EFRC Training Workshop
Design and operation of reciprocating compressors

Pulsations & Vibrations

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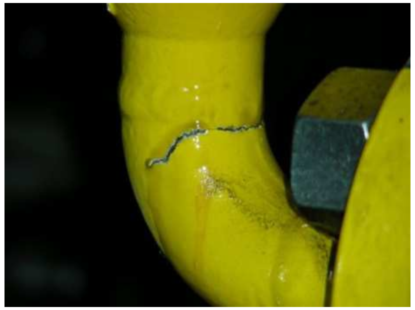
Outline

- Pulsations
- Vibrations
- Pulsation and vibration analysis according to API 618 standard, 5th edition

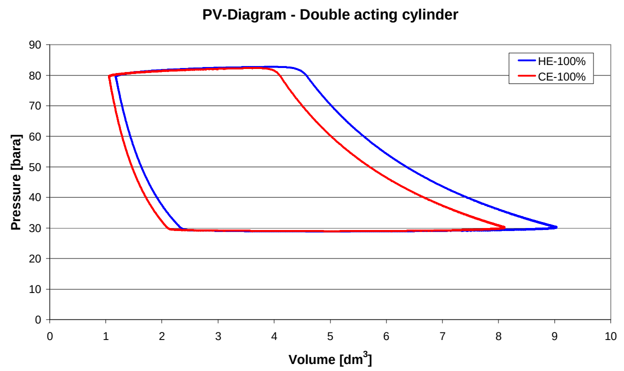
Why Pulsation & Vibration Analysis?

- Pulsations and vibrations should be minimized to avoid:
  - – Integrity issues in piping (fatigue)
  - – Increased wear of compressor parts
  - – Increased power consumption
  - – Flow metering errors
  - – Hammering of check valves

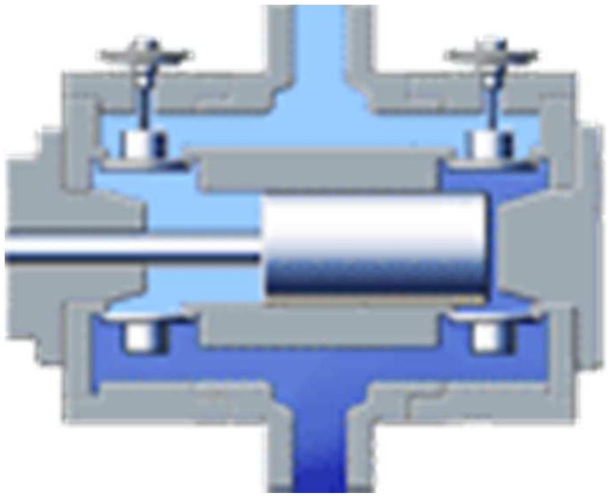


EFRC

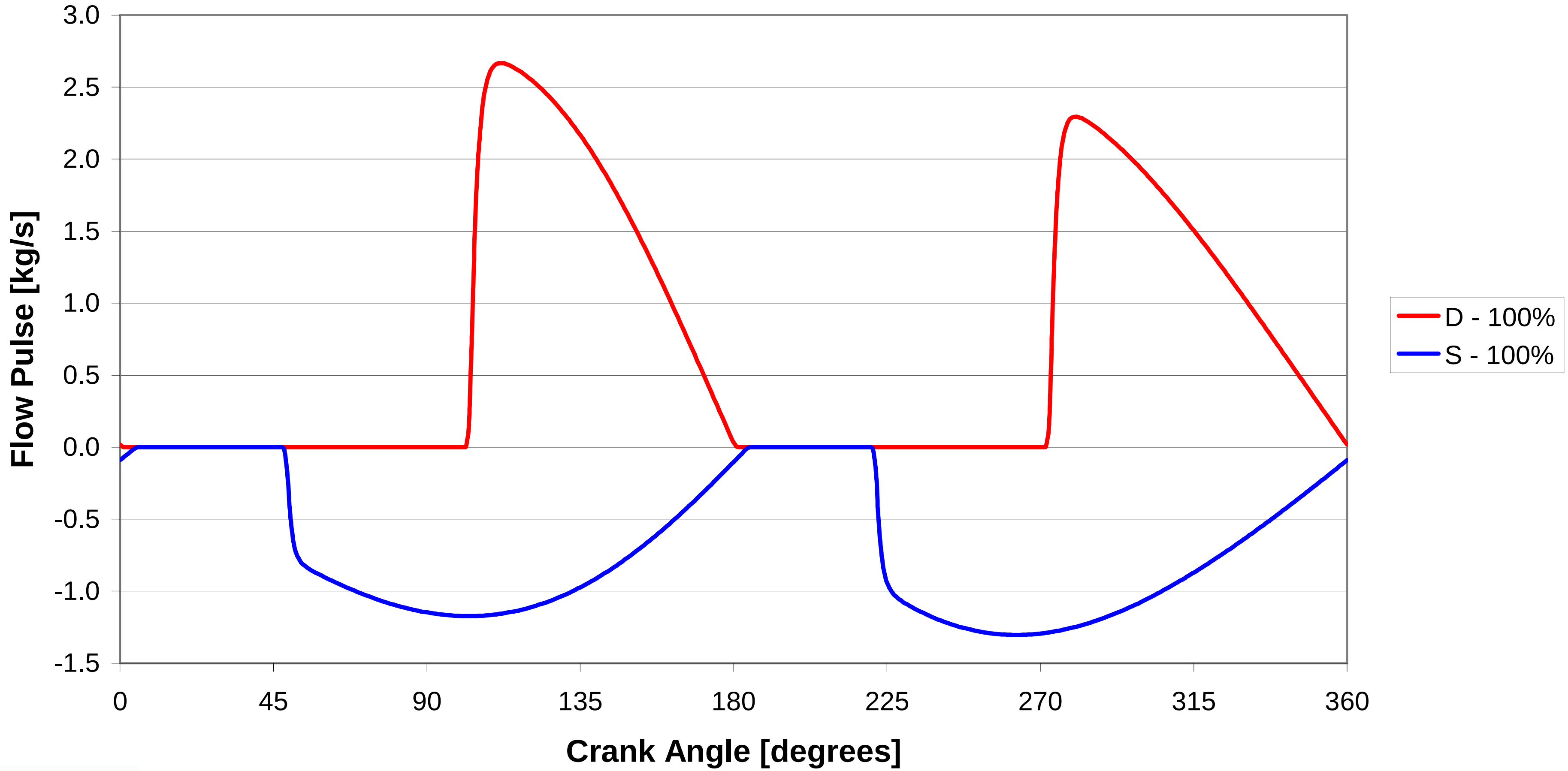
Pulsations



- Intrinsic feature of the reciprocating compressor
- Fluctuations of pressure and flow in the gas
- Occurs at suction and discharge side
- Strong interaction with the pipe system



Reciprocating, double-acting



Pulsation frequencies

- The frequency of pressure fluctuations is related to the *compressor speed*
- The frequency content of pulsations is essential for the propagation in pipe systems and interaction with the mechanical structure
- Fourier analysis is used to ‘match’ the time and frequency domain

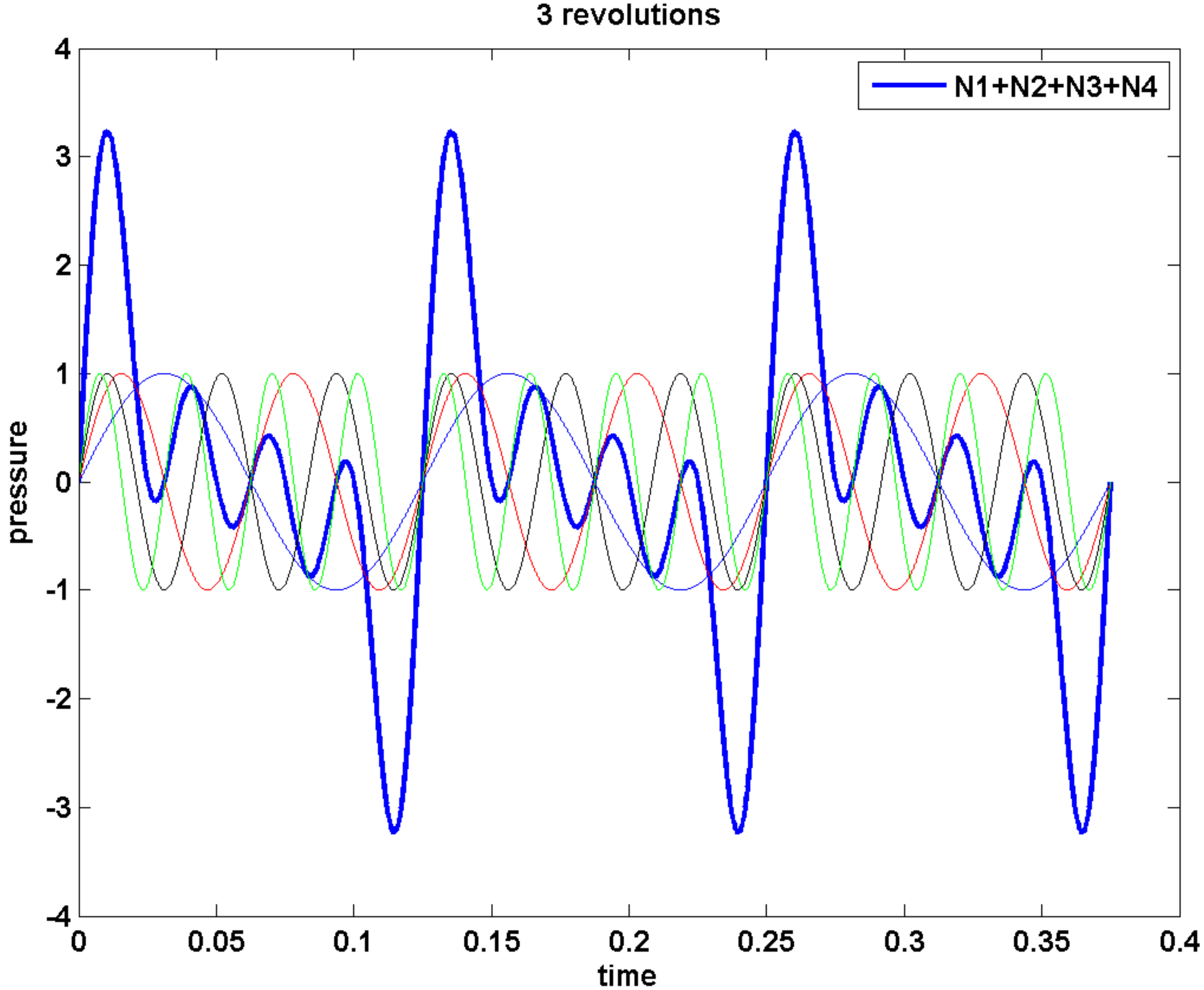
Harmonic distribution

- Higher harmonics of the fundamental are present in the pulsations:

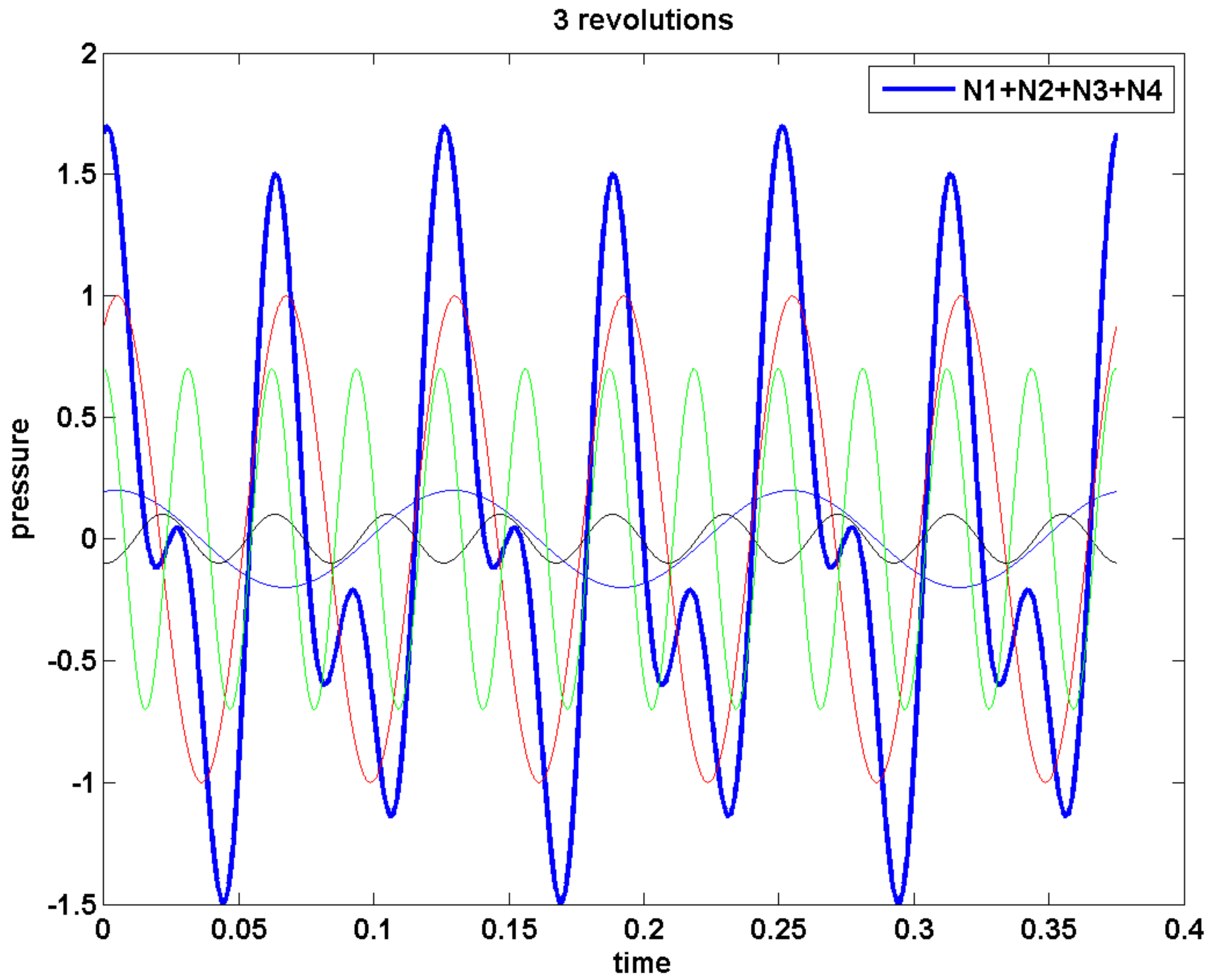
$$p(t) = \sum_i A_i \sin\left(2\pi f_i t + \phi_i\right) \qquad f_i = i * \frac{RPM}{60}$$

