. Generally, the pulsations at the suction side are less severe than at the discharge side. Note however, that also on the suction side, the flow will be unsteady (pulsating).



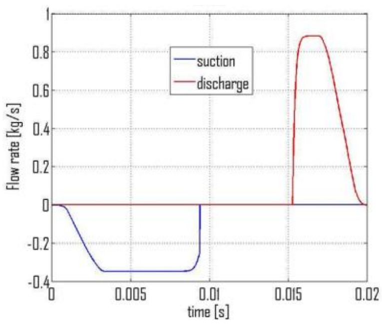


Figure H.1. Example of a PV chart in the left picture with the suction and discharge flow pulses (for one pocket) in ideal case (internal pressure ratio = external pressure ratio) in the right picture.

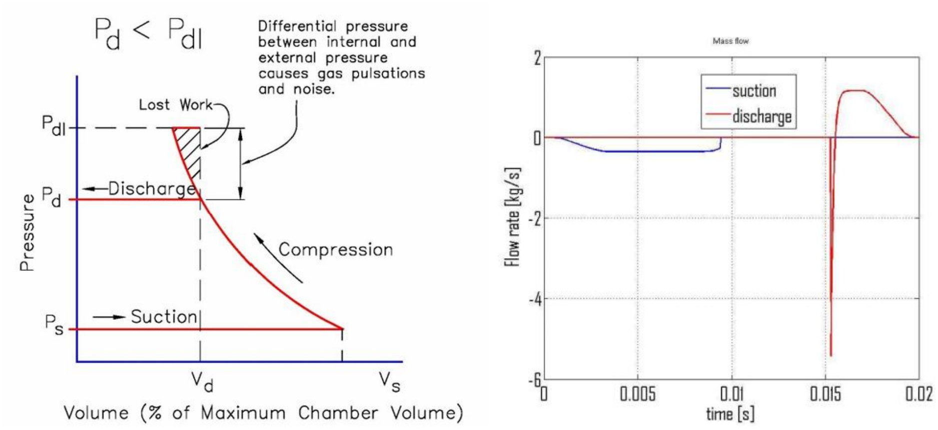


Figure H.2. Example of a PV chart in the left picture and suction and discharge flow pulses (for one pocket) in under-compression case (internal pressure ratio < external pressure ratio) in the right picture. A negative (backflow) pulse is observed in the discharge flow which causes the high amplitude pulsation with a high frequency.

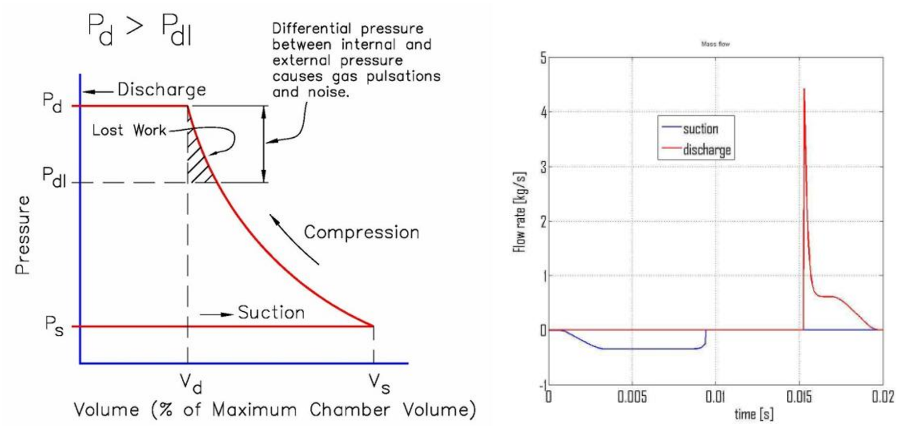


Figure H.3. Example of a PV chart in the left picture and suction and discharge flow pulses (for one pocket) in over-compression case (internal pressure ratio > external pressure ratio) in the right picture. A positive pulse is observed in the discharge flow which causes the high amplitude pulsation with a high frequency.

