Training session
Pulsation & Vibration Control

June 26th - 27th 2019
Delft, The Netherlands

EFRC
EUROPEAN FORUM for RECIPROCATING COMPRESSORS

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TNO innovation for life

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DYNAFLOW RESEARCH GROUP.

Pulsations

- Fundamentals
- Analysis
- Pulsation Control
- API 618

Range of possible problems due to excessive pulsations

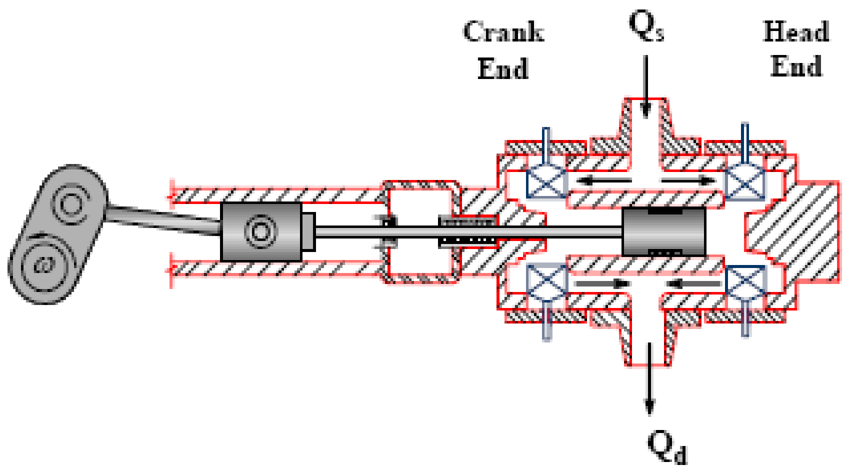
Pulsations are the pressure fluctuations caused by reciprocating equipment which travel through the piping system

Problems due to (excessive) pressure pulsations:

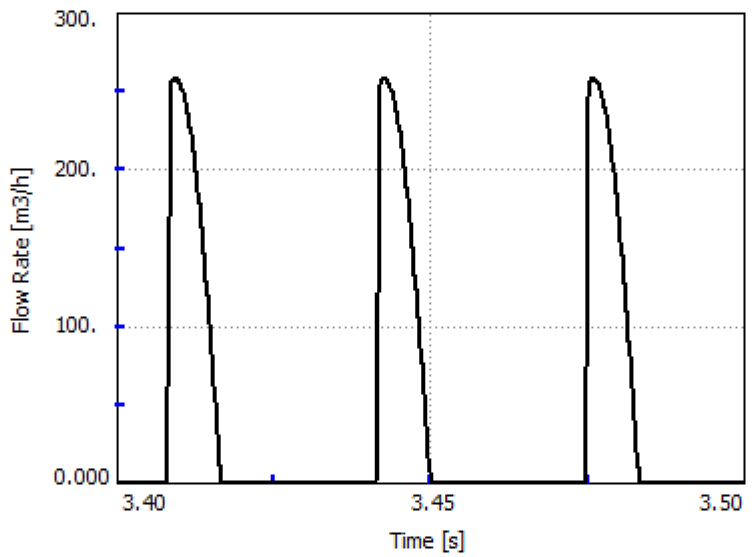
- Shaking forces causing vibration and fatigue failure
- Pipe rupture due to high pressure (extreme case)
- Reduced service life of reciprocating valves and reduced compressor performance
- Issues flow-meters and check valves
- Noise

Reciprocating Compressors generate oscillating flow

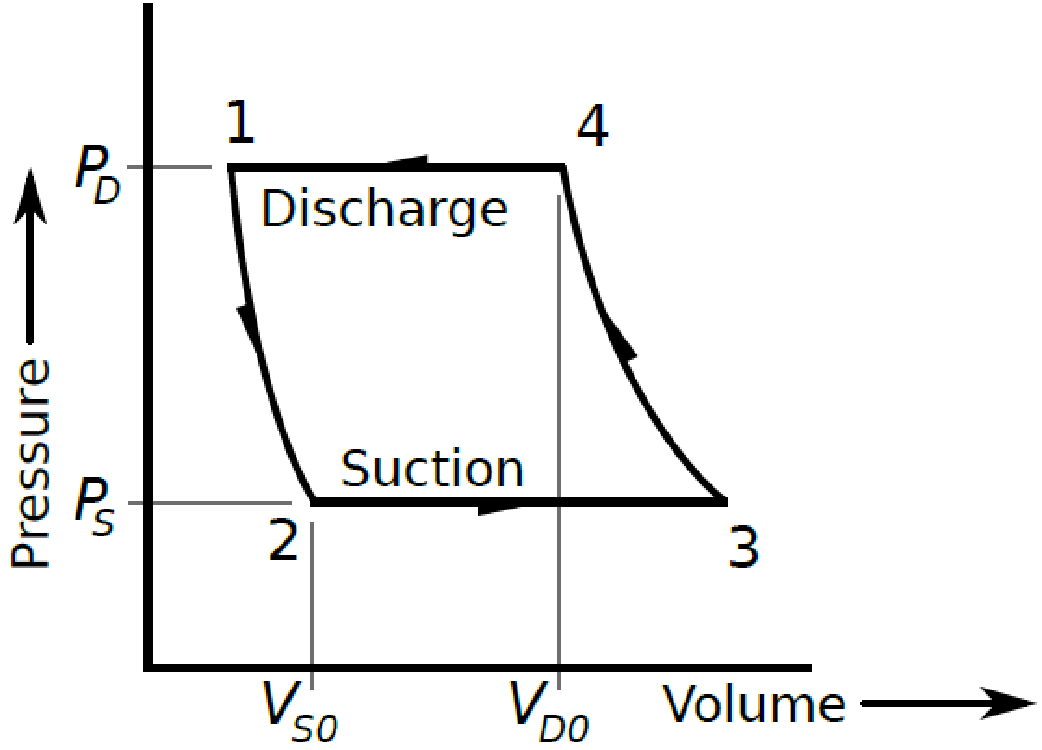
Both at suction and discharge



Periodically changing flow



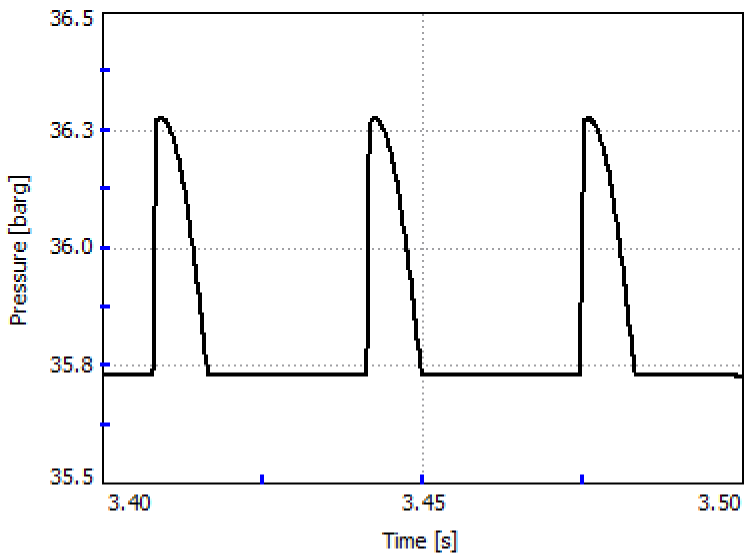
Pressure/Volume (PV) diagram of a piston stroke



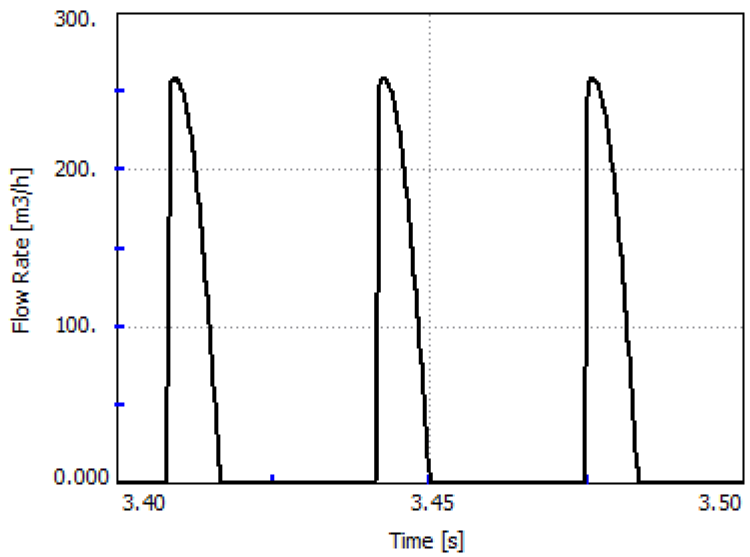
- 1-2 Suction Stroke – Suction valve closed
- 2-3 Suction Stroke – Suction valve open
- 3-4 Discharge Stroke – Discharge valve closed
- 4-1 Discharge Stroke – Discharge valve open

Oscillating flow proportional with oscillating pressure

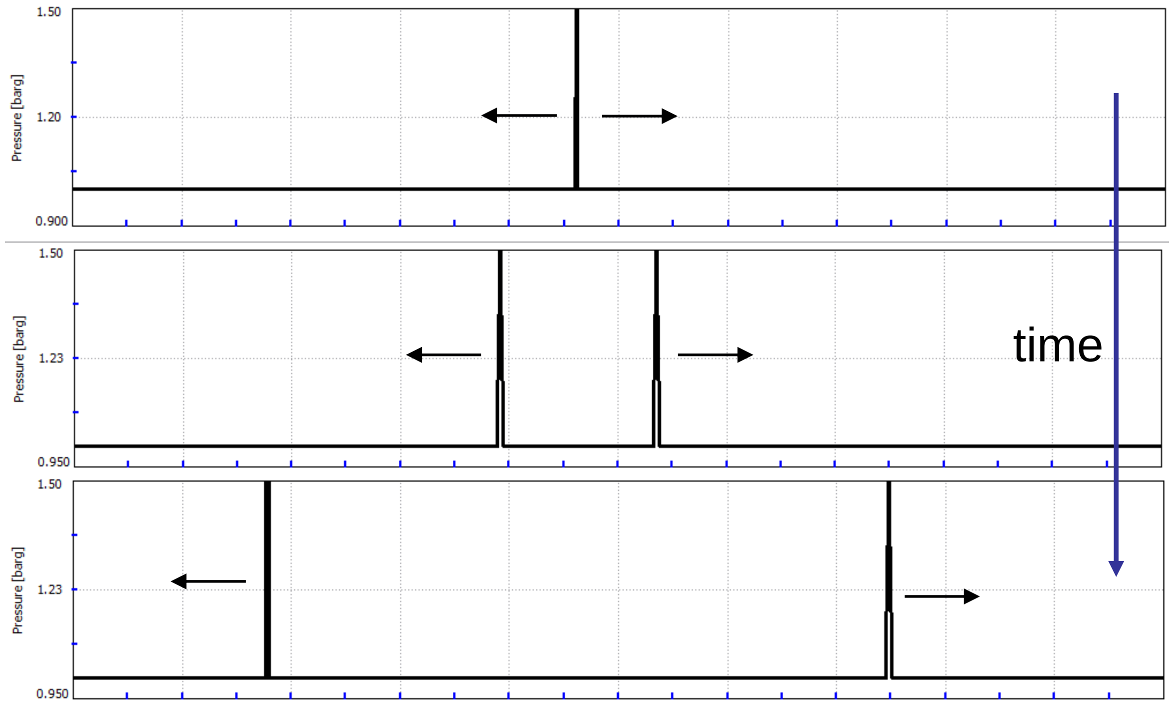
Periodically changing pressure



Periodically changing flow



Pulsations propagate according to wave equation



$$\frac{\partial^2 p}{\partial t^2} - c^2 \frac{\partial^2 p}{\partial x^2} = 0$$

Pulsations are acoustic waves

For reciprocating compressors pulsations are 1D (plane wave through the pipe).