- Example: compressor running at 480 rpm
- Fundamental frequency = 8 Hz
- Consider first 4 harmonics (8,16,24,32 Hz)

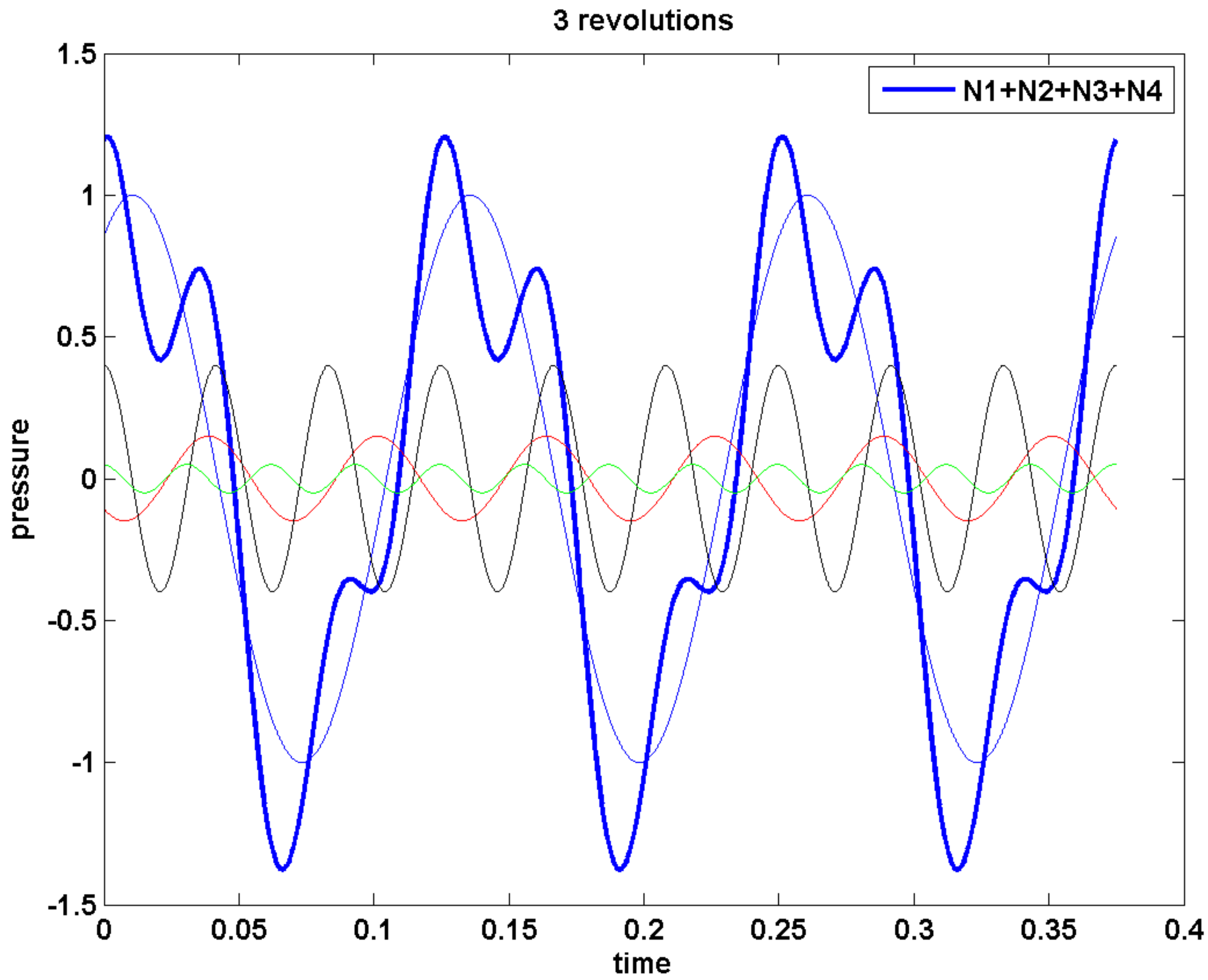
Illustration



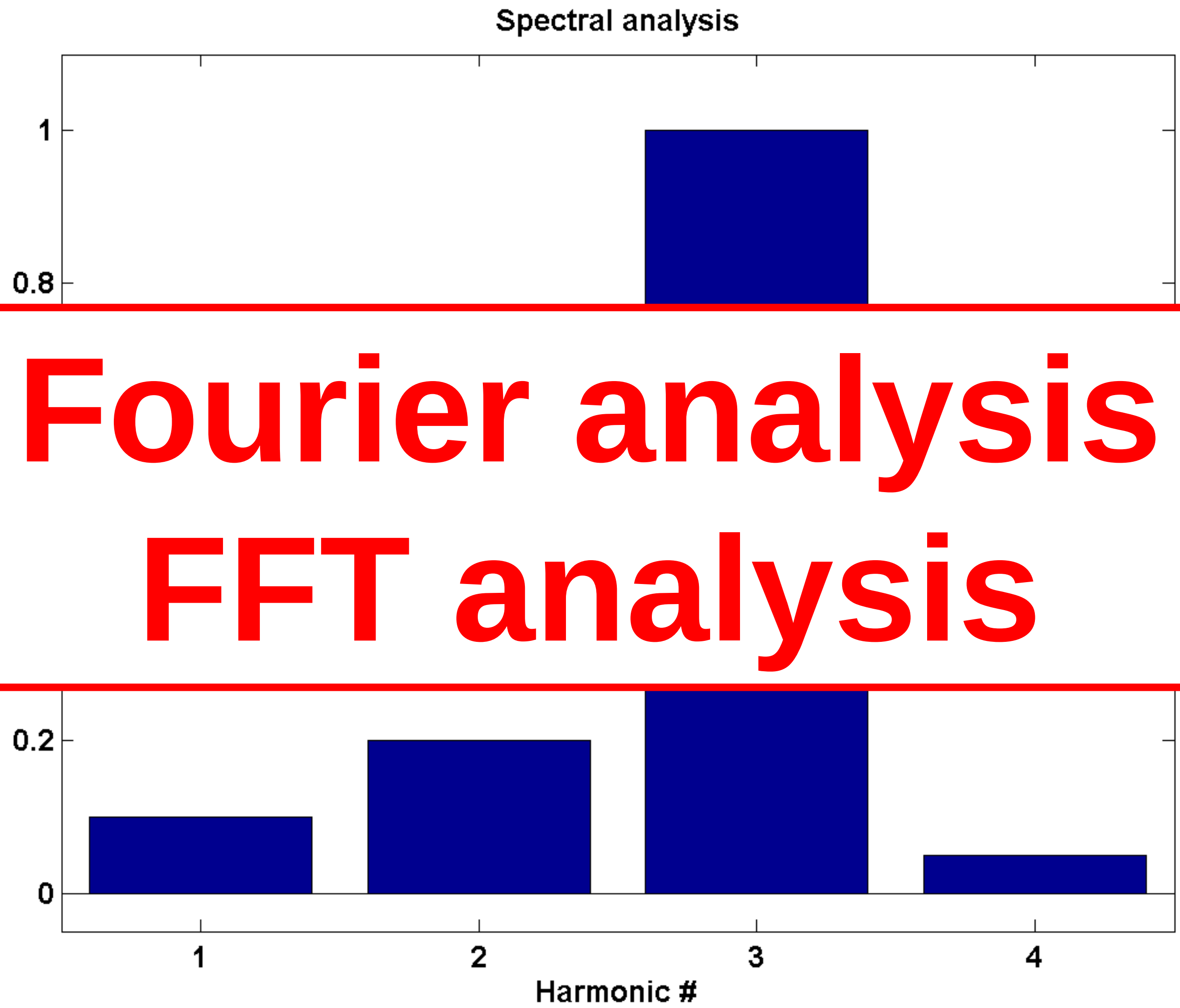
Double-acting cylinder



Single-acting cylinder



What's this???



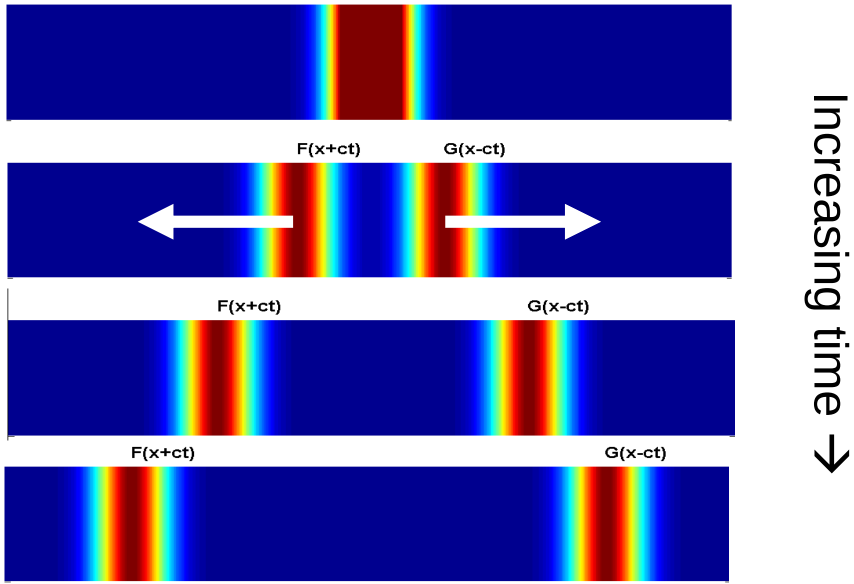
Fourier analysis
FFT analysis

Propagation

- Wave equation

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

- Pulsations are *acoustic waves*, propagating at the speed-of-sound *c*
- Pulsations travel up- and downstream

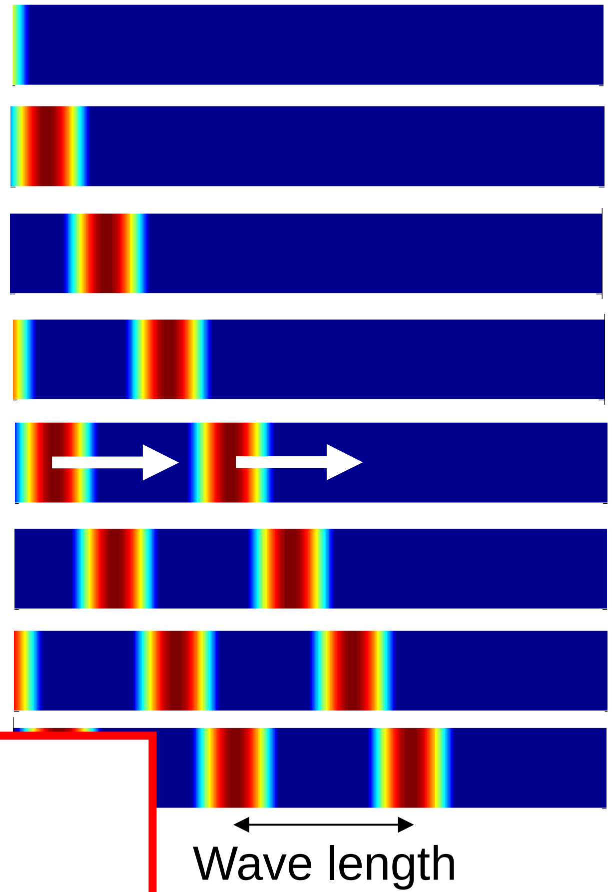


Speed of sound

- Increases with increasing T $c \sim \sqrt{T}$
  - Generally higher at discharge side than at suction side
- Decreases with increasing Molecular Weight $c \sim \frac{1}{\sqrt{MW}}$
  - CH_4 ~ 400 m/s
  - CO_2 ~ 260 m/s
  - H_2 ~ 1300 m/s

Wavelength

- Assume a harmonic source of frequency *f*
- Speed-of-sound *c*
- Acoustic wavelength $\lambda \sim \frac{c}{f}$



- Example:

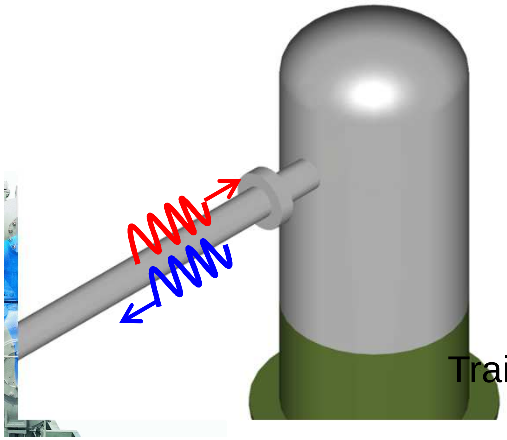
In general: $\lambda >> D_{pipe}$

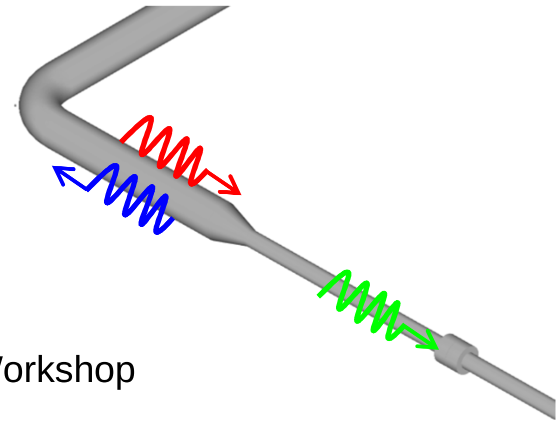
Damping

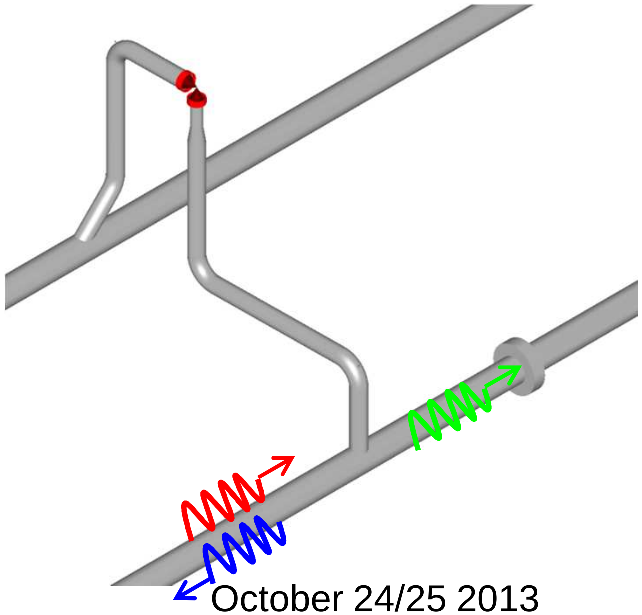
- Damping mechanisms
  - Turbulence, wall friction, heat exchange with the wall, viscosity …
- In general, damping is small
- Effective damping for L >> 10*λ
- Example: L >> 500 m
- Pulsations propagate over large distances!!