7.1.9 *Annex I (informative) Design Approach Flowchart for PD Pumps (New)*

1. The purple encircled part of the flowcharts as shown in Figure I.1 does not make sense because it is now an infinitely loop because there is no action required in the backwards loop.

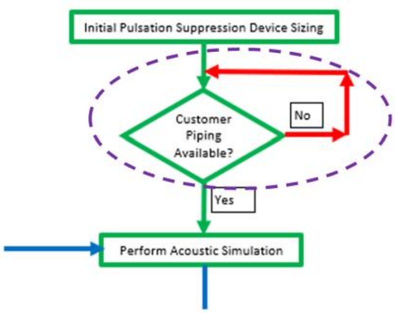


Figure I.1 Part of the flowchart

2. According to the requirements of section 5.3.5.2 for a pulsation study, the design criteria shall be fulfilled for the pulsation level at the line connection as specified in 5.3.7.2, the inlet pressure vs liquid vapor pressure as defined in 5.3.7.3, the pressure-limiting valve protection in 5.3.7.4, the pressure drop as specified in 5.3.8.1. However, the shaking forces criterion shall not be evaluated in the pulsation study which means that the flowchart is not correct on this point and the shaking forces as shown in the purple encircled part Figur I.2 shall be removed.

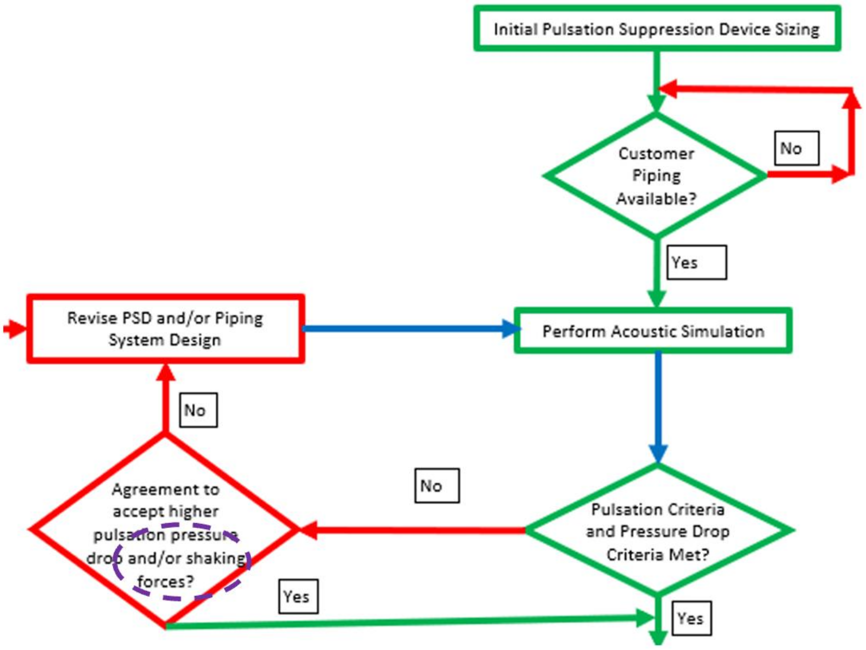


Figure I.2 Part of the flowchart

3. If in a pipe span calculation the separation margin criterion is not fulfilled, an additional pipe support or stiffening of the pipe support structure is the most effective method to change the mechanical natural frequencies. No costly modification of the PSD or pipe system is ever carried out and for that reason the flowchart as shown in Figure I.3 is not correct and there shall not be a back loop to the pulsation analysis.