Plane wave frequency limit: $f < 0.59 * c/D_i$

Screw compressors typically produce both plane waves and higher order (3D) waves

Turbo compressors typically produce 3D waves only

Propagations of pulsations happen at the speed of sound inside the pipe

Pulsations can travel upstream and downstream in the flow

Travel with the speed of sound

- Increases with temperature
- Reduces for higher molar weights
- Typical between 300 – 1000 m/s
- Pure hydrogen ~1500 m/s

$$wave\ speed = a = \sqrt{\frac{\frac{c_p}{c_v}p}{\rho}} = \sqrt{\frac{\frac{c_p}{c_v}RT}{M}}$$

c_p/c_v is ratio of specific heats

p is line pressure

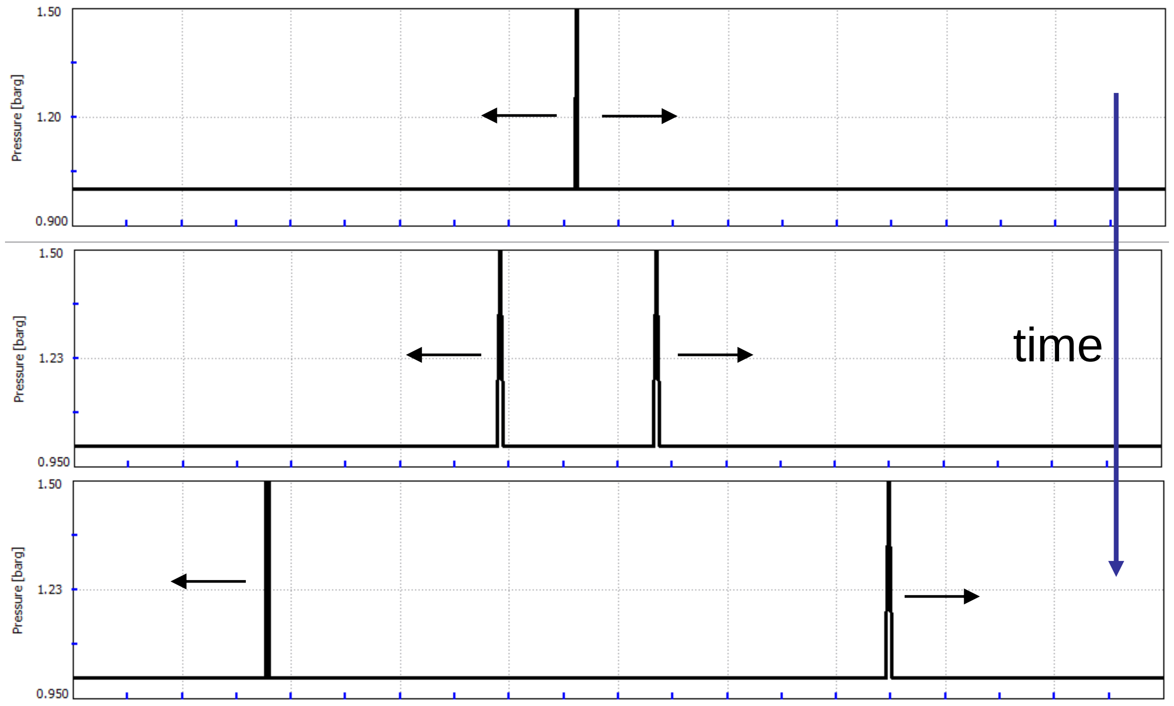
ρ is gas density

R is gas constant

T is the gas temperature

M is the molar mass of the gas

Pulsations propagate according to wave equation



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Pulsations are acoustic waves

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Screw compressors typically produce both plane waves and higher order (3D) waves

Turbo compressors typically produce 3D waves only

Damping of pulsations inside system typically very small

Damping of pulsations in pipe occurs due to frictional losses:

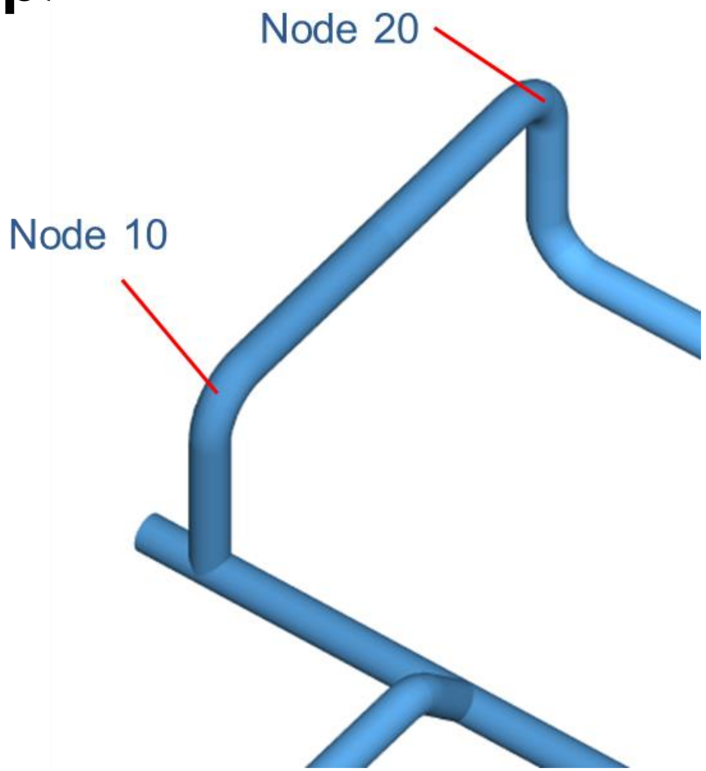
- Turbulence
- Wall friction
- Pressure drop over equipment or valves
- Viscosity of the fluid

At large distances (order larger than wave length in system, think several hundred meters).

Therefore, for typical systems, damping in system is small

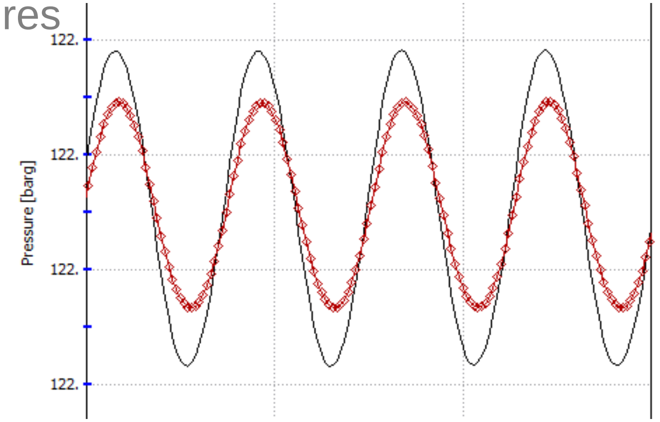
Oscillating force cause by temporary pressure difference in straight pipe segment

Elbow pair

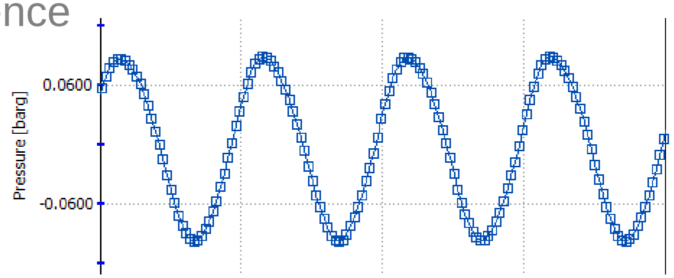


Pressure difference between elbows

Nodal pressures

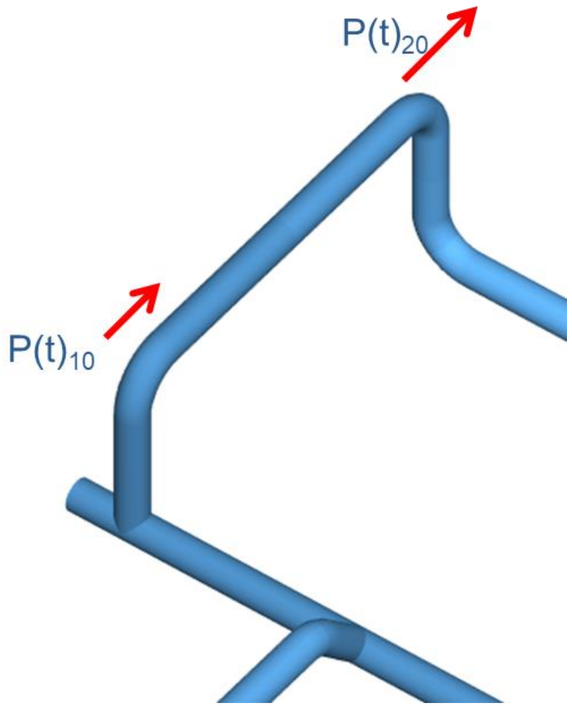


Pressure difference



The pressure difference leads to an oscillating unbalanced force

Net Pressure difference



Pressure difference between elbows

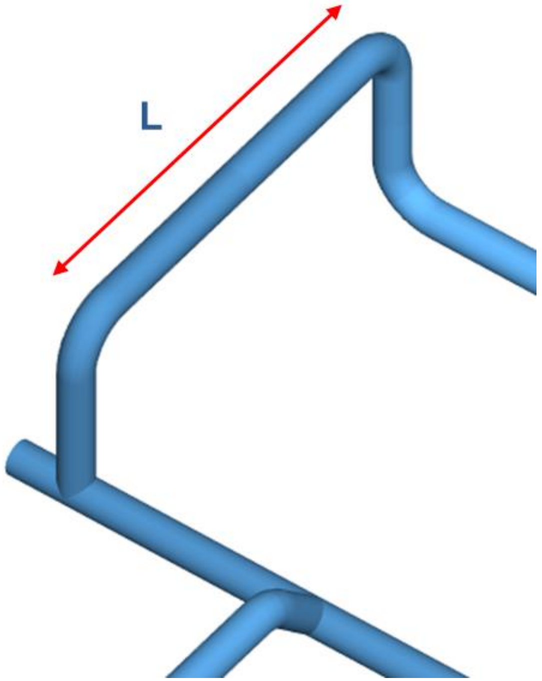
Oscillating unbalanced force, F(t), arises between elbow pairs.

$$F(t) = (P(t)_{10} - P(t)_{20})A_{pipe}$$

- $F(t)$ is the force
- $P(t)_{10}$ is the pressure at node 10
- A_{pipe} is the internal cross section of the pipe

Unbalanced force magnitude depends pulsation magnitude and phase shift

Distance between elbow pair



Phase shift related to distance

The phase shift φ depends on L and the wave length λ:

$$\varphi = \frac{2\pi L}{\lambda} = \frac{2\pi L f}{c}$$

Unbalanced force amplitude equals

$$F = A_{pipe}\,\Delta_{pp}\,\sin\left(\frac{1}{2}\,\varphi\right)$$

With Δ_{pp} the peak-to-peak pressure pulsation.