Reflection and transmission

- Acoustic waves reflect at ‘discontinuities’ in the piping
  - Diameter change
  - Temperature change
  - Side branches

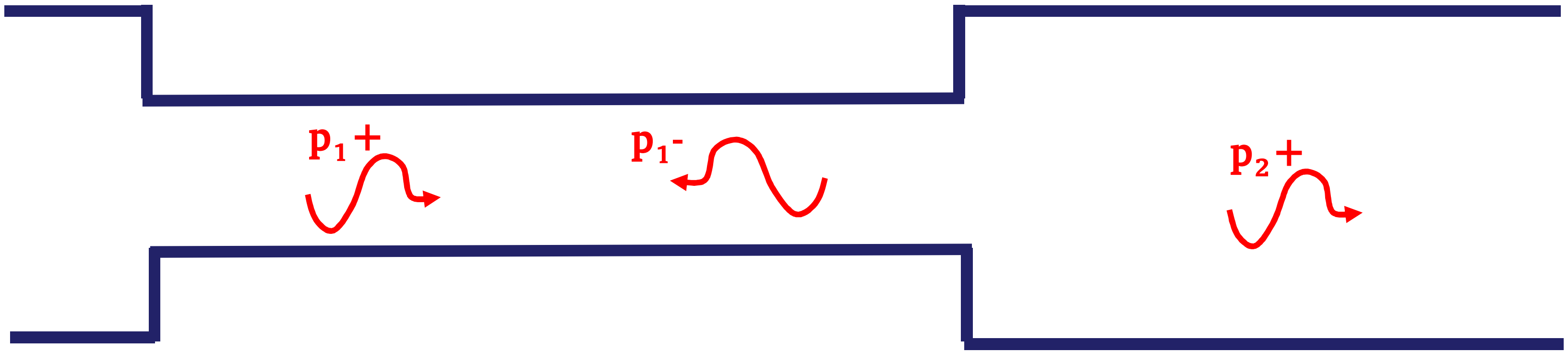






Interference of waves

- Constructive / destructive interference

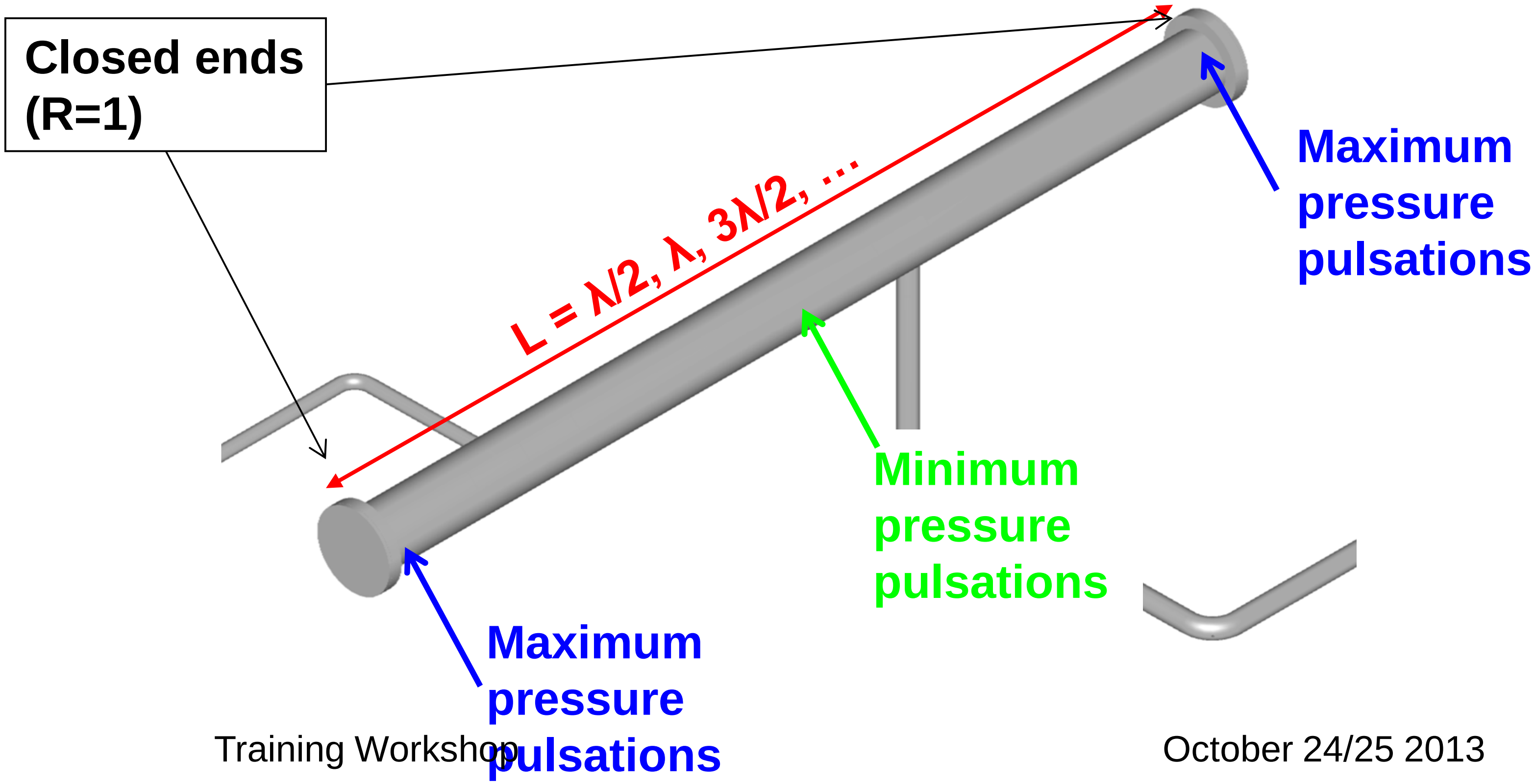


- Depending on the phase of the ...) and maxima (anti nodes) in pulsation amplitude

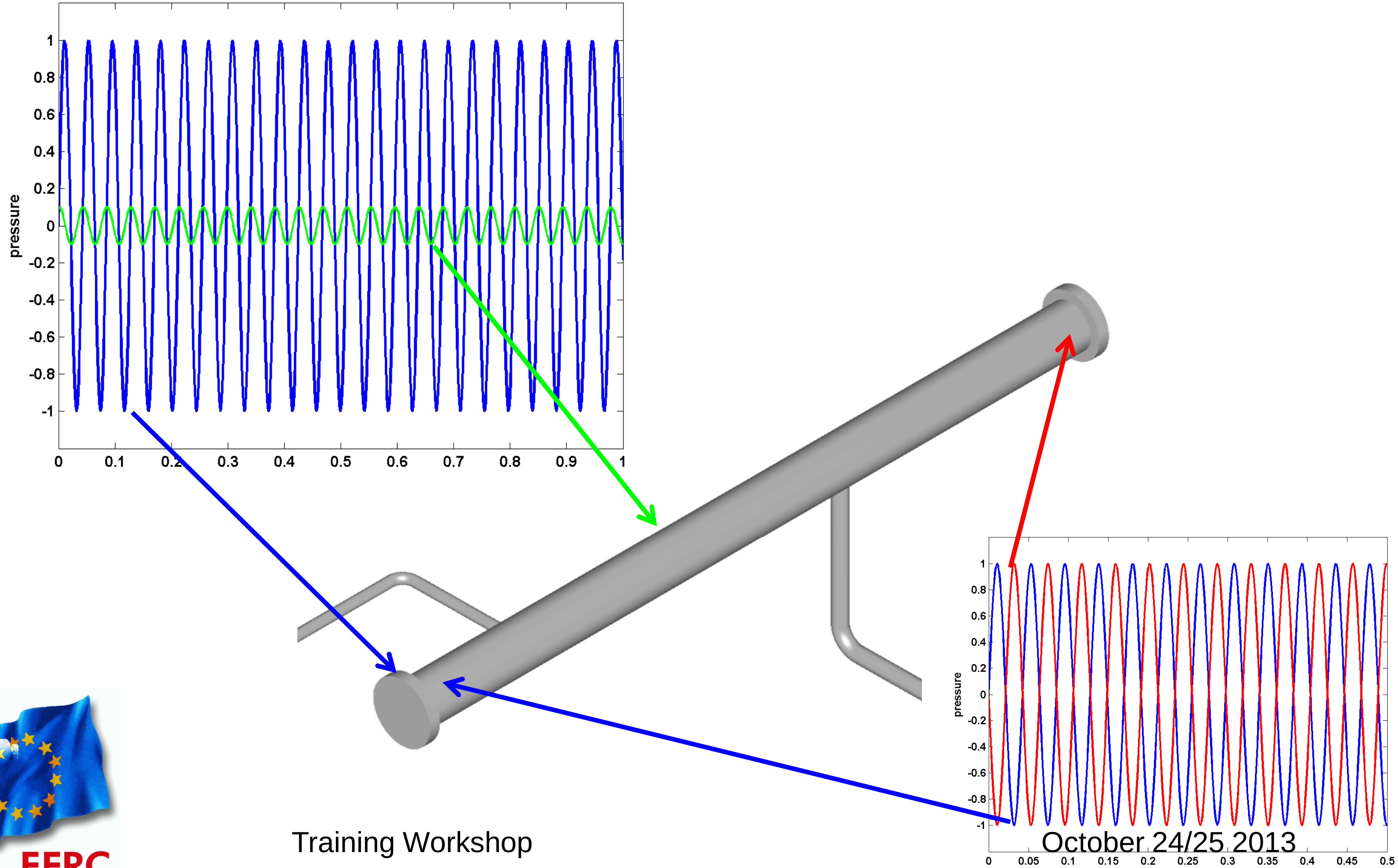
Acoustic resonance!

Standing waves

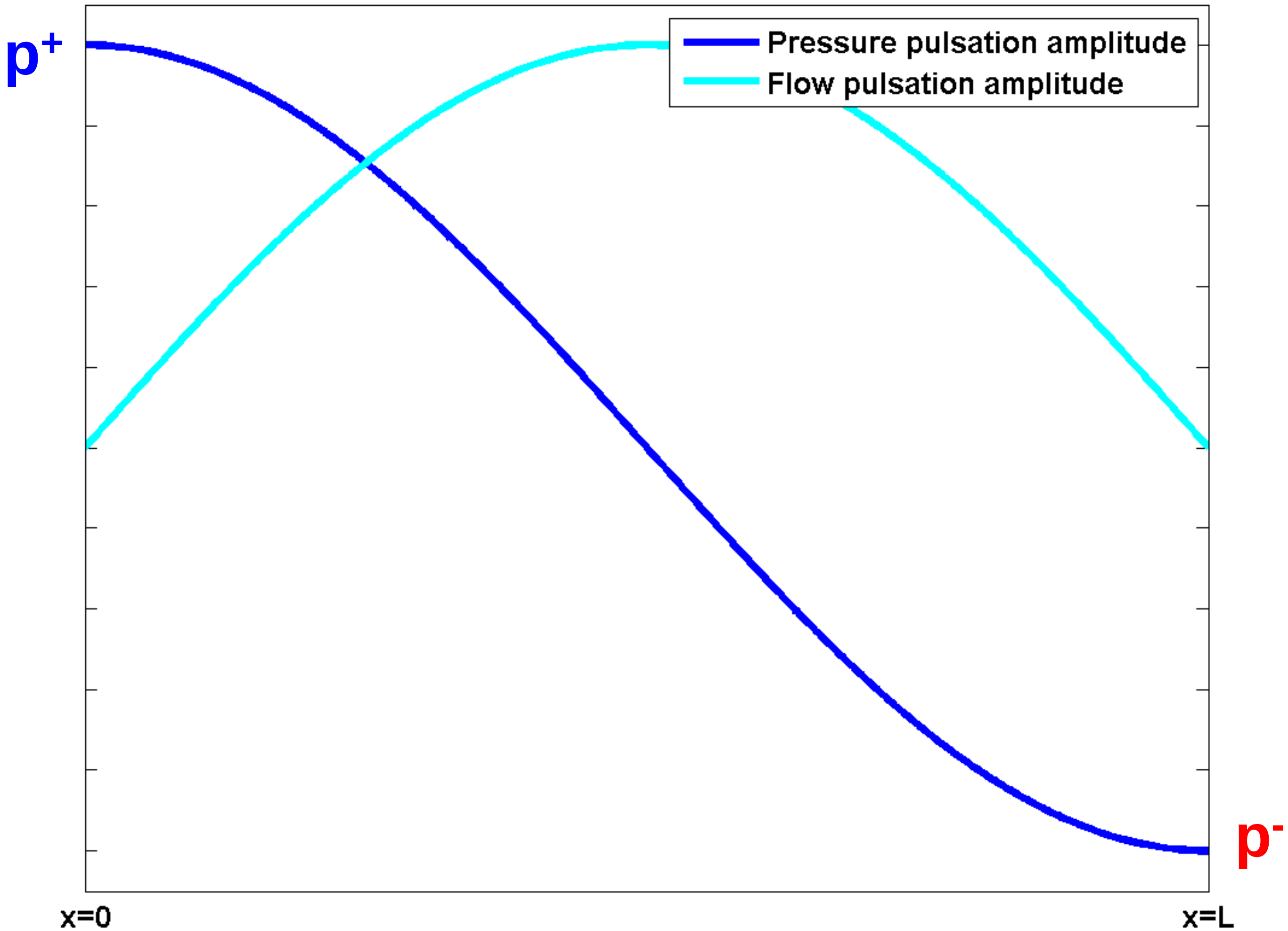
- Maximum amplification occurs when the wavelength matches the pipe length