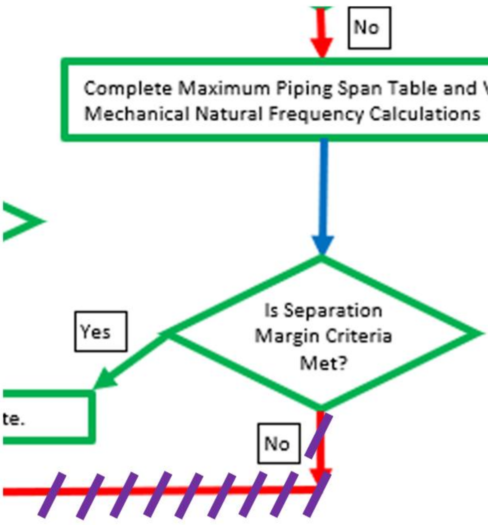


Figure I.3 Part of the flowchart

4. If during the mechanical response analysis, the vibration or cyclic stress level exceeds the limits, the normal situation is that the layout of the pipe supports (additional supports, (parts of) pipe support reinforcement, etc.) will be adjusted. In those cases, only the mechanical analysis will have to be performed again, and a pulsation study is not necessary. This is in contrast to the flowchart which directs the analysist back to the pulsation study. In rare situations, it may be necessary to change the pipe thickness, pipe routing or a redesign of the pulsation damper, but at that stage of the project the costs are likely to be unacceptably high.

7.1.10 Annex J (informative) Cavitation Considerations for PD Pump Systems (New)

This is a completely rewritten section on cavitation which replaces Annex E (Net Positive Suction Head Versus Net Positive Inlet Pressure) of the third edition of the API 674 and section 3.1.3.5 (Pump Cavitation) of the first edition of the API RP 688.
It gives a better understanding of cavitation and the term acceleration head which has now been replaced by pulsations cause by the acceleration of the liquid.

7.2 Bibliography (New)

See API 688 standard for an overview.

8 References

The references given in part 1 of this EFRC report also apply for part 2 of this EFRC report.

[1] G. Alberts, St. Belfroid, "*Effect of Pulsations on Separator Efficiency",* TNO Science & Industry, EFRC Conference 2007, Antwerp Belgium

[2] EFRC Guidelines *"How to avoid liquid problems in reciprocating compressor systems", v*ersion 3, August 2018

[3] M.C.A.M. Peters, *"Damping of low frequency noise in piping systems by means of perforated plates",* ImechE conference 2003.

[4] D.W.Bechert, *"Sound absorption caused by vorticity shedding demonstrated with a jet",* Journal of Sound and Vibration, 70(3), 1980.

[5] M.C.A.M.Peters et al, *"Quasisteady aero acoustic response of orifices",* Journal of the Acoustical Society of America, 110 (4), 2001.

[6] Leonard van Lier and Harry Korst Department, "Mitigation of high-frequency pulsations, using Multi Bore Restriction Orifices", TNO Delft, The Netherlands, EFRC Conference, 2007 Prague, Czech Republic.

9 Annexes with detailed descriptions of the adjustments of API 688 section 5.1 and 5.2

9.1 Annex A: Summary of detailed descriptions of the adjustments of section 5.1 (Reciprocating Compressors (Ref. API 618))

5.1.1. General (7.11.1)
This is a completely new description of section 7.11.1 of the sixth edition: “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 (7.11.1.2) Recommended pulsation and vibration control techniques
Unchanged

5.1.1.2 (• 7.11.1.3) Preliminary sizing, pulsation suppression devices
Unchanged, with the remark that this section is no longer preceded by a bullet.

See also API 618 section 7.11.1.3 in part 1 of this EFRC report for an explanation.

5.1.1.3 (7.11.1.4) Exceeding of acceptance criteria
Unchanged

5.1.1.4 (• 7.11.1.5) Analysis to be specified by purchaser
Unchanged, with the remark that this section is no longer preceded by a bullet.

5.1.2 Alternate Operating Conditions (• 7.11.2)
Unchanged, with the remark that this section is no longer preceded by a bullet.

See also API 618 section 7.11.2.1 and 7.11.2.2in part 1 of this report for an explanation.

5.1.2.1 (7.11.2.1)
The loading steps for stepless capacity control have now been added and the number of steps for this unloading device has been specified as follows: 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 (7.11.2.1)
This section contains now the last requirement of section 7.11.2.1 of the sixth edition of the API 618: “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 (7.11.2.2) Two or more gases with dissimilar molecular weights
Unchanged

5.1.3 Multiple Unit Additive Effects (7.11.3)
5.1.3.1 (• 7.11.3.2) Multiple compressors in parallel
Unchanged, with the remark that this section is no longer preceded by a bullet.

5.1.3.2 (• 7.11.3.3)
Unchanged, with the remark that this section is no longer preceded by a bullet.