Pulsations

- Fundamentals
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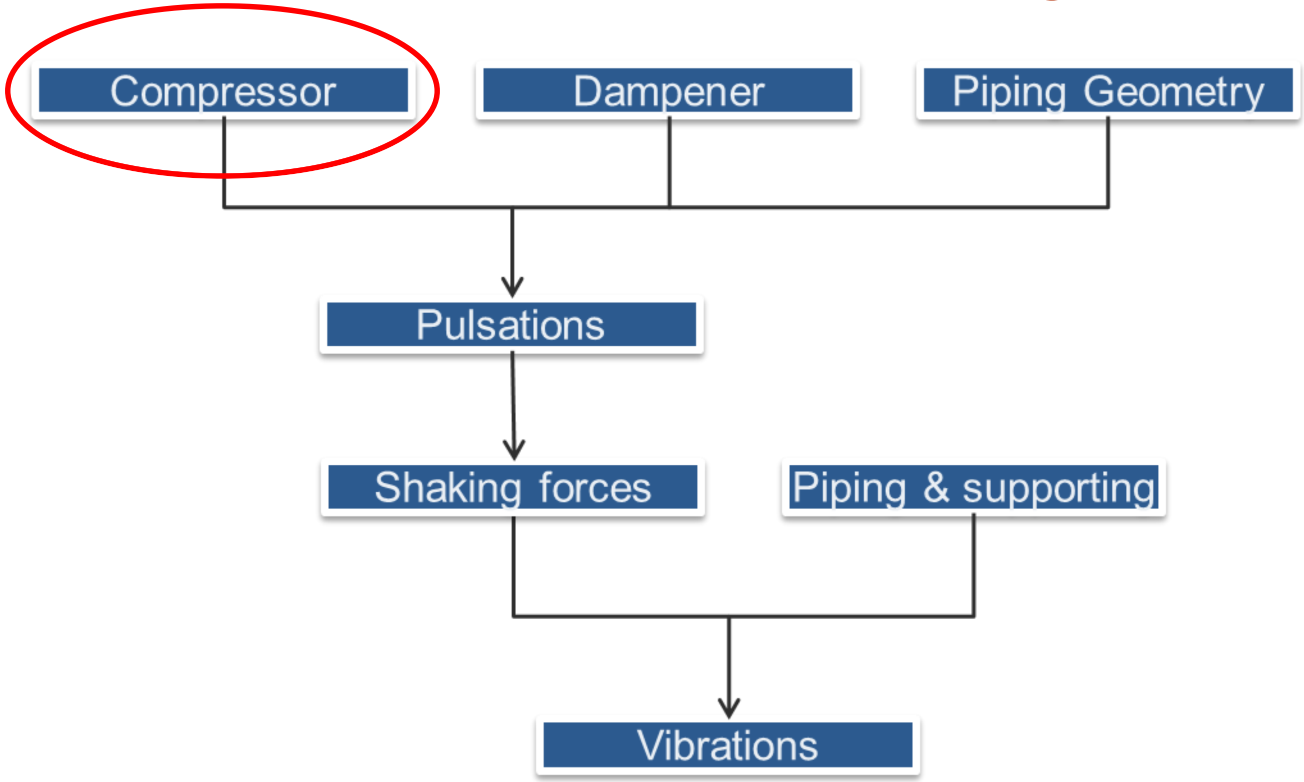
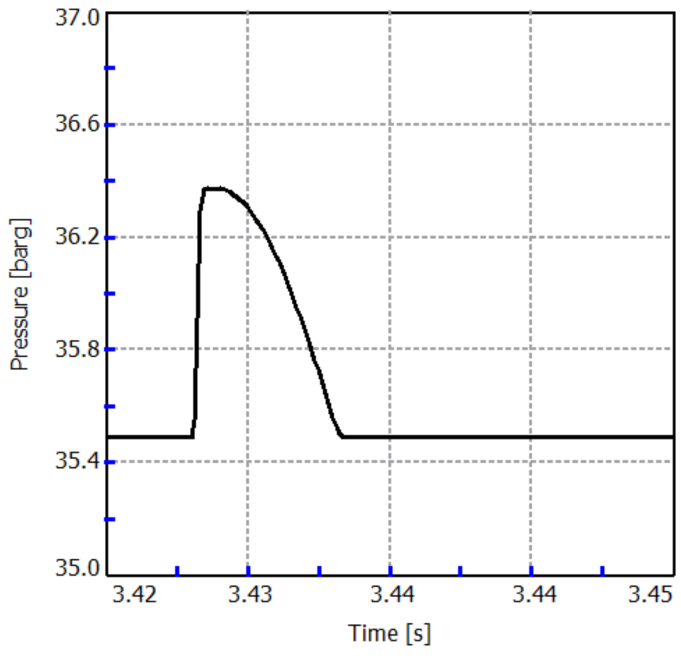
- API 618