Standing waves

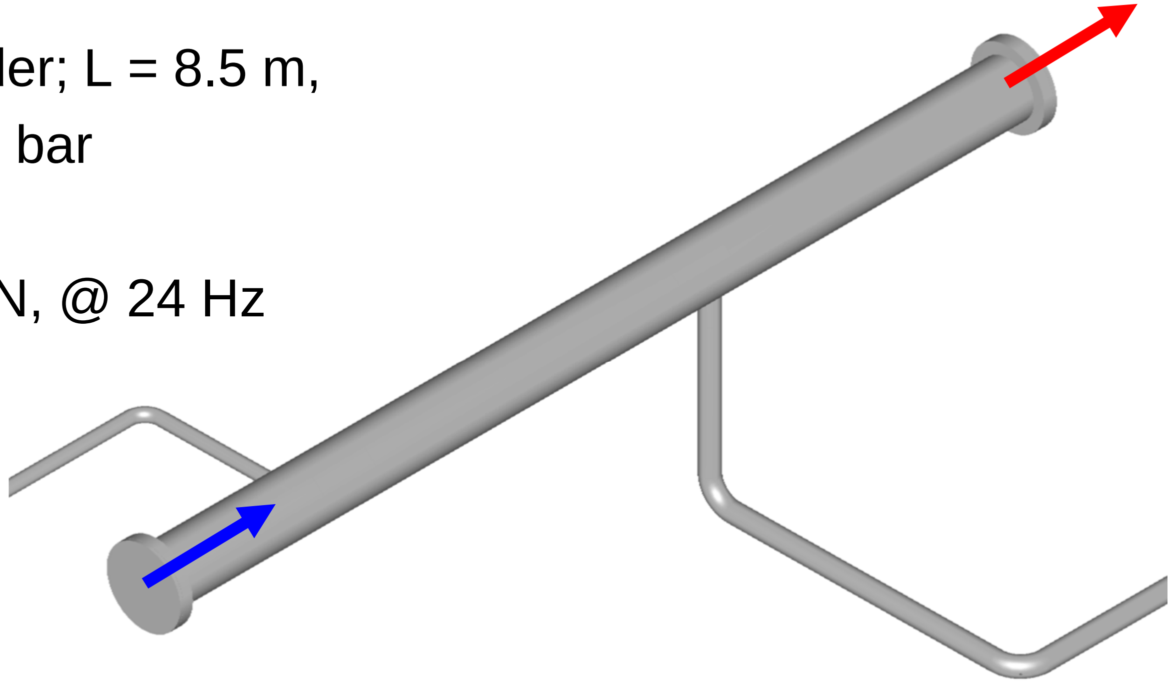


Standing wave (λ/2)



Shaking force on piping

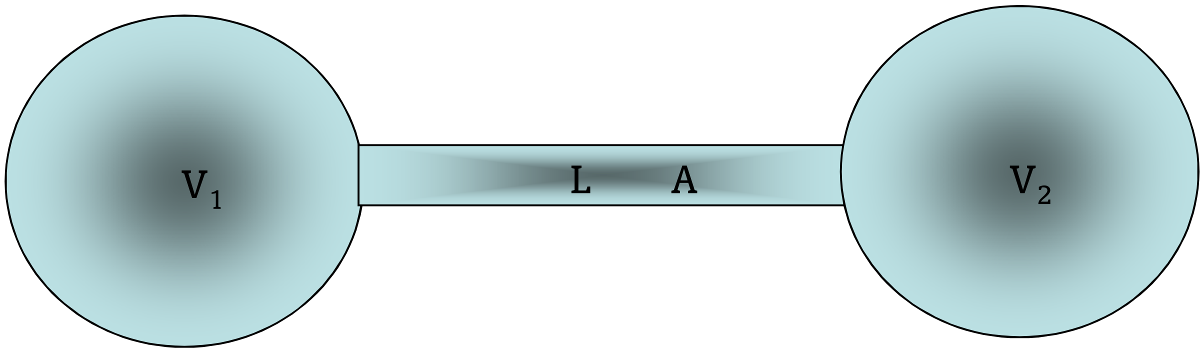
- 20” header; L = 8.5 m,
- P_0 = 200 bar
- p’ = 1%
- F = 40 kN, @ 24 Hz

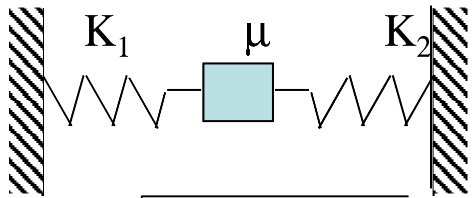


- Also acting on elbows, reducers, tees …

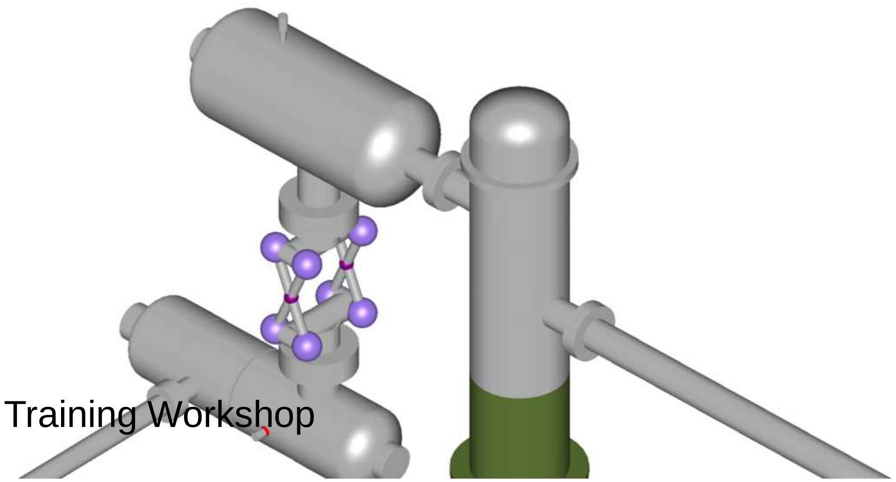
Helmholtz resonance

- Acoustic interaction between one or more volumes, connected by piping



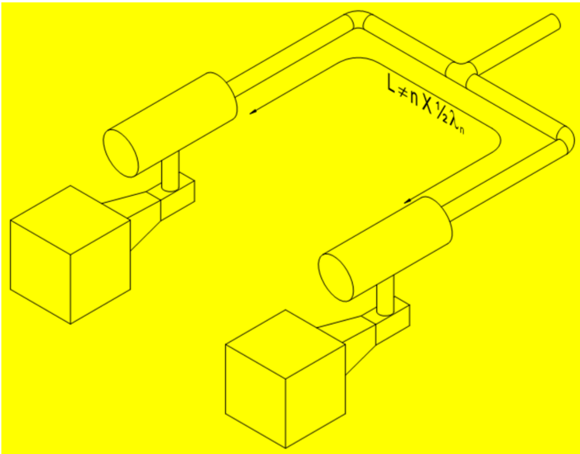


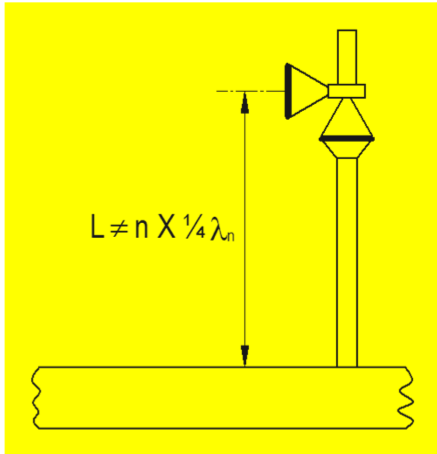
$$\omega_o = \sqrt{\frac{c^2 A}{L}\left(\frac{1}{V_1} + \frac{1}{V_2}\right)}$$



Control of pulsation issues

- Avoid coincidence of pulsation and resonance frequencies

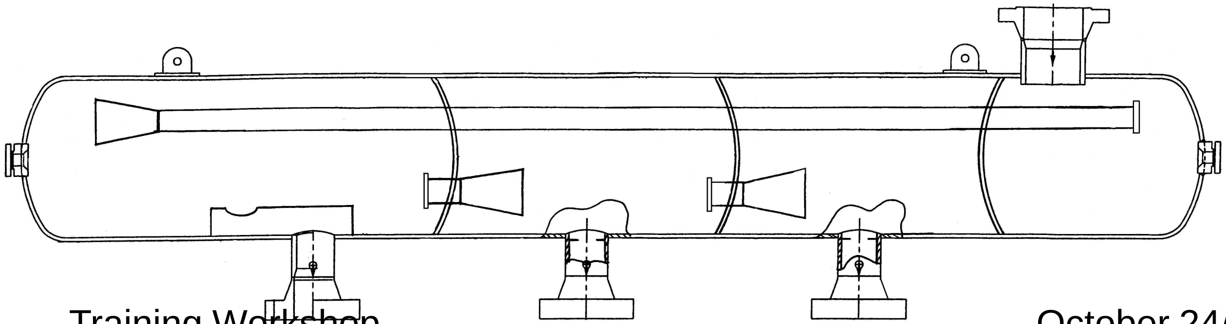




- Pulsation dampers
- Restriction orifice plates

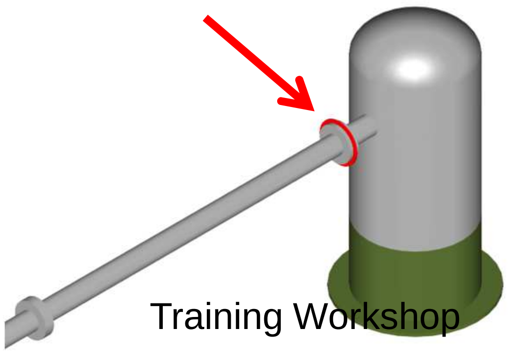
Pulsation dampers

- Mitigate the transfer of pulsations from the compressor to the piping,
- while reducing:
  - pulsations near compressor valves
  - Shaking forces on the damper
  - Pressure drop over damper



Restriction Orifice plates

- Mitigate the acoustic resonances in the piping
- Essential for performance:
  - – Location in the pipe system
  - – Pressure drop
  - – Layout of the orifice (for higher frequencies)



Mechanical Vibrations

Theory of Mechanical Vibrations

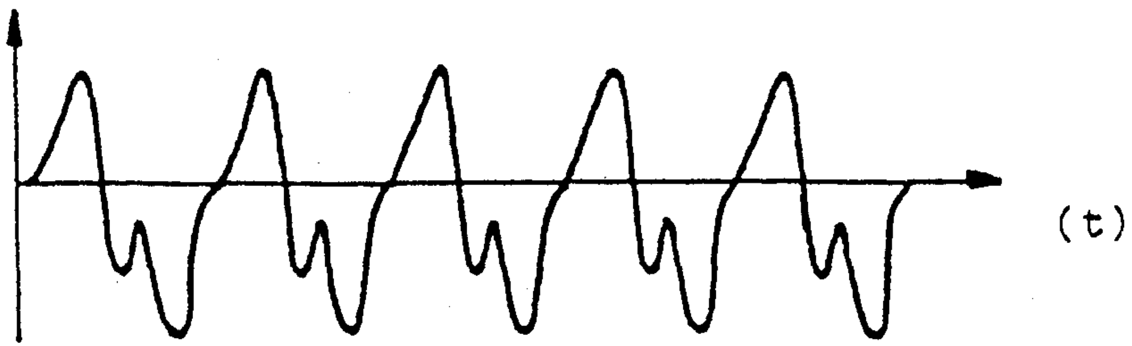
- Why a mechanical vibration analysis?
  - Too high vibration and cyclic stress (fatigue) can occur, even in case pulsation levels are within the allowable levels: mechanical resonances
- What is a vibration?
  - A vibration is a more or less regular movement of a body as a function of *time*
- Examples:
  - Oscillating motor due to internal combustion
  - Oscillating flag stag due to the wind
  - Pipe vibrations caused by a reciprocating compressor (pulsation-induced shaking forces and mechanical loads)