5.1.4 Design Approach Selection (7.11.4)
Unchanged

See also API 618 section 7.11.4 in part 1 of this report for an explanation.

5.1.5 Design Approach 2—Acoustic Simulation and Piping System Mechanical Review (7.11.5)
5.1.5.1 General (7.11.5.1)
The description of some requirements has been adjusted with preservation of content.

5.1.5.2 Compressor System Acoustical Model (7.11.5.2)
The following requirement has been added: "When stepless capacity control is used, the frequency range shall be sufficient to cover the higher harmonics as generated by the stepless control system."

5.1.5.3 Piping System Model (7.11.5.3)
The following requirement has been added: "In some cases, a downstream volume may be introduced to terminate the scope of the acoustic model. Refer to Annex A for a discussion of the extents of the acoustic model."

Identical to section 7.11.5.3 (Piping System Model) of the API 618 as summarised in part 1 of this EFRC report, the reference of this section to the design limits of section 5.1.7.4 may be confusing because section 5.1.7.4 consists of the following sections:

1) Section 5.1.7.4.1 and 5.7.4.2 Allowable Compressor Cylinder Flange Pressure Pulsation.
2) Section 5.1.7.4.3 Refers to limits defined by 5.1.7.4.1 or 5.1.7.4.2, the allowable shaking force (5.1.7.5, includes 5.1.7.5.1, 5.1.7.5.2 and 5.1.7.5.3), or vibration design criteria (5.1.7.6), or cyclic stress levels (5.1.7.7).
3) Section 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.

Especially, the reference to the allowable shaking force section 5.1.7.5.2 (Allowable Piping System Acoustic Shaking Force), section 5.1.7.6 (Vibration design criteria), section 5.1.7.7. (Cyclic stress levels) may be confusing because for the DA2 analysis, the following design limits shall apply:

- Section 5.1.7.4 Allowable Compressor Cylinder Flange Pressure Pulsation.
- Section 5.1.7.5.3 Allowable Cylinder Mounted Pulsation Suppression Device Shaking Force.
- Section 5.1.8.1 Allowable Pressure Drop.
- Section 5.1.8.2 Separation Margins.
- Section 5.1.8.3 Flow Measurement Error.

5.1.5.4 Pre-study (7.11.5.4)
Unchanged

See also the explanations as given in section 7.11.5.4. of the sixth edition of the API 618 in part 1 of the report. The API 688 sections number given in the remarks are as follows for the second edition of the API 688:

- Pulsation levels at line side nozzle: 5.1.7.4
- Pressure-drop requirements in section: 5.1.8.1
- Shaking force requirements *of the pulsation suppression device* in section: 5.1.7.5
- Pulsation levels at cylinder flange: 5.1.7.2
- Pulsation levels at cylinder valves: 5.1.7.3

5.1.5.5 Mechanical Review and Piping Restraint Analysis (7.11.5.5)
Unchanged

5.1.6 Design Approach 3—Acoustic Simulation and Piping Restraint Analysis Plus Mechanical Analysis (7.11.6)
See also the explanations as given in API 618 section 7.11.6 in part 1 of this report.

5.1.6.1 General (7.11.6.1)
5.1.6.1 (7.11.6.1.1)
Unchanged

5.1.6.1.1 (7.11.6.1.2) Stresses in pulsation suppression devices
The following requirement has been added: "… that result from pulsation-induced shaking forces and pressure-induced static force."

5.1.6.1.1 (New)
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 (7.11.6.2)
See also the explanations as given in API 618 section 7.11.6.2 in part 1 of this report.

5.1.6.2.1 (7.11.6.2)
The following explanations have been added to the standard:
"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 influenced by the assumed boundary conditions. When properly applied, the two methods should give comparable 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."

It has been indicated that the accuracy of either method is influenced by the assumed boundary conditions which is fully correct. However, the calculation of the MNF's of both the piping and the compressor system model can hardly be carried out by a FAM analysis because the geometry of the compressor cylinder, distance piece, crosshead guide, frame, pulsation dampers including the details of nozzle connections, flexible pipe supports etc. (see section A.3.2.3. and A 3.3. through A.3.8 of the second edition of the API 688), cannot be modelled accurately by using a FAM. This means that the FEM approach is more in favour than the FAM approach because it is more accurate and faster.