Schematic road map for pulsations and vibration understanding

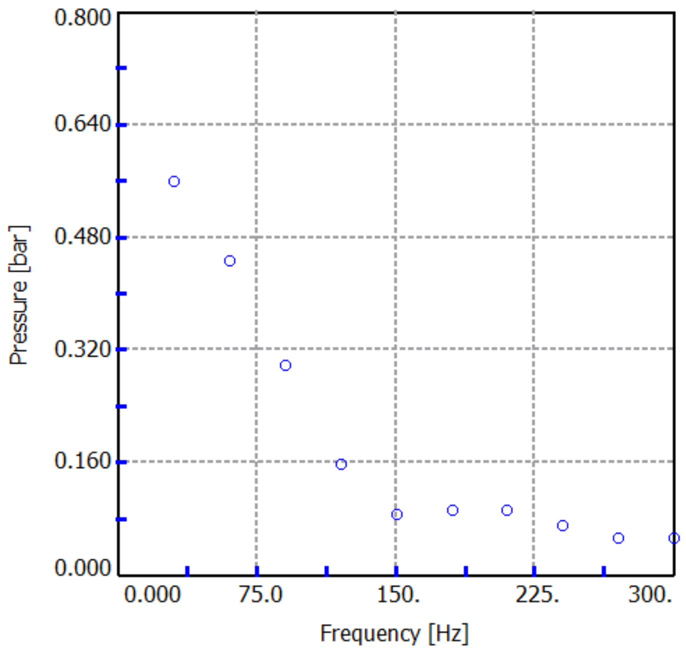


Periodic excitation represented in the frequency domain

Pressure profile head end cylinder

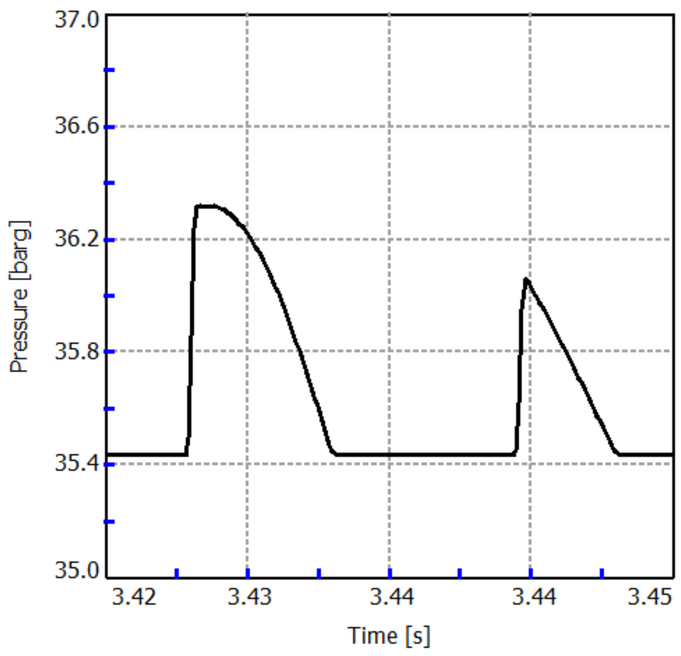


Same signal in frequency domain

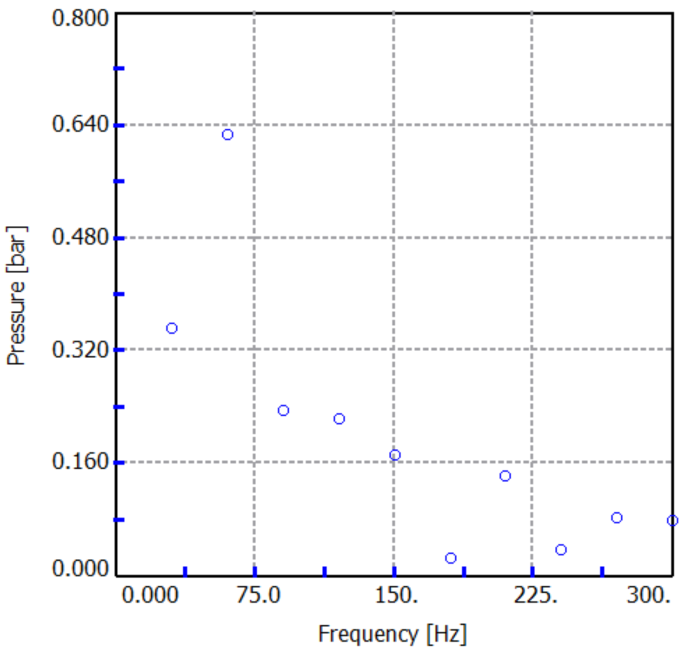


Excitation depends on compressor characteristics

Double acting

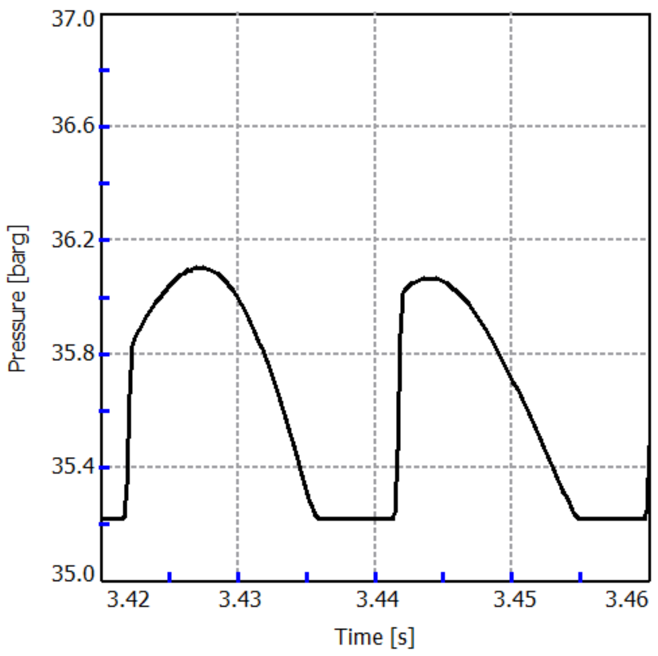


Same signal in frequency domain

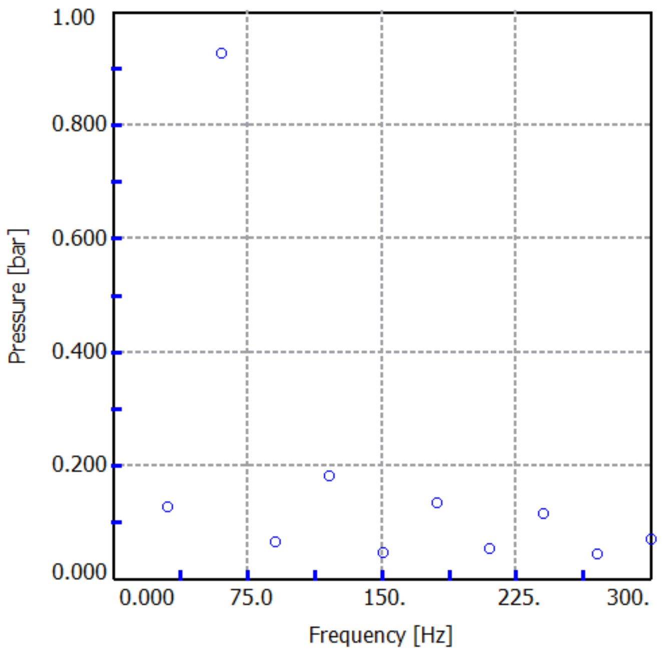


Excitation depends on compressor characteristics

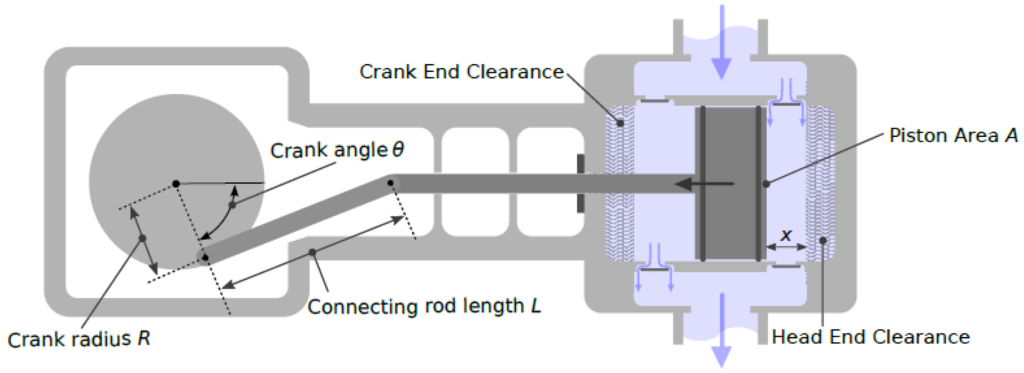
Lower pressure ratio



Same signal in frequency domain



Other aspects that influence excitation



- Clearance volume
- Piston area
- Crank and connecting rod
- Running speed
- Gas properties
- Mode of operation

Pulsations

- Fundamentals
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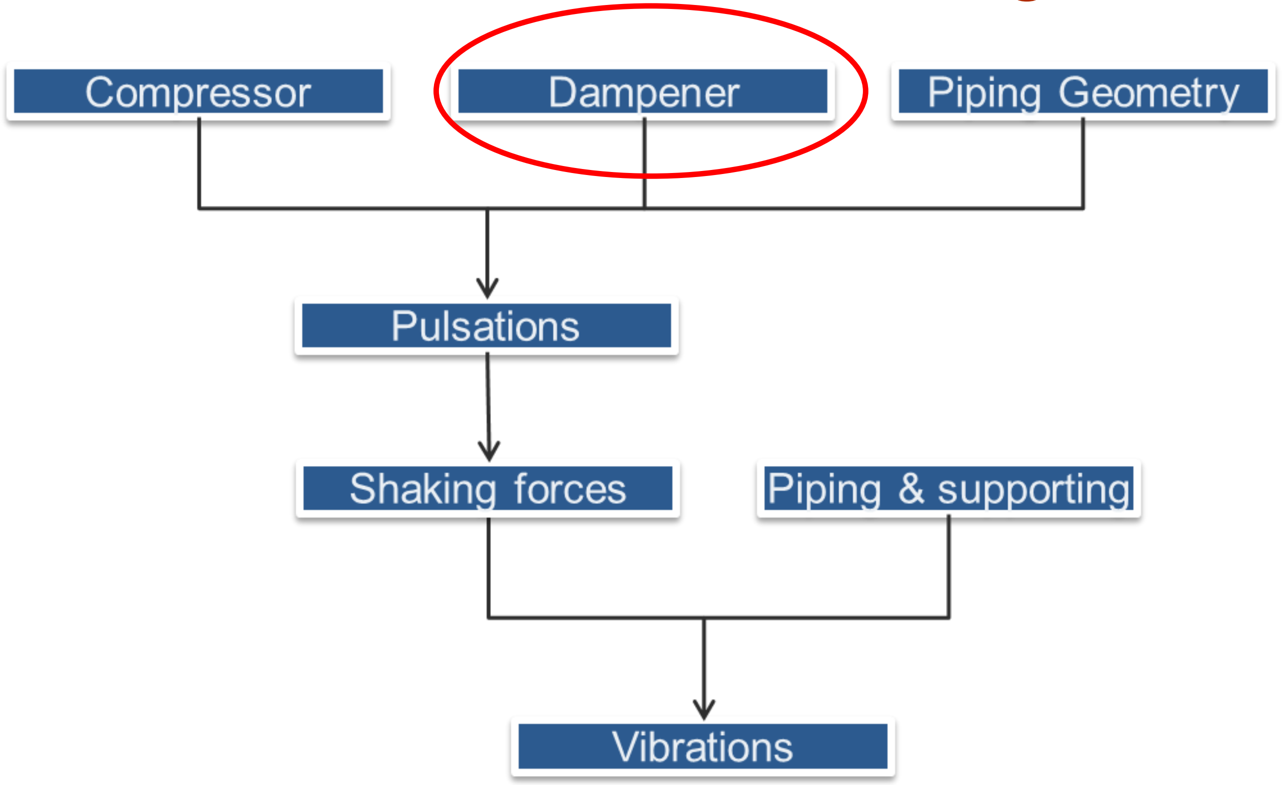
- API 618

Schematic road map for pulsations and vibration understanding



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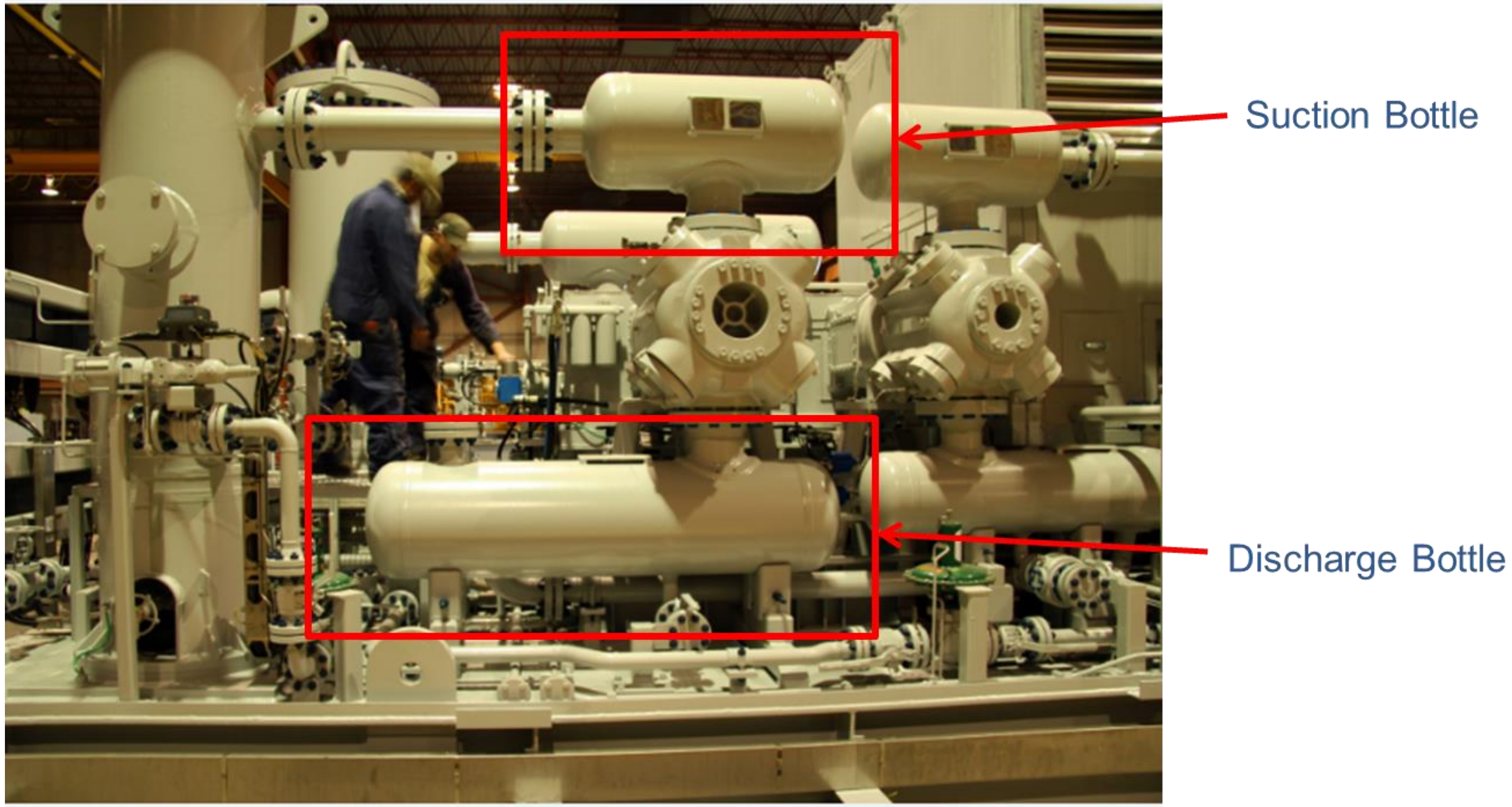
Pulsation Bottles are typically placed on the suction and discharge lines of reciprocating compressors