Vibration Forms

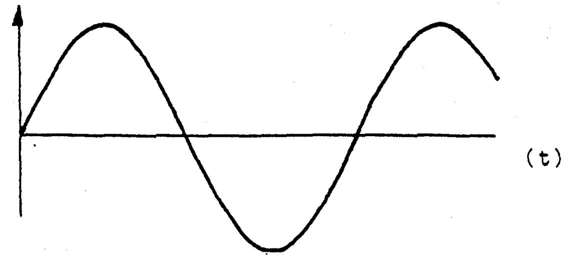
- Non periodic:
  - A vibration which amplitude is not repeated
- Examples:
  - Opening of a relief valve
  - Vibrations of a bridge due to traffic
  -

- Periodic vibration:
  – Vibration whereby the amplitude is repeated after a discrete time period

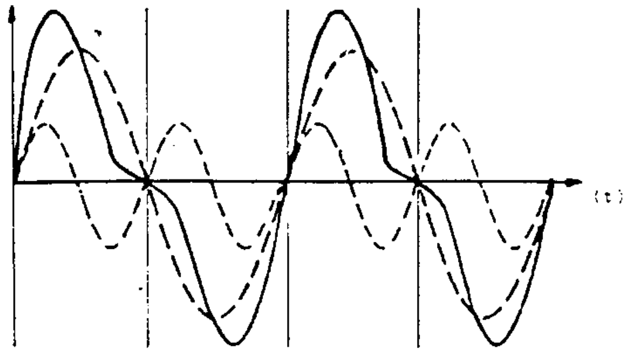
- Examples:
  – Combustion forces in a cylinder
  – Unbalanced loads of reciprocating compressors
  –



- Harmonic vibration
  - Vibration of which the amplitude is a sine
  - Can only occur when the excitation force exists of only one frequency component
- Example:
  - Unbalance of a rotor

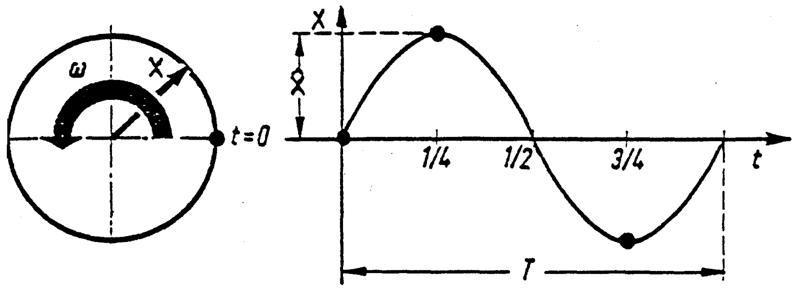


- Two or more summed harmonic vibrations with different frequencies:
  - periodic vibration but not harmonic



Definitions of Vibrations

- Period (T):
  - The time after which the vibration repeats
- Frequency:
  - the reciprocal value of T (f=1/T) [Hz]
- Amplitude (X):
  - The maximum value of the sine over one period
- Phase angle:
  - the argument of the sine function as follows: $X = x\sin(\omega t + \theta)$
- Circle frequency:
  - 2π times the frequency f

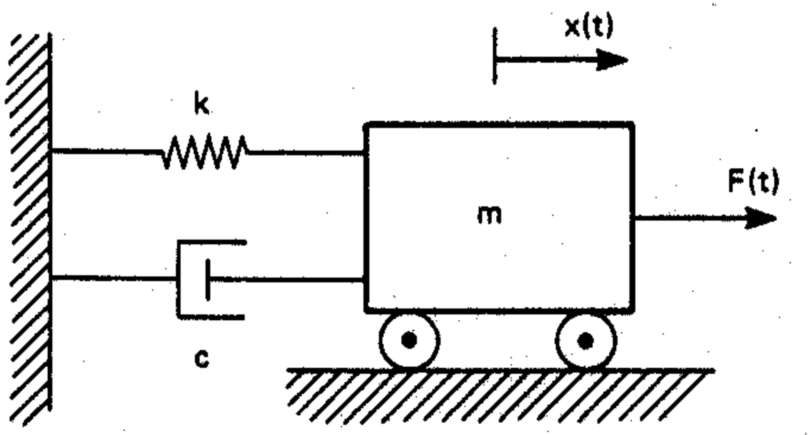


Single Degree Of Freedom Systems (SDOF)

- General equation of motion:

$$m\ddot{x} + c\dot{x} + kx = Fe^{j\omega t}$$

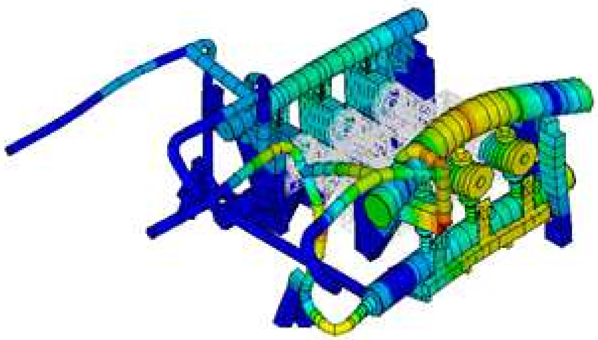
where
x displacement
$\dot{x}$ velocity
$\ddot{x}$ acceleration
F excitation force
j $\sqrt{-1}$
ω excitation frequency
k spring stiffness
c viscous damping
m mass



Free <u>Undamped</u> Vibration

- Solving the following equation: $m\ddot{x} + kx = 0$
- Gives following solution: $\omega_0 = \sqrt{k/m}$

  ω_0 = mechanical natural frequency
- Mode shape (eigen vector):
  - It is the *ratio* of the amplitudes at various points
  - Represents a deformation pattern of the structure
  - for the corresponding natural frequency
  - The actual amplitude depends on initial conditions and position and magnitude of forces

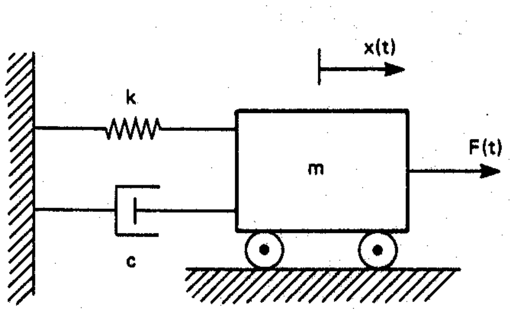


Forced Vibration of SDOF Systems

- General equation of motion

Pulsation forces and mechanical loads

$$m\ddot{x}+c\dot{x}+kx=Fe^{j\omega t}$$



- Steady state solution:

$$x=\left[\frac{1}{1-(\omega/\omega_0)^2+j\,2\,\zeta\,\omega/\omega_0}\right]\frac{F}{k}\,e^{j\omega t}$$

- Displacement x is proportional to the applied force, the proportionality factor being

$$H(\omega)=\left[\frac{1}{1-(\omega/\omega_0)^2+j\,2\,\zeta\,\omega/\omega_0}\right]$$

complex
frequency response

– The peak at frequency:

$$\omega = \omega_0 \sqrt{\left(1 - 2\zeta^2\right)}$$

– Damping ratio:

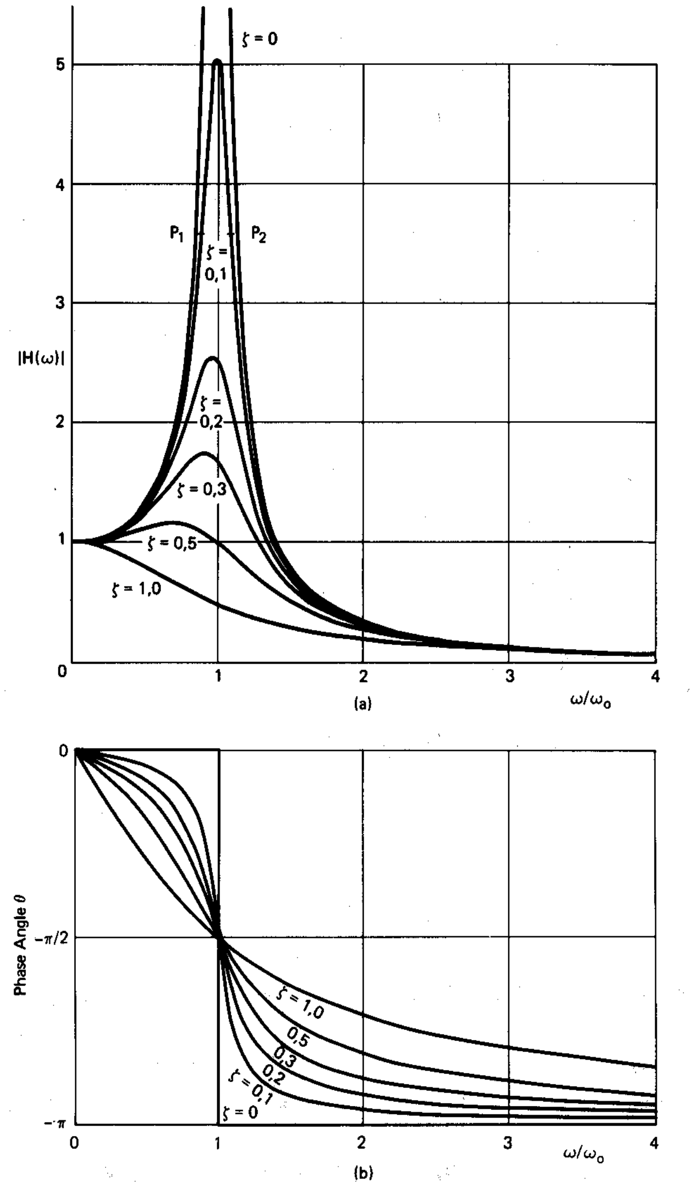
$$\zeta = \frac{c}{c_c} = \frac{1}{2}\frac{c}{\sqrt{km}}$$

– The peak value of |H (ω) | is given by:

$$\left|H(\omega)\right| = \frac{1}{2\zeta\sqrt{1-\zeta^2}}$$

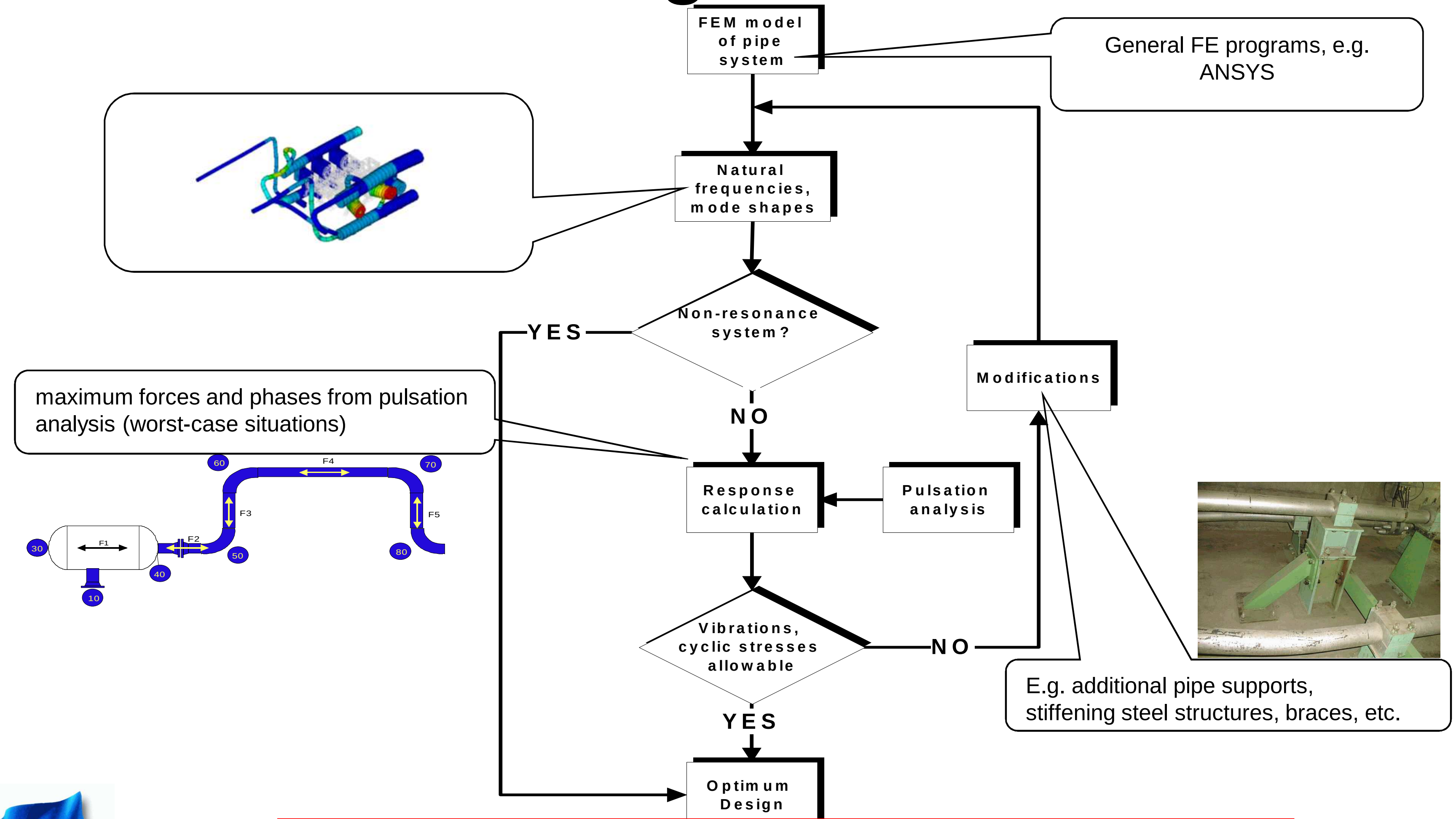
– For light damping ($\zeta < 0.05$) the magnification is:

$$\left|H(\omega)\right| \cong \frac{1}{2\zeta} = Q$$



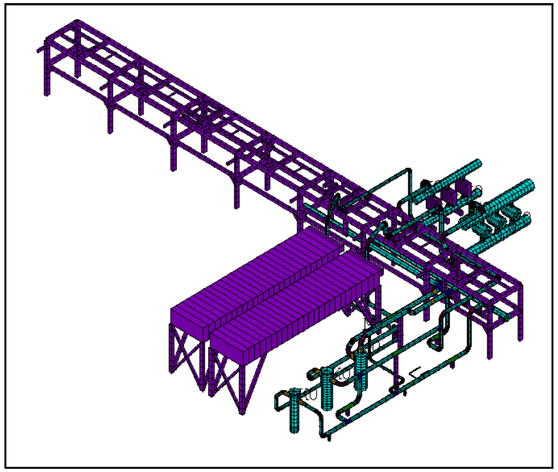
|H (w) | and q as a function of w/ w0

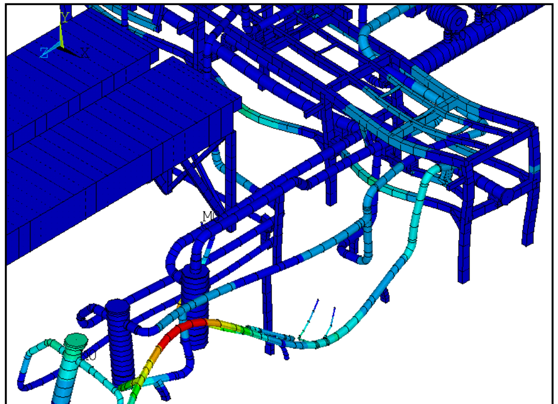
Mechanical response analysis according to API 618