5.1.6.2.2 (7.11.6.2)
The requirement of this section is the requirement of the second paragraph of section 7.11.6.2 of the sixth edition of the API 618.

5.1.6.3 Step 3b1—Forced Mechanical Response Analysis of the Compressor System (7.11.6.3)
5.1.6.3.1 (7.11.6.3.1)
It has been indicated that 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. This is not correct because the allowable vibration level may be exceeded but, in that case, a cyclic stress calculation shall be conducted.

The piping in this context consists of the piping *and* the pulsation damper that are attached to the compressor cylinder.
See also API 618 section 7.11.6.3 in part 1 of this report for additional explanations.

5.1.6.3.2 (7.11.6.3.2)
The reference to the European Forum for Reciprocating Compressors (EFRC) vibration guidelines is no longer included and this is replaced by the following requirement: “The allowable vibration limits for the compressor cylinders and frame shall be supplied by the compressor OEM.”

5.1.6.4 Step 3b2—Forced Mechanical Response of the Piping System Model (5.11.6.4)

5.1.6.4.1 (7.11.6.4) Exceeding separation or shaking force criteria
Unchanged

It has been indicated that the piping system model shall include all of the piping that was included in the acoustic simulation. This may not be sufficient in many cases because the mechanical boundary may differ from the acoustic boundary. An adequate mechanical boundary is e.g. the second stiff support upstream and downstream of an acoustical boundary condition e.g. a closed valve.

5.1.6.4.2 (7.11.6.4)
This section contains the last requirement of section 7.11.6.4 of the second edition of the API 688.

5.1.7 Design Criteria (7.11.7)
5.1.7.1 General (7.11.7.1)
Unchanged

5.1.7.2 Maximum Allowable Pressure Pulsation at Compressor Cylinder Flanges (7.11.7.2)
Unchanged

5.1.7.3 Maximum Allowable Pressure Pulsation at Compressor Cylinder Valves (New)
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 (7.11.7.3)
Unchanged

5.1.7.4.1 (7.11.7.3.1) Allowable level for pressures > 3.5 bara
Unchanged

5.1.7.4.2 (7.11.7.3.1) Allowable level for pressures < 3.5 bara
Compared to the equation in the sixth edition of the API 618 this equation is correct

See also API 618 section 7.11.6.3.1 in part 1 of this report.

5.1.7.4.3 (• 7.11.7.3.2) Exceeding design criteria limits
Unchanged

5.1.7.4.4 (7.11.7.3.3) Flow pulsations in elements sensitive to such phenomena
Unchanged

5.1.7.5 Allowable Acoustic Shaking Force (7.11.7.4)
Unchanged

5.1.7.5.1 General (7.11.7.4.1)
Unchanged

5.1.7.5.2 Allowable Piping System Acoustic Shaking Force (7.11.7.4.2)
Compared to the sixth edition of the API 618, a further distinction has been made between ground and rack supported systems and potentially resonant (separation margin smaller (SM) than 20%) and potentially non-resonant (separation margin smaller than 20%). A distinction has now been made between a pipe diameter indicated in NPS and in DN. It is common to identify pipes by inches using NPS (Nominal Pipe Size). The metric equivalent is DN (Diametre Nominel). The metric designations conform to International Standards Organization ISO (ISO 6708) usage and apply to all plumbing, natural gas, heating oil, and miscellaneous piping used in buildings. An overview of the NPS and DN diameters is shown in Table 3.2.