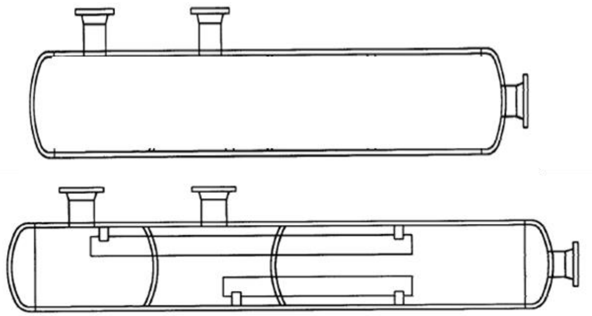
Transmission loss can be used to describe effectiveness of PSD vs frequency

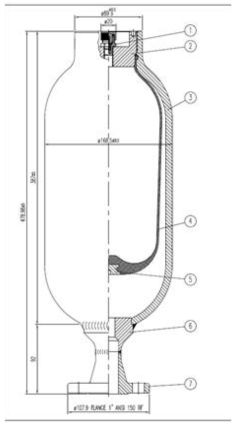
$$TL(f) = 20log_{10}\left(\frac{P_{inc}}{P_{trans}}\right) = 10log_{10}\left(\frac{P_{inc}^2}{P_{trans}^2}\right)$$

P_{inc} is the incident pressure fluctuation (compressor)

P_{trans} is the the transmitted pressure fluctuation (connecting piping

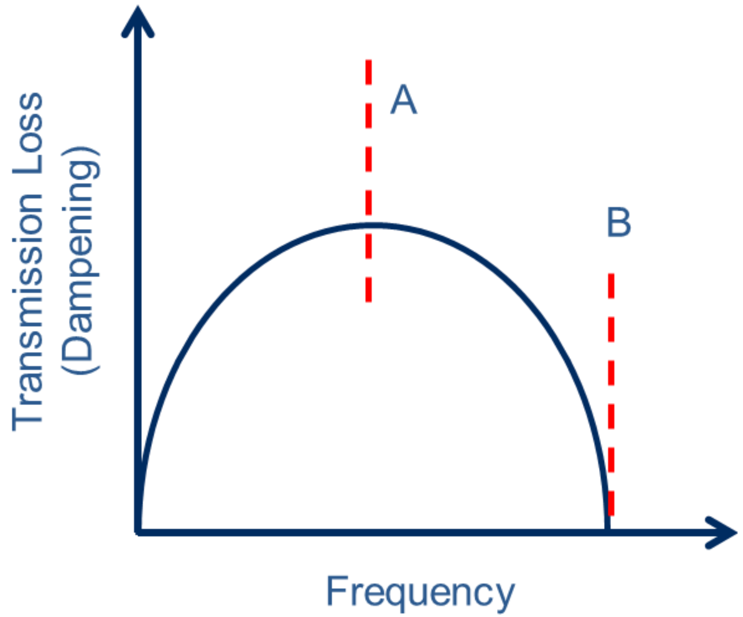
Large variety of damper (names) are used – all with pro’s and con’s





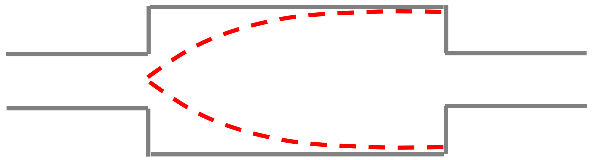
- Simple Bottle
- Double bottle with choke tubes
- Side branch dampers (mostly for pumps, gas filled)
- Reactive dampers
- Spherical resonators
- Etc.

Simple bottle – easy to implement but frequency regions of poor to no damping



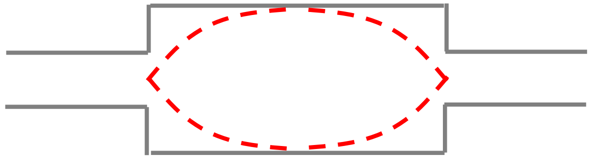
A: Maximum Dampening (destructive interference) (quarter wave of flow)

recip



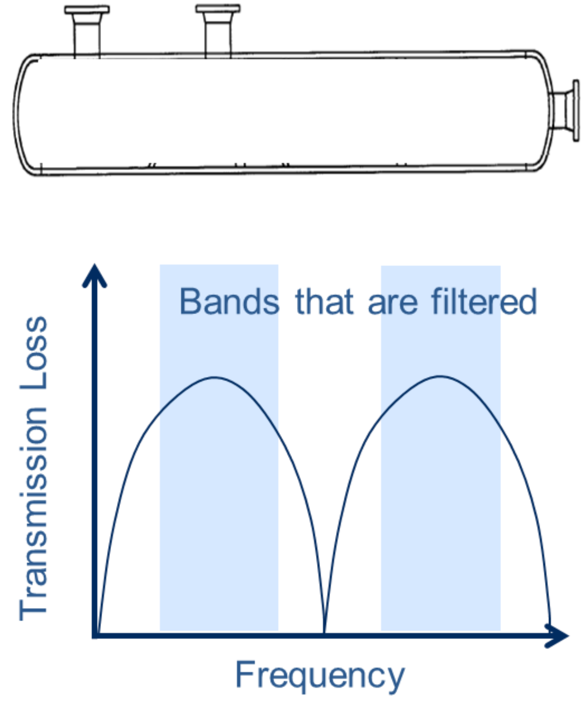
B. No Dampening/resonance, constructive interference (half wave of flow)

recip

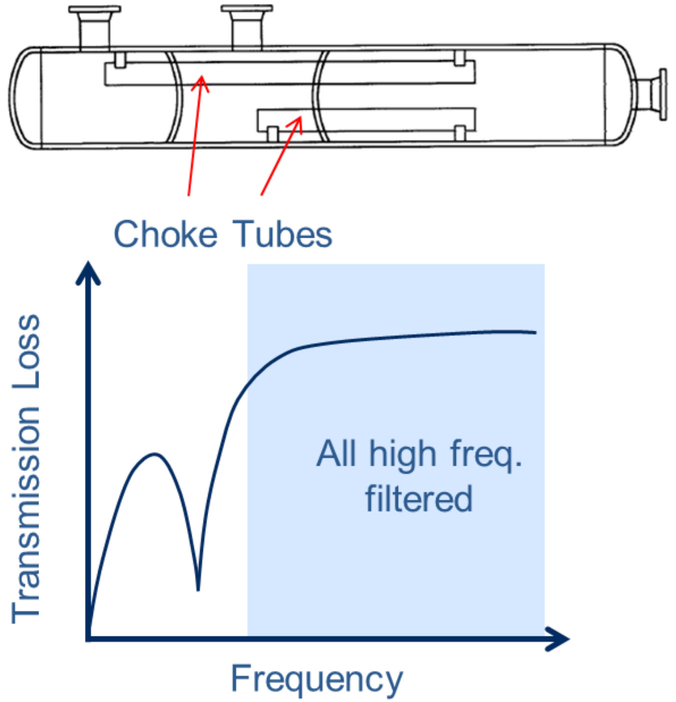


Choke tubes improve damping over larger frequency range

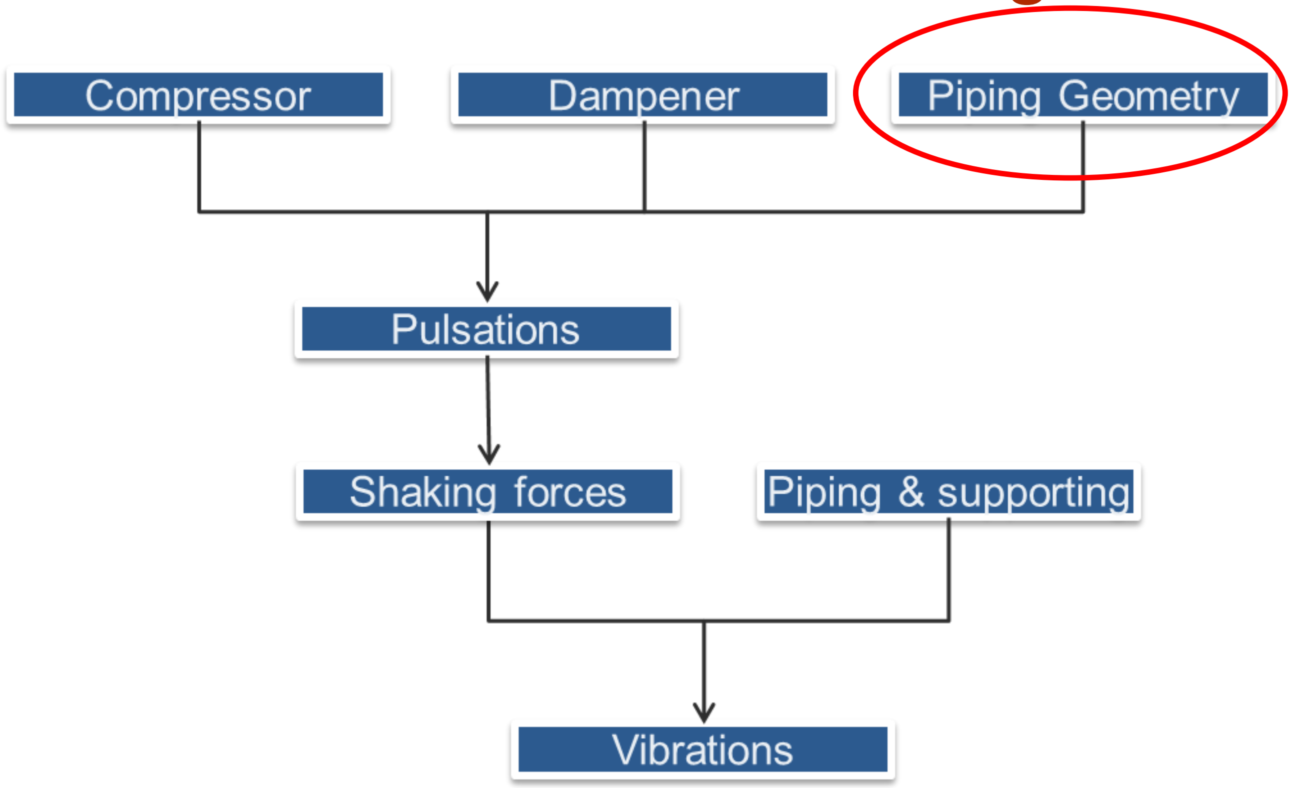
No choke tubes (band pass)