Avoid unacceptable vibrations and fatigue failures

Example of an Underground Gas Storage (UGS) system



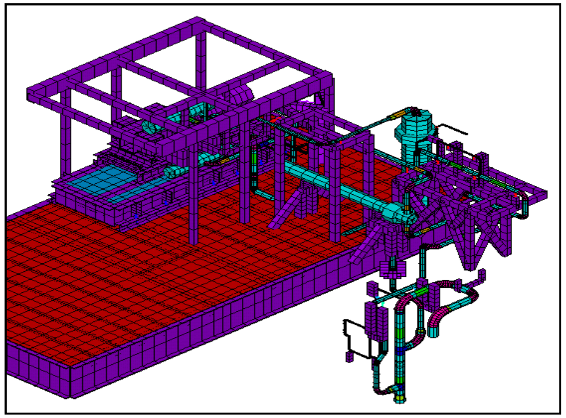


Mode shape at 23.14 Hz

<u>Challenges UGS systems</u>:

1. Large variation in pressure ratio's and flows
2. Many unloading conditions:
   - HE unloaders, stepless flow reverse control
   - Variable speed

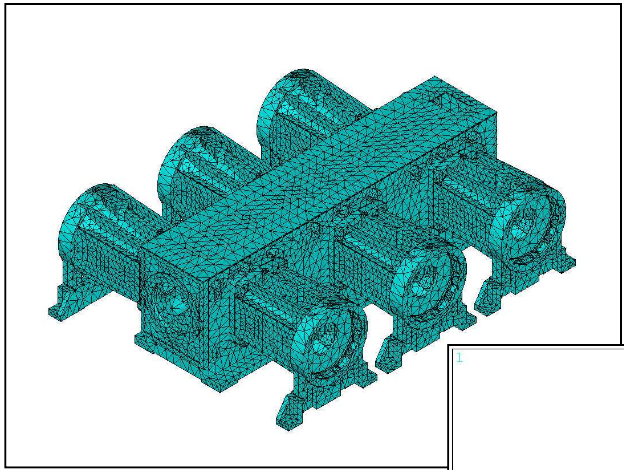
Some examples of off-shore models



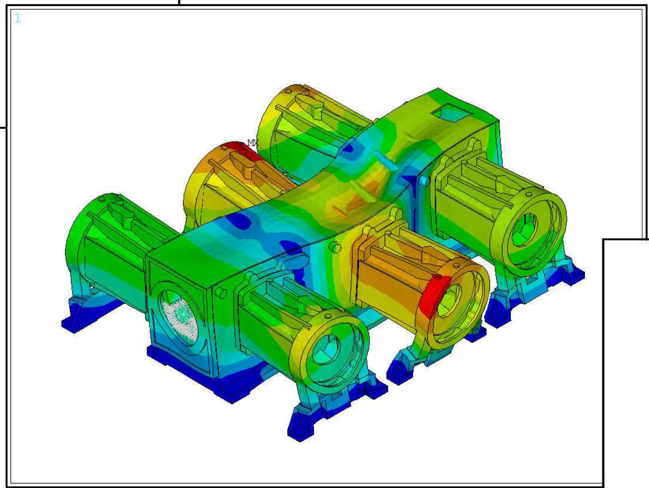
<u>Challenges in offshore systems:</u>

- Noise limitations in living quarters
- Space limitations
- Dead weight limitations
- Flexible deck structure

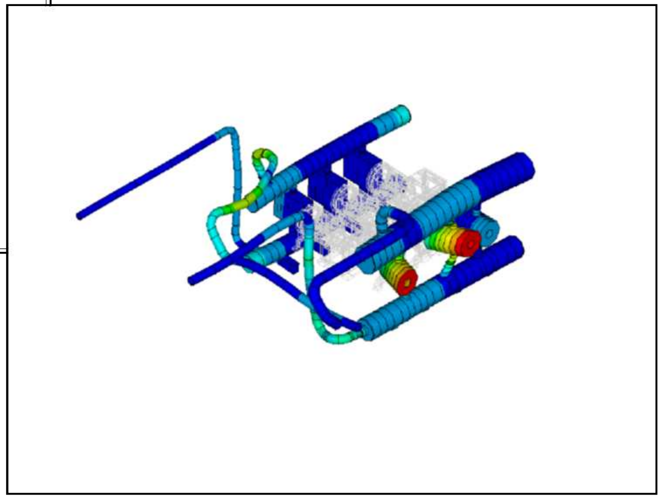
Examples of Compressor Finite Element models



Mode shape at 156.7 Hz



Mode shape at 94.7 Hz

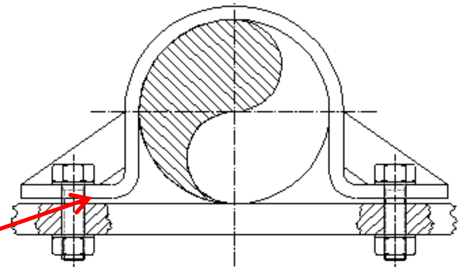


Mitigation Vibrations & Cyclic Stress Levels

- <u>Shifting resonances</u>:
  - frequencies of excitation forces should not coincide with mechanical natural frequencies

$$\omega_0 = \sqrt{k / m}$$

  - Shifting to higher values by:
    - Additional pipe supports
    - Stiffer pipe support structures
    - Difficult to achieve for variable speed compressor

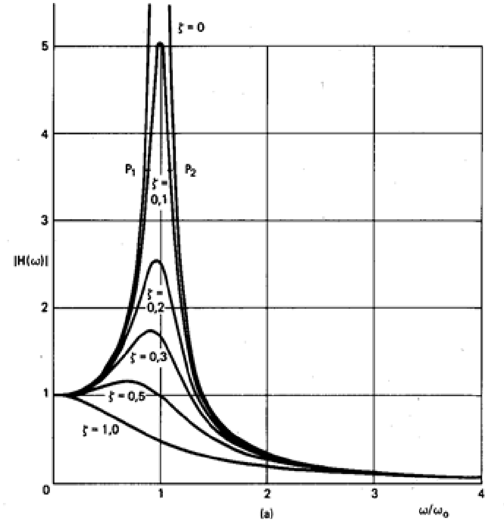


  - Shifting to lower values by:
    - Increasing mass

Mitigation Vibrations & Cyclic Stress Levels

- Damping:
  - Most effective in resonance conditions
  - Viscous dampers

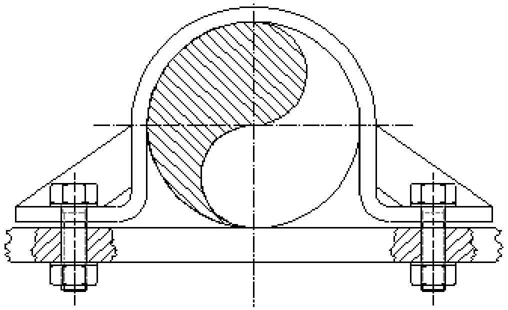
Pipework Damper



- Lower cyclic stress:
  - Increase wall thickness
  - Add braces

Pipe Supports

- Rigid clamps are required for systems which are subjected to vibrations
- Disadvantage of additional pipe clamps:
  - possibility of too high pipe stress caused by thermal expansion

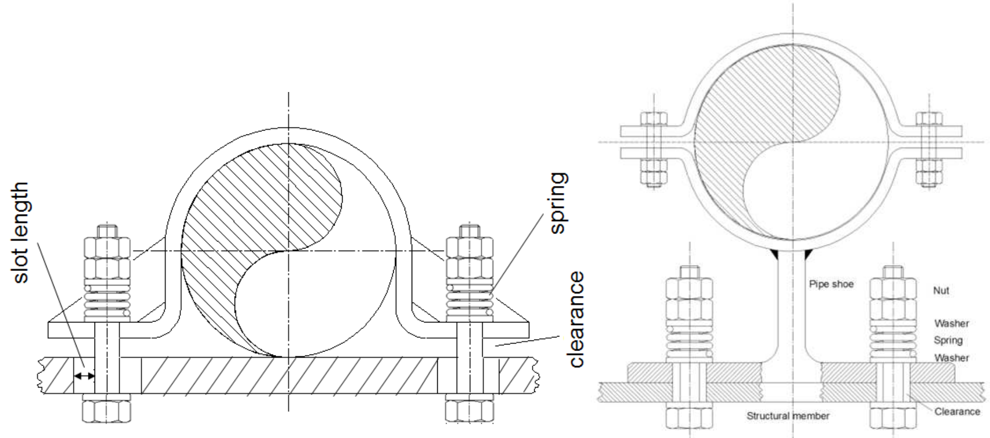


Example of a rigid clamp

- Solution:
  - spring hold down supports

- Required spring preload:

$$Fn \geq \frac{Fw}{f} \quad [N]$$



Examples of spring hold down supports

F_w= pulsation-induced reaction force

f = friction coefficient (0.3 steel-steel; 0.1 steel-teflon)

API 618 Standard

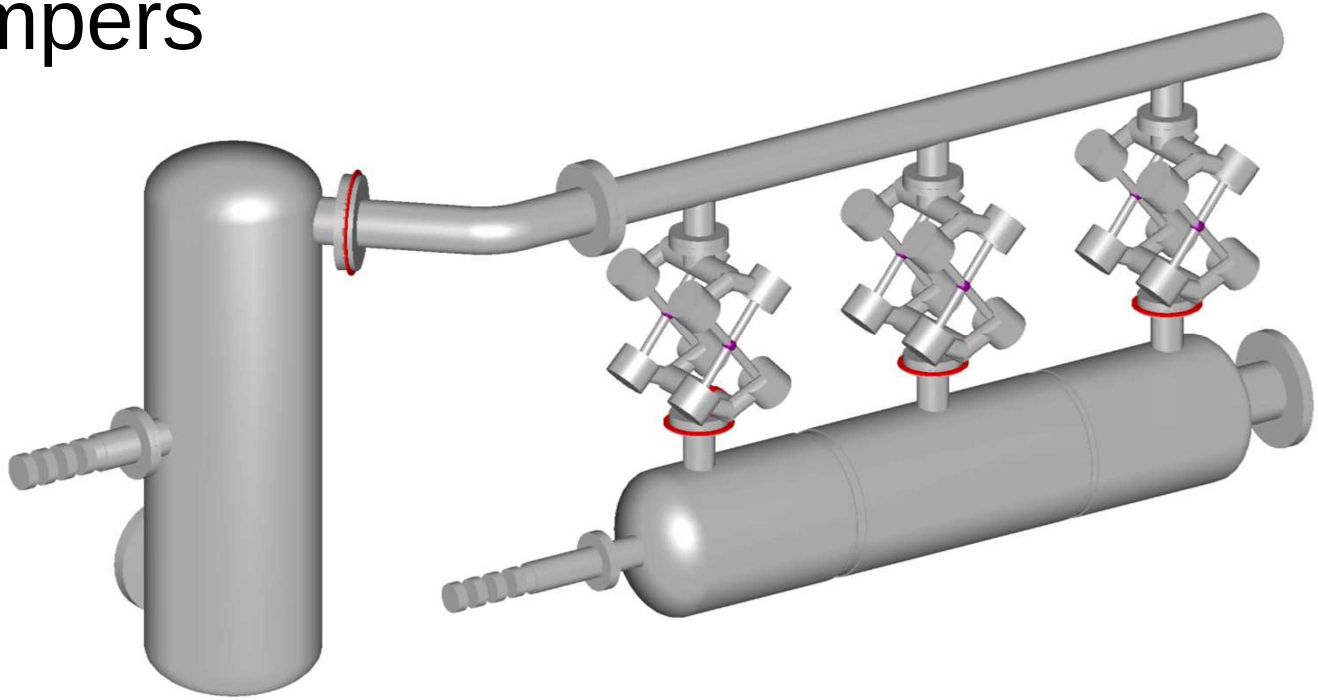
- Specifies allowable pulsation, vibration & cyclic stress levels
- Stipulates a design approach

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)	[illegible]	[illegible]	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