Table 3.2 Overview of pipe diameters in NPS and DN

ISO 6708 - Pipework components - DN (Nominal Pipe Size NPS)

Nominal Pipe Size - NPS		**Outside Diameter *(mm)***			
DN (mm)	**Inch**	**ISO 6708 Pipework components**	**DIN EN 10220 Seamless Steel Pipes**	**DIN EN 10255 Threaded Tube**	**ASME**
10	3/8	17.2			
15	1/2	21.3	20.0	21.3	21.3
20	3/4	26.9	25.0	26.9	26.7
25	1	33.7	30.0	33.7	33.4
32	1 1/4	42.4	38.0	42.4	42.2
40	1 1/2	48.3	44.5	48.3	48.3
50	2	60.3	57.0	60.3	60.3
-	2 1/2	-	-	73.0	73.0
65	-	76.1	76.1	76.1	-
80	3	88.9	88.9	88.9	88.9
-	3 1/2	-	-	101.6	101.6
100	4	114.3	108	114.3	114.3
125	-	139.7	133	139.7	-
-	5	-	-	141.3	141.3
150	6	168.3	159	168.3	168.3
200	8	219.1	216	219.1	219.1
250	10	273.0	267	273.0	273.0
300	12	323.9	318	323.9	323.8
350	14	355.6	368	355.6	355.6
400	16	406.4	419	406.4	406.4
450	18	457	470	457	457
500	20	508	521	508	508
600	24	610	622	610	610
700	28	711	720	711	711
800	32	813	820	813	813
900	36	914	920	914	914
1000	40	1016	1020	1016	1016
1200	48	1220		1219	1219

5.1.7.5.3 Allowable Cylinder Mounted Pulsation Suppression Device Shaking Force (7.11.7.4.3)

The value of SF_{dmax} has been adjusted from 45,000 (N) of equation (12) of the sixth edition of the API 618 to equation (8) below:

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

- SF_{dmax} is the maximum pulsation suppression device non-resonant peak-to-peak force guideline in N (lbf).
- MACCRL is the maximum allowable continuous combined rod load defined in API 618 in N (lbf).
- Number of Cylinders is number of cylinders to which the pulsation suppression device is attached.

The allowable shaking force is now a function of the maximum allowable continuous combined rod load defined in API 618 and the thought behind this is that the vibration amplitude of the compressor manifold system (cylinder, distance piece, crosshead, frame and pulsation dampers) is a function of the strength of the compressor frame, which is a logical thought.

Most dampers installed horizontally on the compressor cylinders have their main axis perpendicular to the piston rod, which means in general that the vibrations will also be perpendicular to the piston rod. Generally, the distance piece and the crosshead are less stiff in the direction perpendicular to the piston rod axis, and the question now is whether this new criterion does not lead to excessively high allowable shaking forces of the Mounted Pulsation Suppression Device compared to those of the sixth pressure of the AP 618.

5.1.7.5.4 Gas Rod Load Induced Forces (New)
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 (7.11.7.5)
All values of vibration has been given in peak-to-peak in this standard and the user shall be aware of the fact that in b) the value is given in 18 mm/s 0-p which may be overlooked.

5.1.7.7 Allowable Cyclic Stress (7.11.7.6)

5.1.7.7.1 (7.11.7.6.1)
The allowable peak-to-peak cyclic stress has now been reduced from 180 N/mm^2 to 96 N/mm^2, considering all stress concentration factors present and with all other stresses within applicable code limits.

This is a reduction factor of about two and is due to the availability of better fatigue limit curves for infinite lifespan.
Considering all stress concentration factors present means that the stress concentration factors (SCF's) of welds and the stress intensification factors of nozzle vessel connection (SIF's) are present in the mechanical (FE) model.

5.1.7.7.2 (• 7.11.7.6.2)
This section is no longer preceded by a bullet.
The specified piping code providing the design criteria**,** has now been added (e.g. ASME B31.3, ASME B31.8, etc.).

5.1.8 Other Criteria (7.11.8)

5.1.8.1. Allowable Pressure Drop (7.11.8.1)
Unchanged

5.1.8.2 Separation Margins (7.11.8.2)
Unchanged

5.1.8.3 Flow Measurement Error (7.11.8.3)
Unchanged

5.1.9 Documentation Requirements
Unchanged

5.1.10 Summary of Design Steps (New)

In the fourth edition of API 618, the various analysis steps for the 3 design approaches were defined from M.1 to M.11. With these M-options, it was quite clear for the buyer of the pulsation and vibration study which M-items needed to be ordered for a specific design approach. For unknown reasons, the M-options were no longer included in the fifth edition because the standard was expanded with more calculation methods, such as shake force and vibration control, and it was not so simple to define them all with the M-definitions. Due to the ease of use in the past, it was decided to include them again now. However, the introduction of the M-items may cause some confusion, as explained below, especially for the DA2 analysis.