With choke tubes (low pass)

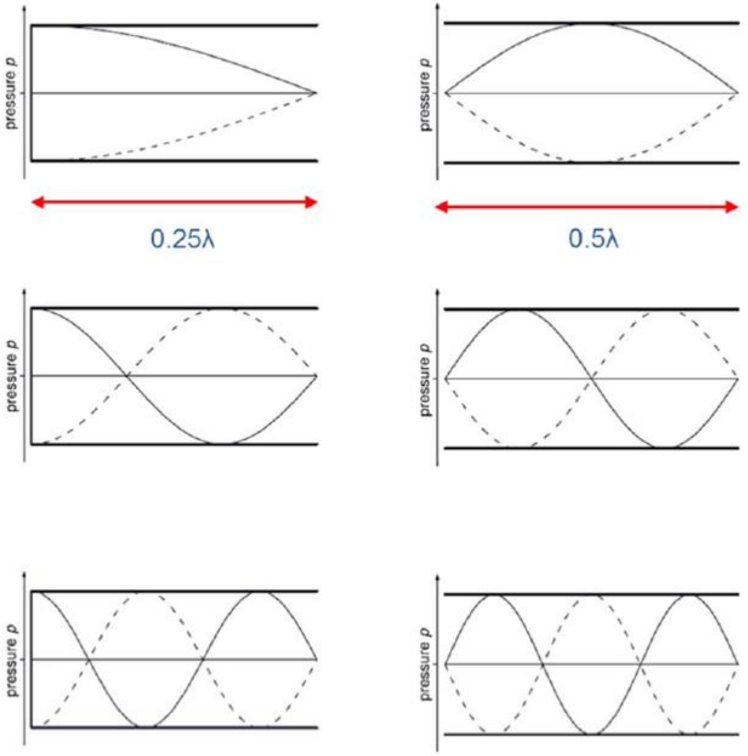


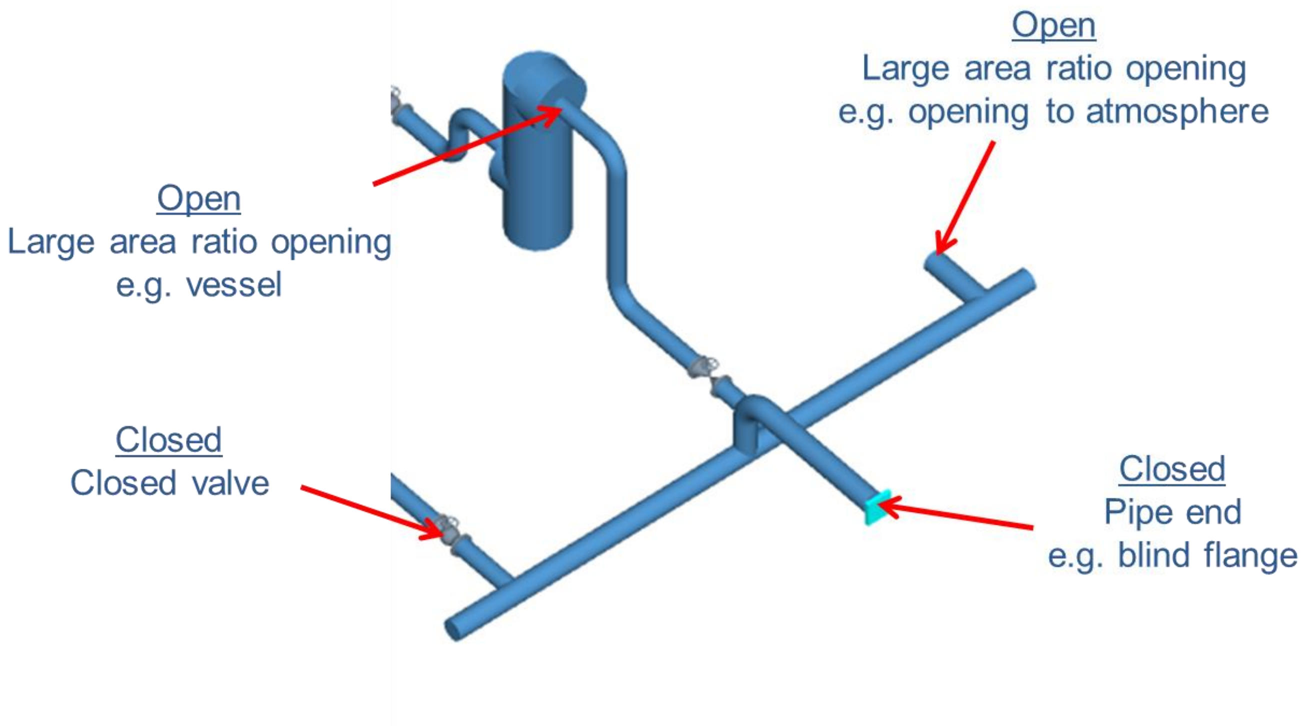
Schematic road map for pulsations and vibration understanding



Pulsations can excite acoustic resonance modes in the piping

Open-closed open-open





Overlap between excitation and acoustic resonance in the system can cause excessive pulsations

Open-closed system

$$f = \left(K - \frac{1}{2}\right)\frac{a}{2L}$$

Open-Open / Closed-Closed

$$f = K\frac{a}{2L}$$

f	resonance frequency
a	wave speed (300-1000 m/s)
L	is the length
K	wave number (1,2,3,…)

Typical problematic resonance modes

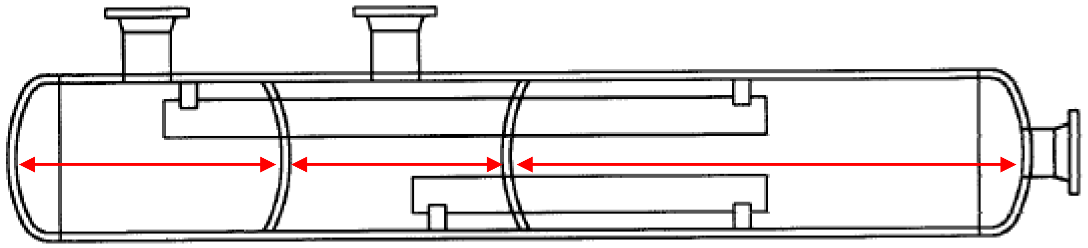
- **Compressor cylinder / pulsation bottle**
- End caps/baffle plates pulsation bottle
- Pulsation bottle / scrubber
- Pulsation bottle of compressors in parallel
- Closed side branch (relief line/circulation line)

Quarter wave, first mode only

Typical problematic resonance modes

- Compressor cylinder / pulsation bottle
- **End caps/baffle plates pulsation bottle**
- Pulsation bottle / scrubber
- Pulsation bottle of compressors in parallel
- Closed side branch (relief line/circulation line)

Half wave and higher order modes (causing high shaking forces on bottle)



Typical problematic resonance modes

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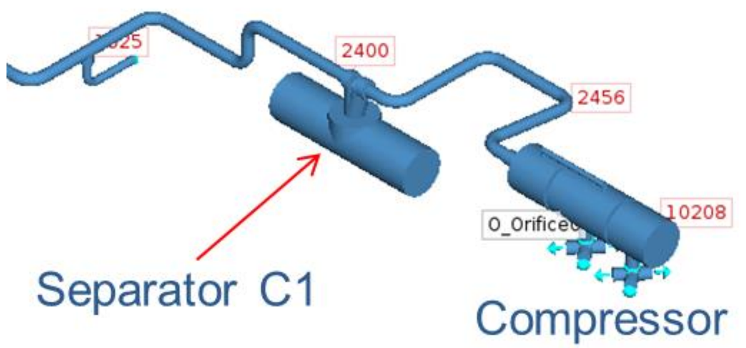
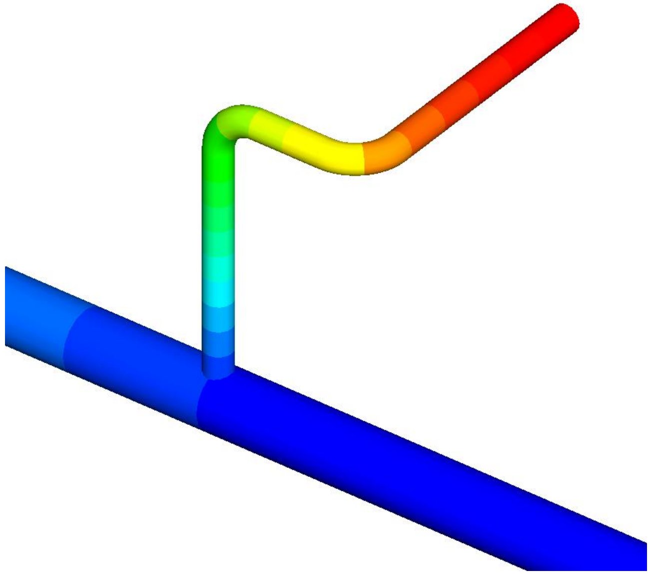
Half wave and higher order modes



Typical problematic resonance modes

- Compressor cylinder / pulsation bottle
- End caps/baffle plates pulsation bottle
- Pulsation bottle / scrubber
- Pulsation bottle of compressors in parallel
- **Closed side branch (relief line/recycle line)**