High power, high pressure

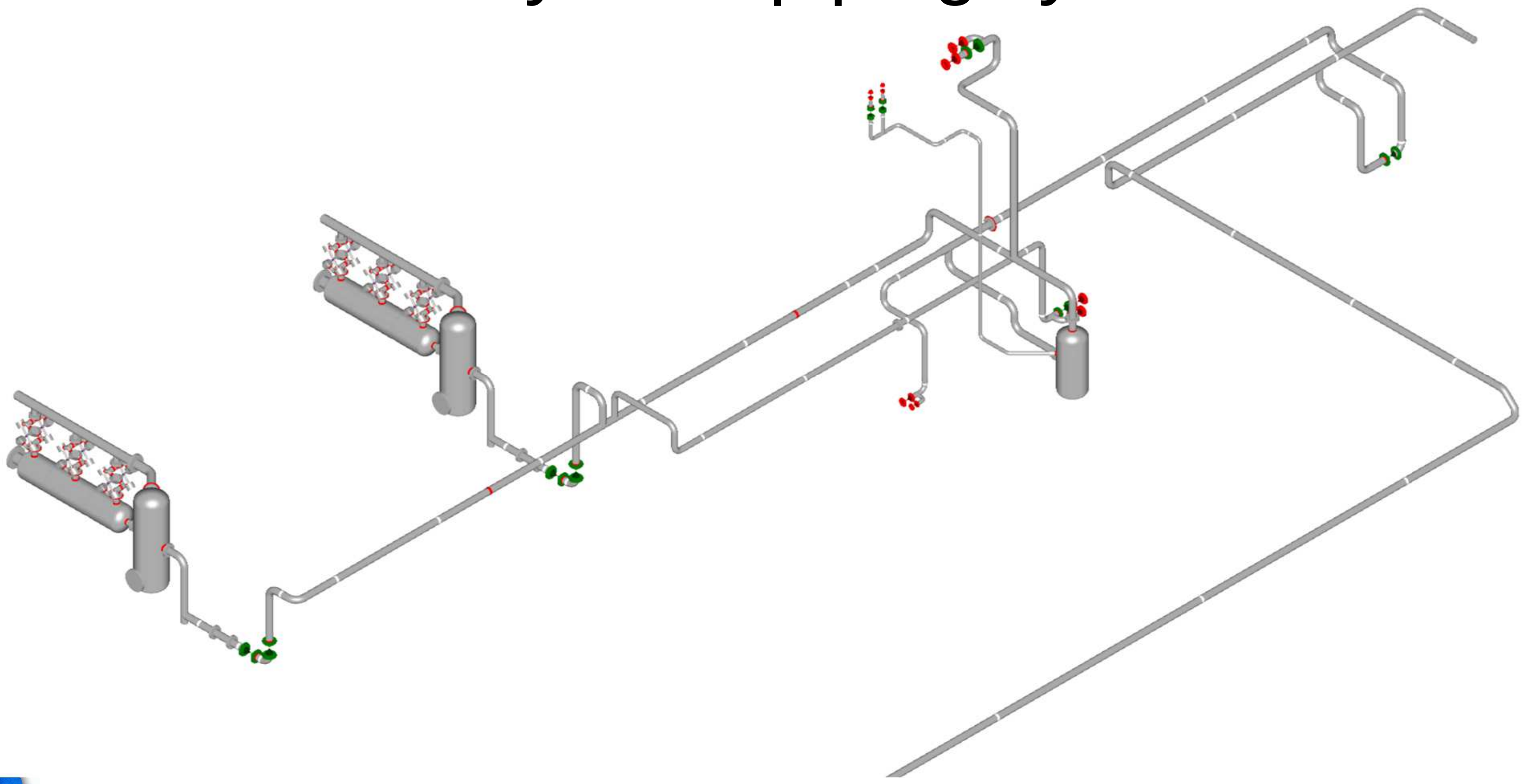
Design approach 3

1) Acoustic evaluation & design of pulsation dampers



Design approach 3

2) Pulsation analysis of piping system



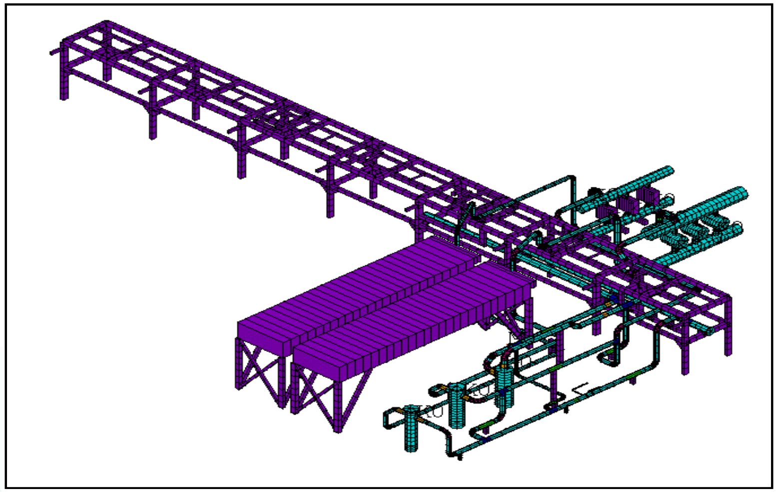
Design approach 3

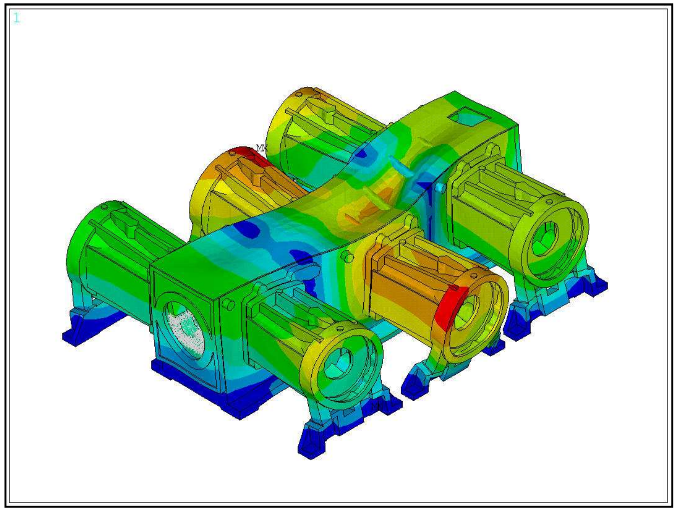
3) Forced mechanical response analysis

Vibrations and cyclic stresses of:

– Piping

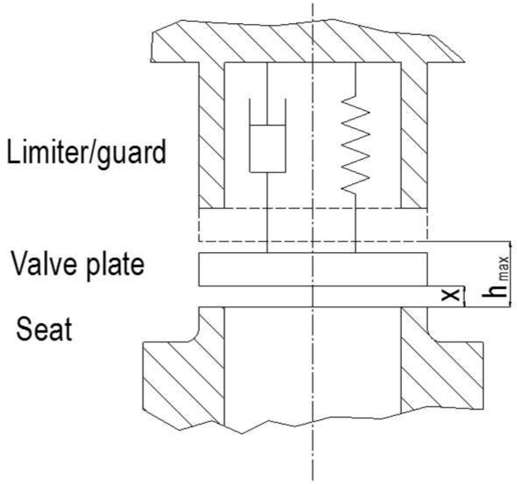
– Compressor manifold

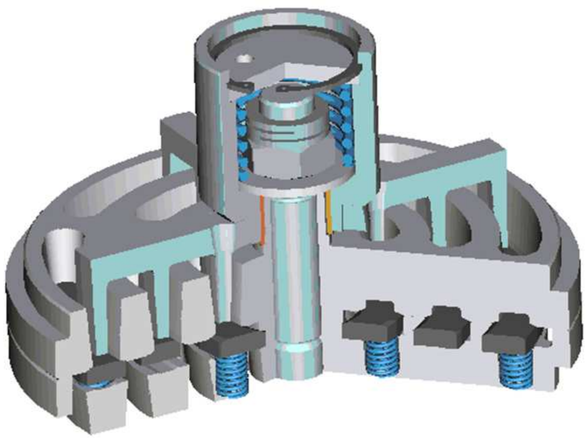


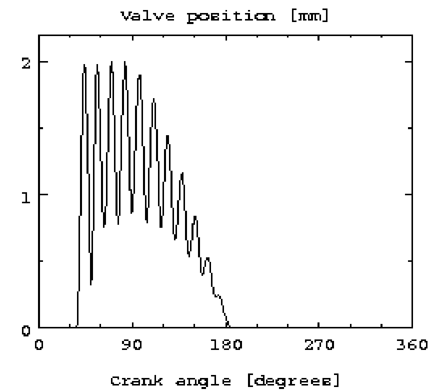


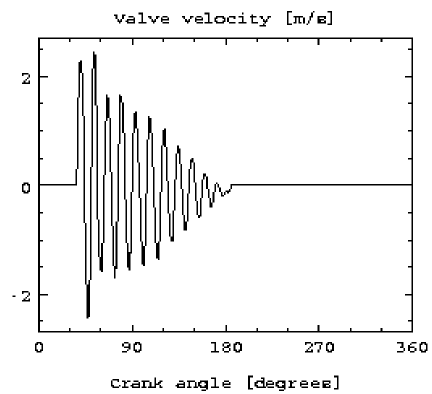
Design approach 3

4) Analysis of compressor valve dynamic behaviour









Thank you for your attention!

Questions ?

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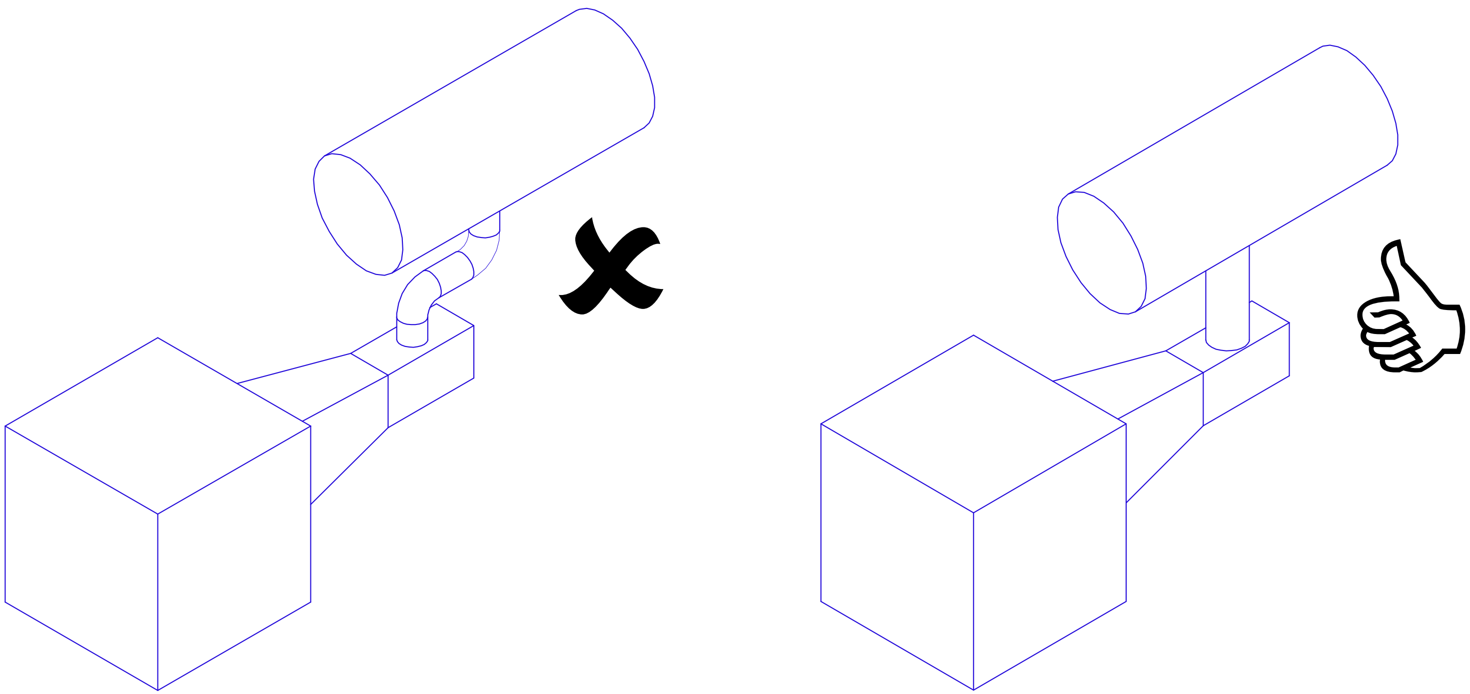
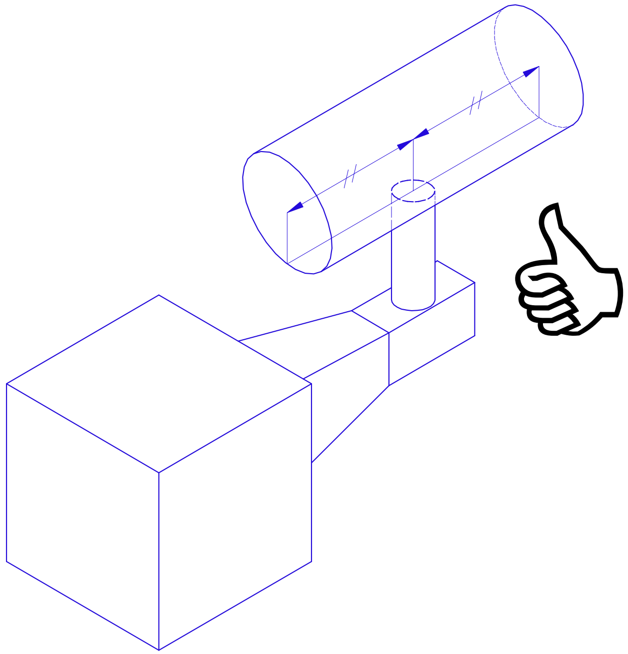
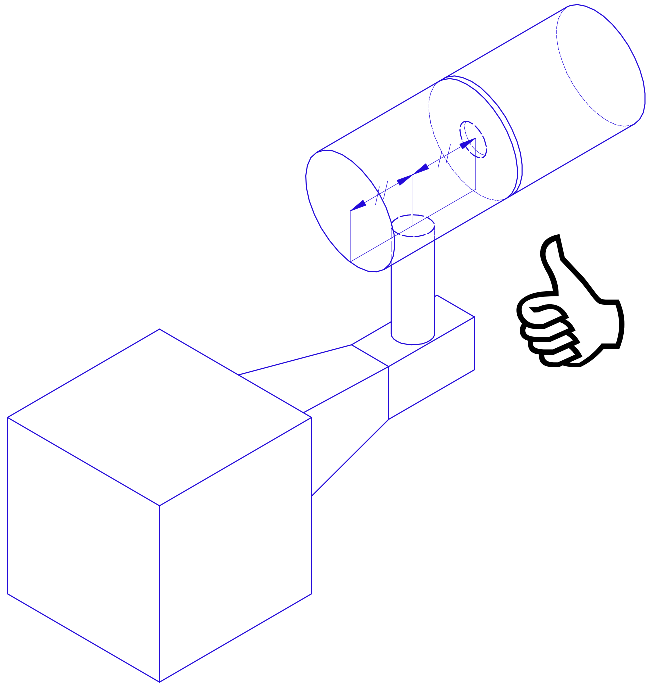
Encore: basic design rules (do's and dont's)