Design approach 2:
According to the API 688 section 5.15, the DA2 consists of the following analysis steps:

- Acoustic pre-study of the dampers (API 688 section 5.1.5.4).
- Pulsation analysis of the piping (API 688 section 5.1.5.1 through 5.1.5.3).
- Pipe support span and vessel/damper MNF calculation (section 5.1.5.5).

The following design criteria shall be fulfilled:

- Pulsation levels at compressor flange (API 688 section 5.1.7.3)
- Pulsation levels in the piping (API 688 section 5.1.7.4)
- Shaking forces on pulsation dampers only (API 688 section 5.1.7.5.3)
- Separation margin for piping and vessels (API 688 section 5.1.8.2)
- Pressure drop (API 688 section 5.1.8.1).

According to API 688 section 5.1.10 the following analysis steps are required for a DA2:

- M.1 Basic design of pulsation suppression devices using proprietary and/or empirical techniques (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.

The M option requirements for the DA 2 analyses are conflicting to the requirements of API 688 section 5.1.10 because of the following reasons:

- M.1 is defined as a part of DA2 for powers < 75 kW which is not defined in any section of the API 688 nor in Table 1 of the API 688.
- M.1 is not a part of DA2 because it requires a *simulation* of the pulsations which is not according to the definition of M.1 and besides that option M.1 is the formerly used DA1 of the fifth edition of the API 618 which is no longer defined in the second edition of the API 688.
- According to the definition of M.2, the shaking forces need to be calculated but it is not explicitly defined that this is only required for the pulsation suppression devices and not for the piping which might be confusing.

Design approach 3

- What is missing in the M options for the calculation of the shaking forces for the pipe system and pulsation suppression devices.
- The steps according DA3 as defined in M.2-M.5 are indicated as the “default” design approach which is not defined in any other section of the API 688. This is conflicting with the requirements as defined in API 688 section 5.6.1 and the DA3 flowchart of Annex D and might be confusing.
- If the user still follows this “default“ design approach, it is the DA2 with the addition of M5 which is the mechanical analysis of the compressor system to determine mechanical natural frequencies (FEM techniques) to maintain adequate separation margins.
  The analyses steps which are not included in the “default” design compared to the “full” DA3 design as defined in section 5.6.1 of the API 688 are the forced response analysis of the piping and the compressor system (M.6. and M.7).

In part 1 of this EFRC report (see API section 7.11) the different design philosophies and design approaches have been explained and one of the disadvantages of the acoustic design approach as defined in M.2-M.5 in the second edition of the API 688, is that it might result in large heavy pulsation dampers or acoustic filters.

Another disadvantage is that this approach may lead to (many) unnecessary additional pipe supports and pipe support structures achieving enough stiffness which is even more challenging for variable speed compressors. This can be avoided by performing a "full" DA3 analysis according to API 688 section 5.6.1 of the API 688, including the final step, the mechanical response analysis of the piping and compressor system.
In general, this last step is very easy and efficient to perform because the mechanical natural frequencies have already been calculated in step M.4 and M.5 and nowadays many pulsation simulation programs have a coupling with the (FEM) mechanical analysis programs. This ultimately leads to an optimal layout of the pipe supports compared to the other methods.

9.2 Annex B: Summary of detailed descriptions of the adjustments of section 5.2 (Rotary Type PD Compressors (Ref. API 619)

5.2.1 (6.9.1)
The following requirement has been added: “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, 2 and 3 of the fifth edition of the API 618 have been replaced by the following two notes:

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 (New)
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 (6.9.1)
The requirement regarding the scope of a pulsation and noise analysis has been adjusted to: “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.”

The following notes have now been added to the existing notes 1 and 3 of the fifth edition of the API 619:

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

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 (New)
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 (New)
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.