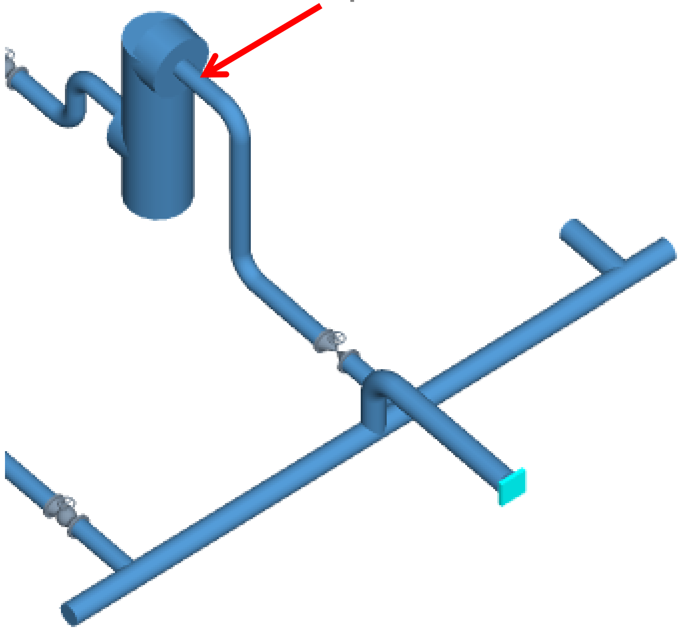
Quarter wave and higher order modes



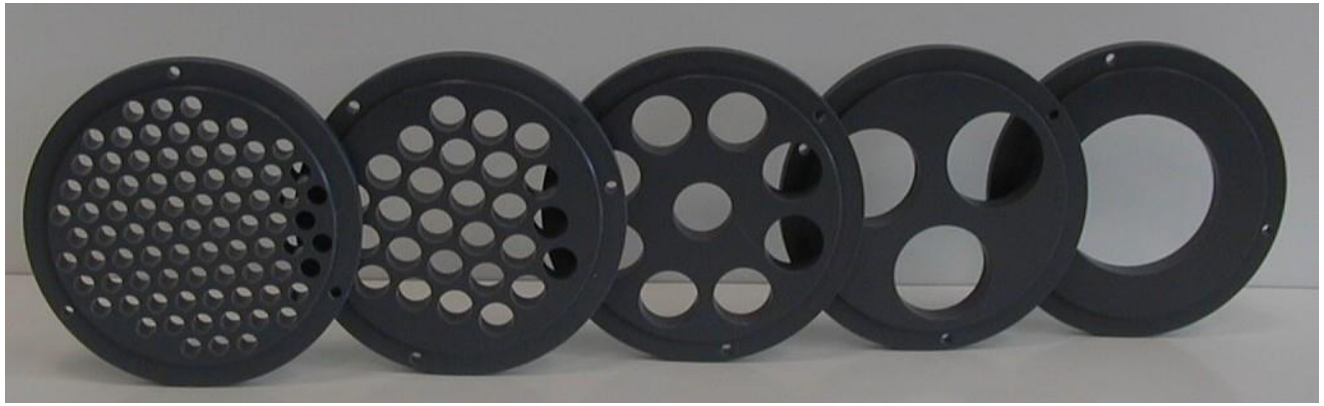
Orifice plates are effective at ‘breaking’ acoustic modes and are effective at vessel connections

- Used to eliminate specific modes of resonance on vessels (open)
- Very effective at eliminating specific resonance mode with low pressure drop
- Also used at inlet pulsation bottle to break high frequency resonance between bottle and cylinder.
- No suppression of pulsations not related to acoustic resonance

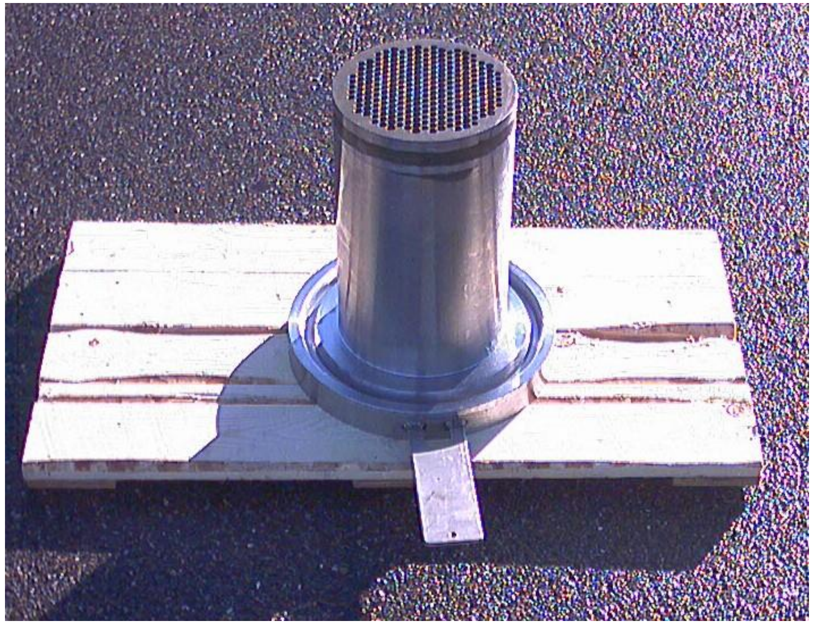
Optimal location for orifice



Multi-hole restriction orifice plates may be applied in case of increased frequencies

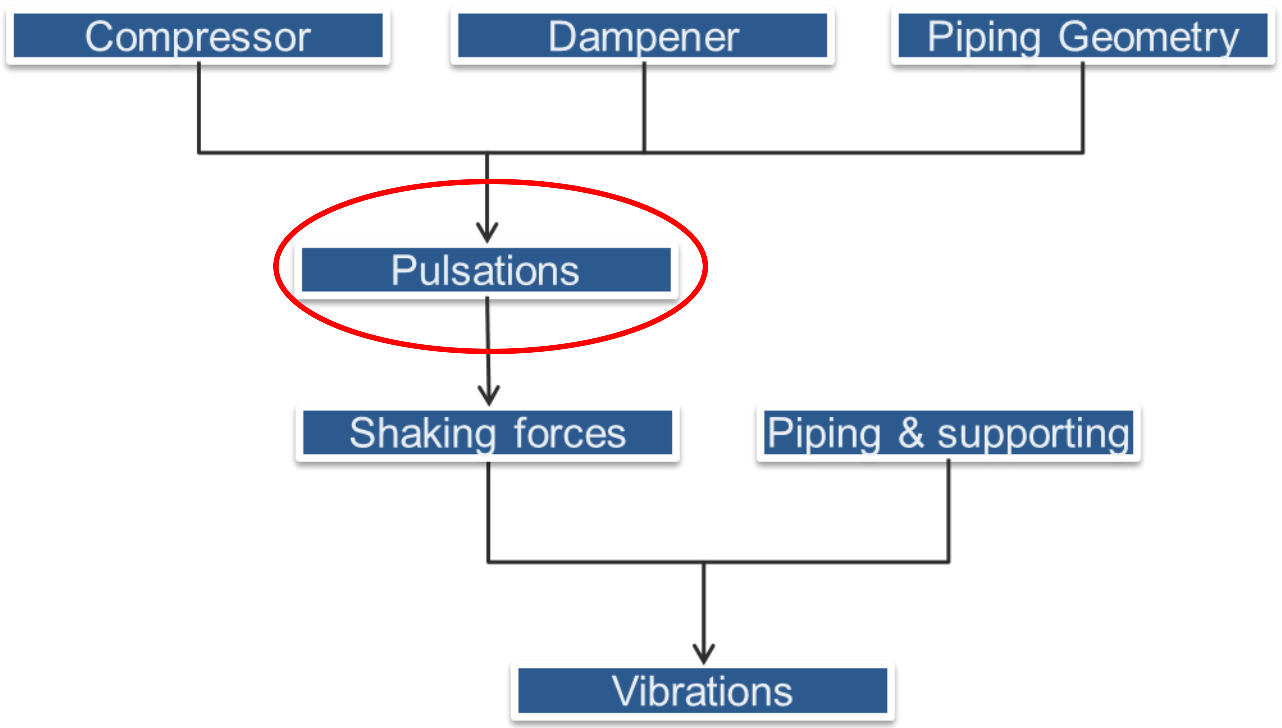


Multi-hole orifices



Special 'sleeved' construction, to place orifice at optimal location

Combination of excitation, damping and piping geometry determine final pulsations



Considerations with regards to pulsation analysis

Compressor (excitation)

- Modes of operation
  - Fixed speed vs variable speed (difficult to dampen all frequencies)
  - Single acting, double acting, variable clearance volume
  - Stepless flow control
- Geometry
- Speed
- Power / displaced volume per stroke

Considerations with regards to pulsation analysis

Damper

- Connected as close to compressor as possible
- Ideally designed to filter frequencies matching compressor characteristics

Piping

- Length relative to wave speed and frequency
- Process gas determining wave speed

Considerations with regards to pulsation analysis

General:

- Sufficient consideration of uncertainty in analysis.
- Analysis should consider a range of variations in terms of compressor speed/wave speed.
- Step size of variations sufficiently small to capture possible acoustic resonance with a good prediction of pulsation level.

Pulsations

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API 618 – several design approaches

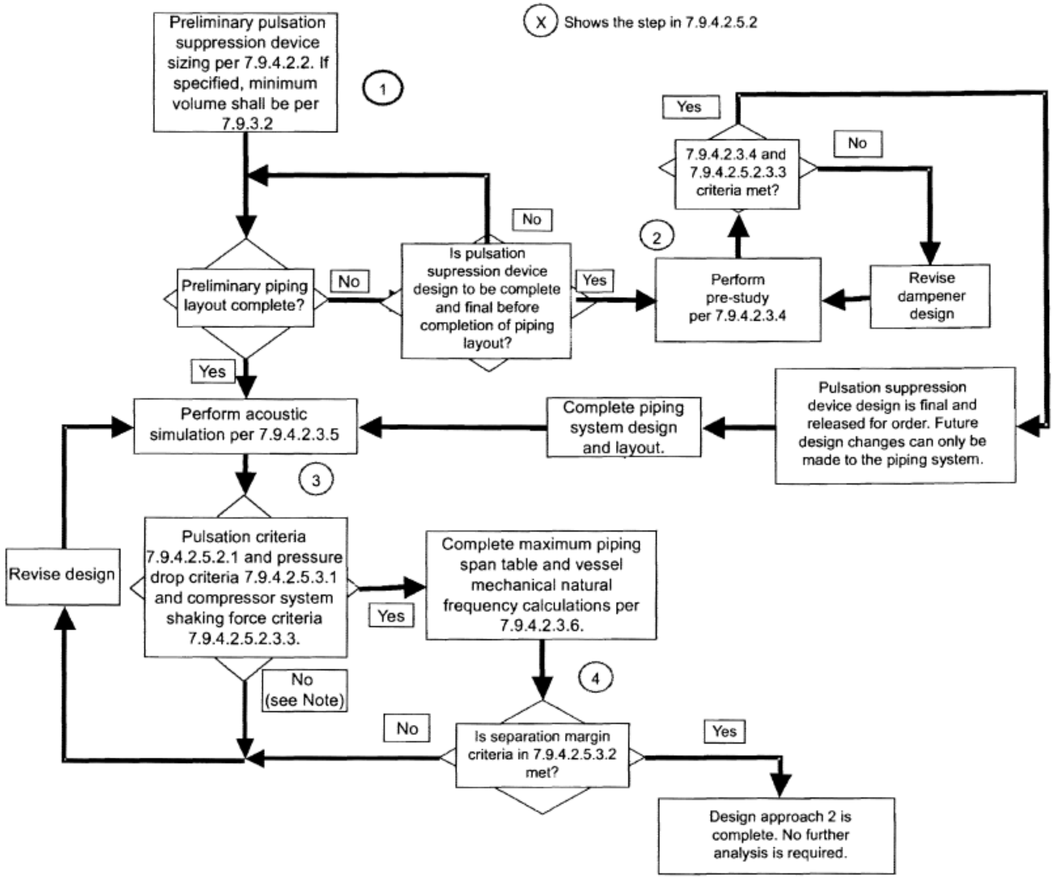
Higher level of design with increasing complexity:

- DA1: Empirical pulsation study
- DA2: Acoustic simulation and piping restraint analysis
- DA3: Acoustic simulation and piping restraint plus Mechanical Analysis

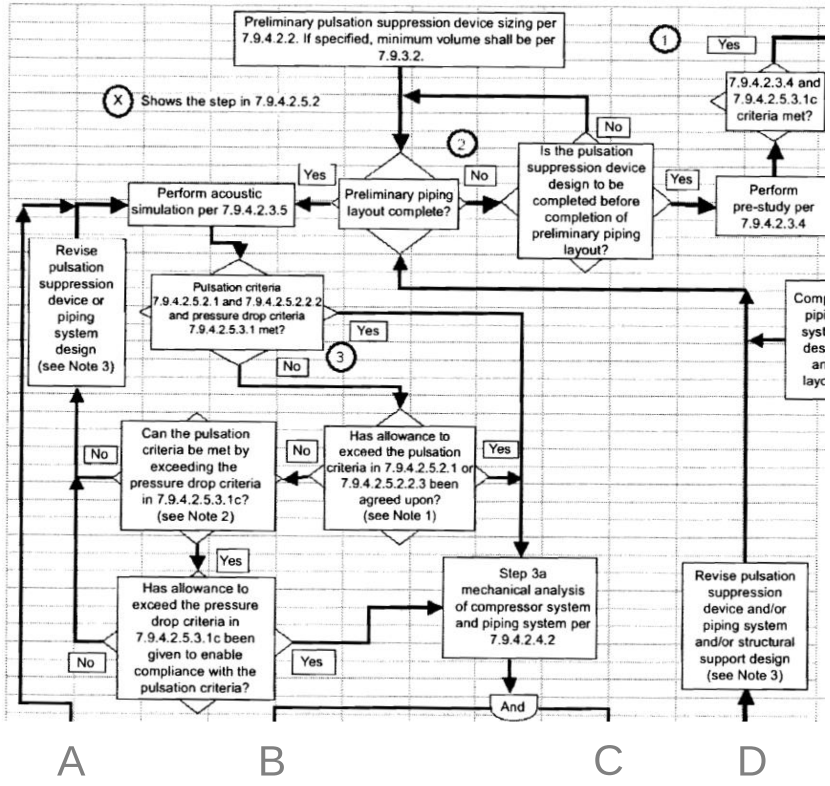
Table 6—Design Approach Selection

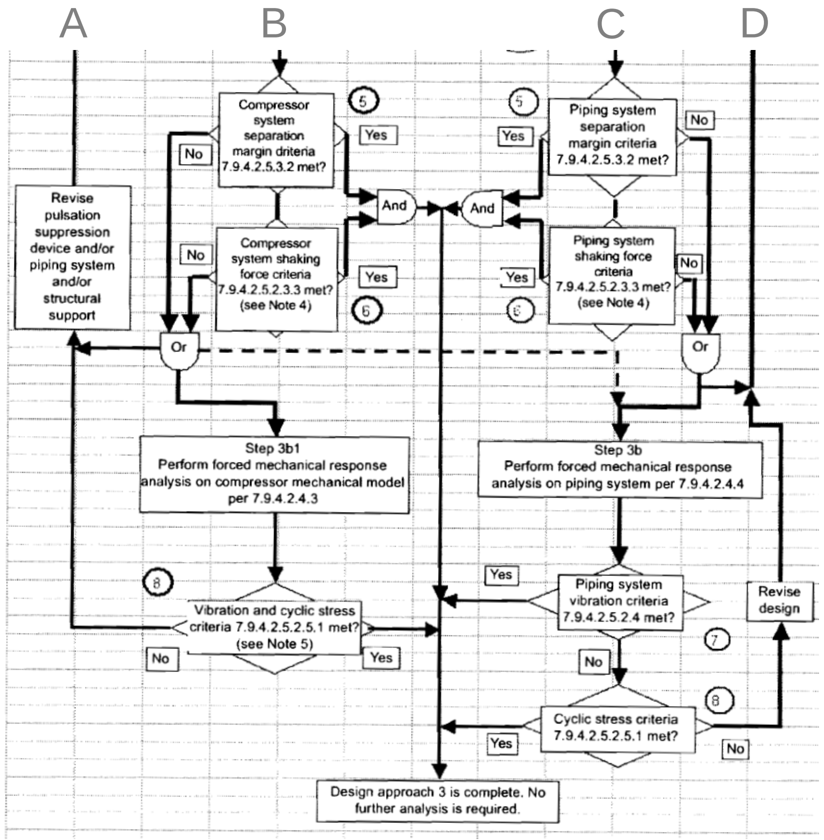
Absolute Discharge Pressure	Rated Power per Cylinder		
	Kw/cyl < 55 (hp/cyl < 75)	55 < Kw/cyl < 220 (75 < hp/cyl < 300)	220 < Kw/cyl (300 < hp/cyl)
$P < 35$ bar ($P < 500$ psi)	1	2	2
35 bar $< P <$ 70 bar (500 psi $< P <$ 1000 psi)	2	2	3
70 bar $< P <$ 200 bar (1000 psi $< P <$ 3000 psi)	2	3	3
200 bar $< P <$ 350 bar (3000 psi $< P <$ 5000 psi)	3	3	3

The work path for Approach 2 is found in Annex M



The work path for Approach 3 is found in Annex M



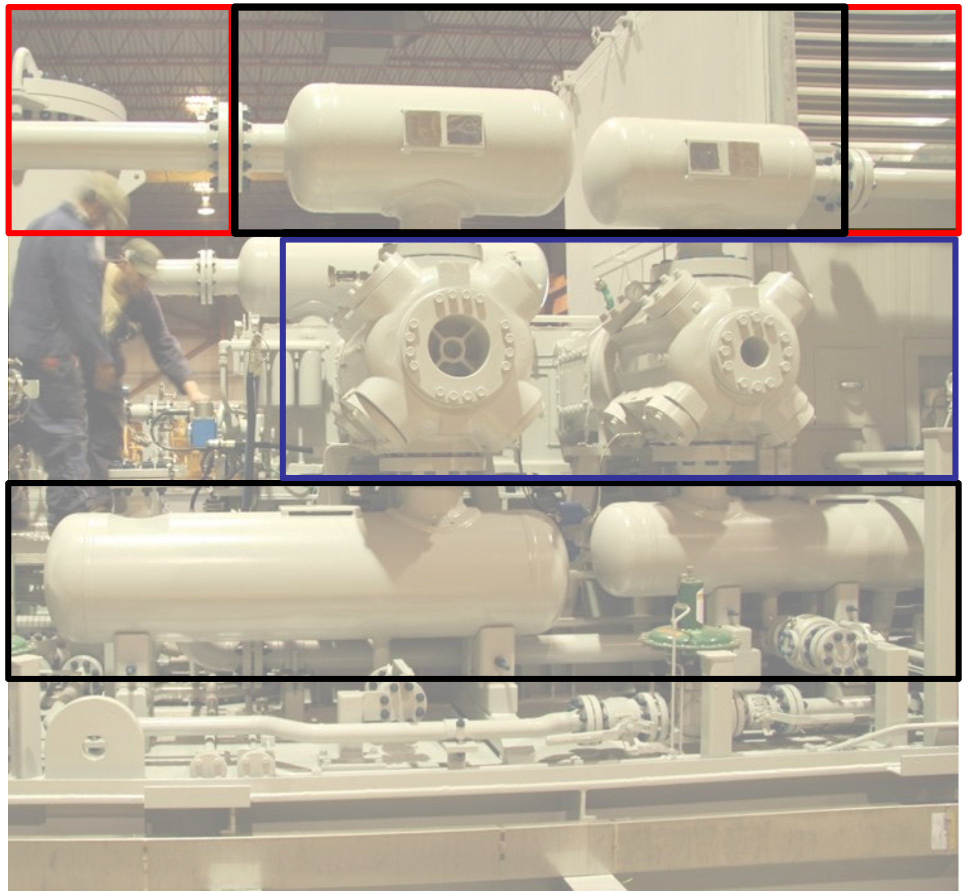


The API 618 recognizes three main regions

1. Compressor cylinder
- Pulsation check at flange

2. Pulsation Suppression Devices
- Pressure drop check
- Shaking force check

3. Piping beyond the PSDs
- Pulsation check
- Shaking force check



Pressure Pulsation limits
At the compressor cylinder flange

Pulsation should be limited to the lesser of 7% of average absolute line pressure or from:

$$P_{cf} = 3R\%$$

Where:

P_{cf} is the maximum allowable peak-peak pulsation [% of PL]

R is the stage pressure ratio

Checks if large resonance occurs in the cylinder passage which can cause flutter of the valves

Pressure Pulsation limits Downstream of Pulsation Suppression Devices (PSDs)

Pulsation should be limited as follows:

$$P_1 = \sqrt{a/(350)}\left(\frac{400}{(P_L \times D_I \times f)^{0.5}}\right)$$

Where:

P_1 is the maximum allowable peak-peak pulsation [% of PL]

a is the speed of sound [m/s]

P_L is the mean absolute line pressure [bar]

D_1 is the pipe inside diameter [mm]

f is the pulsation frequency [Hz]

Note: for a pre-study where the pipe routing is not final use 70/80% of P1 (see Section 7.9.4.2.3.4)

Pressure Pulsation check is done for each harmonic

Where f is:

$$f = \frac{N \times z}{60}$$

N is the shaft speed in rpm
z is 1, 2, 3,…. corresponds to the harmonics at the fundamental and higher order frequencies

Pressure drop limit for a pulsation suppression device

The pressure drop through a pulsation suppression device should be checked (large pressure drop may occur due to the internals)

ΔP is the maximum permissible pressure drop as a percentage of the inlet line pressure

ΔP is the lower value of (1) 0.25% of the line pressure or (2) that calculated using the formula below:

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

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Conclusion

Reciprocating compressors are intrinsic pulsation machines.

Pulsation mechanisms are well known.

Guidance on analysis (API 618) and simulation tools are available in order to mitigate potential pulsation issues in an early design stage