> It is noted that the calculation of the mechanical natural frequencies (MNF's) applies not only to the wall of the silencer but also to the internal components. These are especially important when it comes to an oil separator for oil-flooded screw compressors. Furthermore, it is not always possible to avoid the coincidence of the MNF's with the pulsation excitation frequencies, which is especially the case for variable load compressors. For those cases, it is recommended to perform a vibration and cyclic stress calculation of the silencers to judge if fatigue failure will occur.

5.2.6 (New)
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.

> The 10% margin may be too small for the higher frequencies of these compressors, and it is recommended using a minimum of ±20%.

5.2.7 (New)
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 (6.9.3)
Unchanged

5.2.8.1 (6.9.3)
The requirement of section 6.9.3 of the fifth edition of the API 619 regarding the allowable pressure drop stated in the datasheet has now been moved to this section.

5.2.8.2 (6.9.3)
The requirement of section 6.9.3 of the fifth edition of the API 619 regarding the allowable pressure drop for low pressure applications has now been moved to this section. Low pressure has now been defined as a suction pressure < 1 barg.

5.2.8.3 (6.9.3)
The remark from paragraph 6.9.3 of the fifth edition of API 619 regarding varying operating conditions has now been moved to this section and the following requirement has been added "For machines with highly variable operating conditions (e.g., speed, molecular weight of the gas, pressure, etc.), the above limits may not be achievable in all cases. Unless otherwise stated, the above limits apply to the normal operating condition on the data sheets of the API 619 compressor."

5.8.2.4 (6.9.4)
Unchanged

Figure 2 has been added to make it clear that the 2% peak-to-peak pulsation level applies for pressures of < ≈3000 kPa and equation (11) and (12) applies for pressures > ≈3000 kPa.

5.2.9 (6.9.5)
Unchanged

5.2.10 (New)
The NOTE of section 6.9.6 of the fifth edition of the API 619 regarding the silencer efficiency has now been moved to this section. The following requirement has now been added: “In the case of oil-flooded screw compressors, the oil separator should be as close to the discharge flange as practical.”

It is known that the efficiency of separators decreases when pulsations are present at the inlet. The measured efficiency drop can even reach up to 60% of the yield without a pulsating flow. Maximizing efficiency can be achieved by reducing pulsations and through adequate separator design. The EFRC has developed a guideline "How to Prevent Liquid Problems in Piston Compressor Systems" and paragraph 4.2 provides technical guidelines for the preselection of separators and demisters on how to reduce pulsations among other things.

5.2.11 (New)
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.

Single Bore Restriction Orifices (SBROs) are often used to reduce low-frequency pulsations (f< 100 Hz), and reliable (quasi-stable) models are available to predict performance [2] [3] [4]. However, at higher frequencies, as generated in rotary-type compressors, SBROs are less effective as shown in Figure 5.2.1 because they only generate additional pressure losses without adding significant damping. It is therefore recommended to apply Multi Bore Restriction Orifices (MBRO’s) [8] for frequencies > 100 Hz, of which an example is shown in Figure 5.2.2

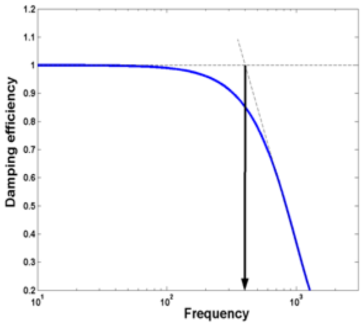


Figure 5.2.1 Effectiveness of a single bore restriction orifice plate (source: TNO)

Figure 5.2.2 Example of a MBRO

5.2.12 (•6.9.8)
Unchanged, with the remark that this section is no longer preceded by a bullet.

5.2.13 (New)
Small-bore connections (DN 40 or NPS 11/2 in.) and smaller shall be addressed per the discussion in Annex F**.**

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

5.2.15 (New)
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.

Sections 6.9.5 through 6.9.7 and section 6.9.7 through 6.9.18 of the fifth edition of the API 619
These sections contain requirements regarding the mechanical design of the pulsation suppression devices and silencers and are no longer included in section 5.2. on pulsation and vibration control of the second edition of API 688.