Keep cylinder connection of pulsation damper as short as possible

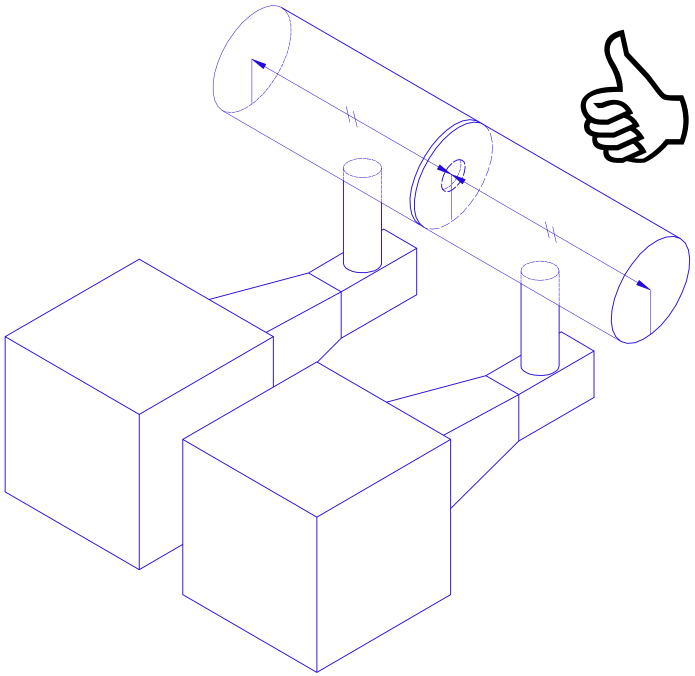


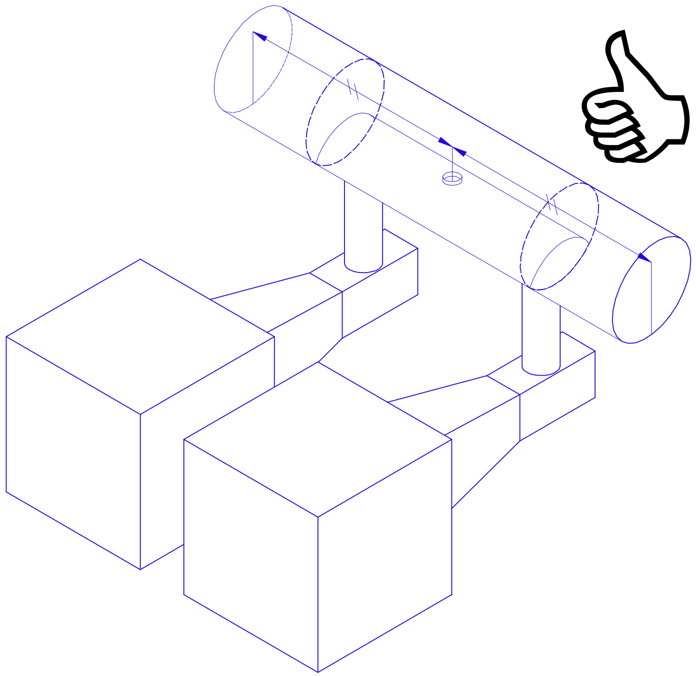
Symmetric layout of cylinder connection in the damper compartment



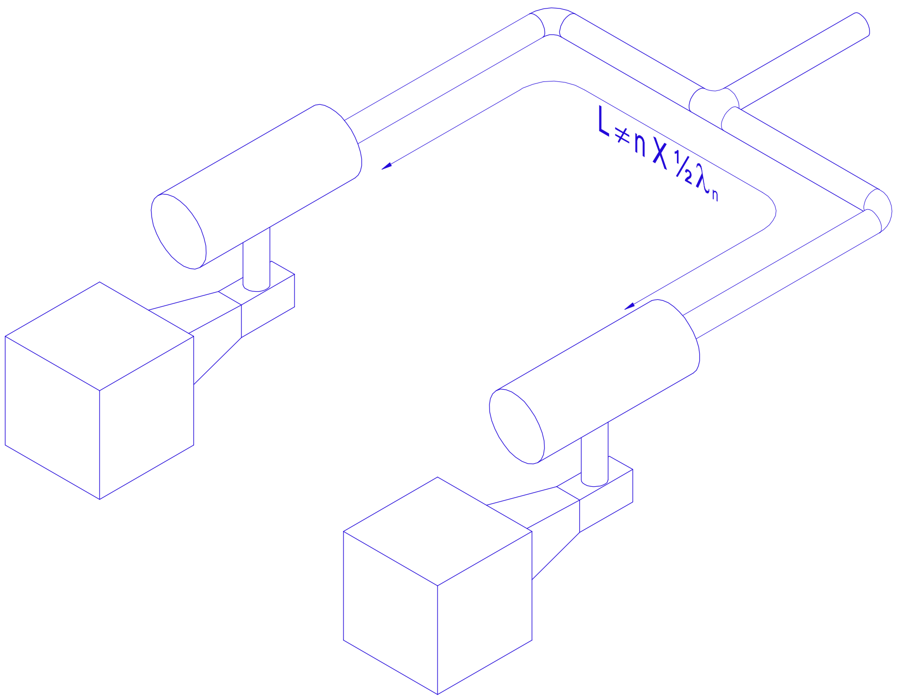


Manifold damper:
Baffle plate or extended cylinder connection

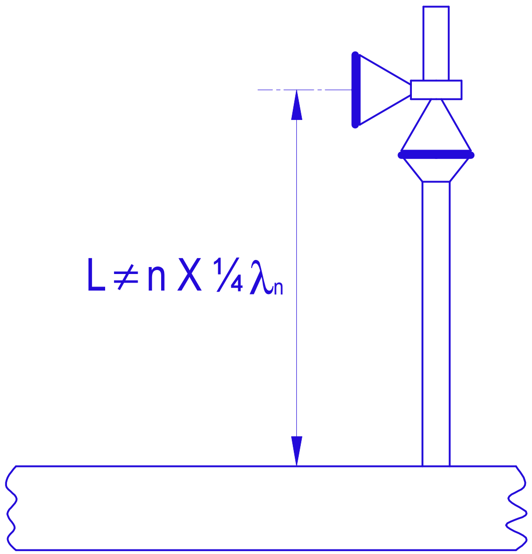




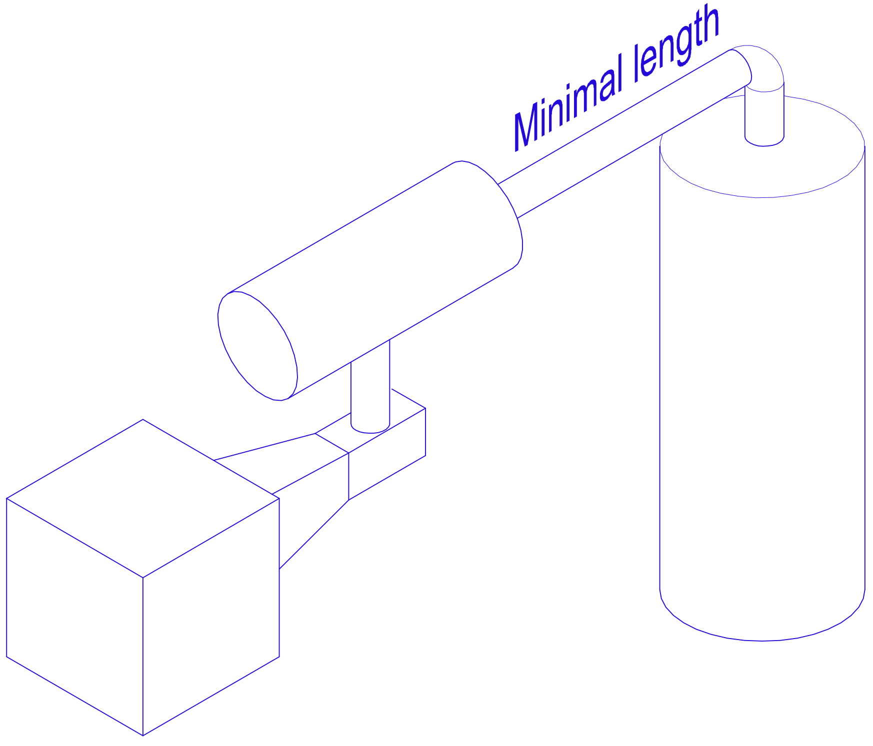
Avoid standing waves and Helmholtz resonance between vessels



Avoid standing waves in closed side branches

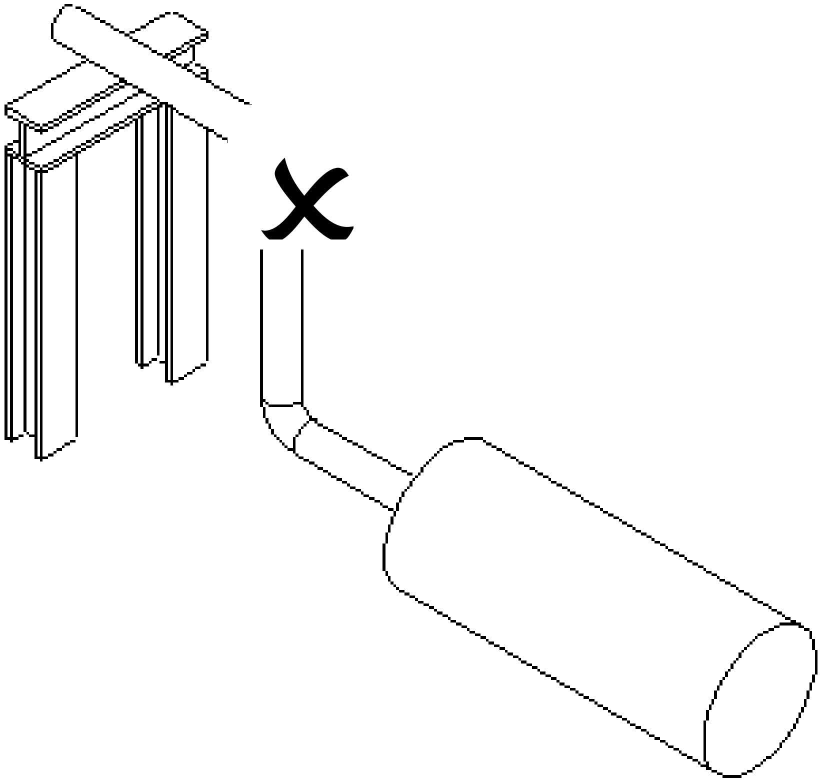


Minimize length between large vessel and damper

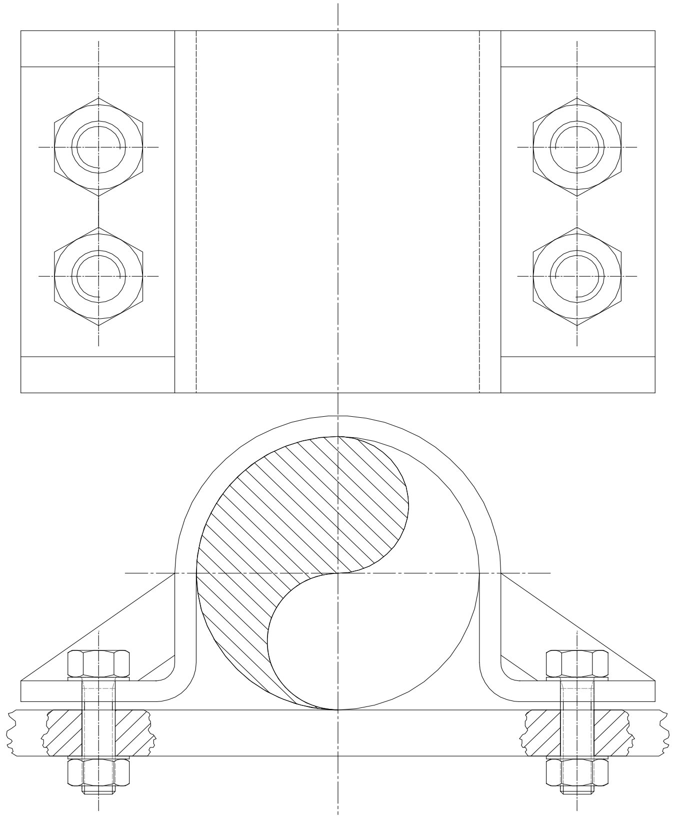


Piping layout, clamps

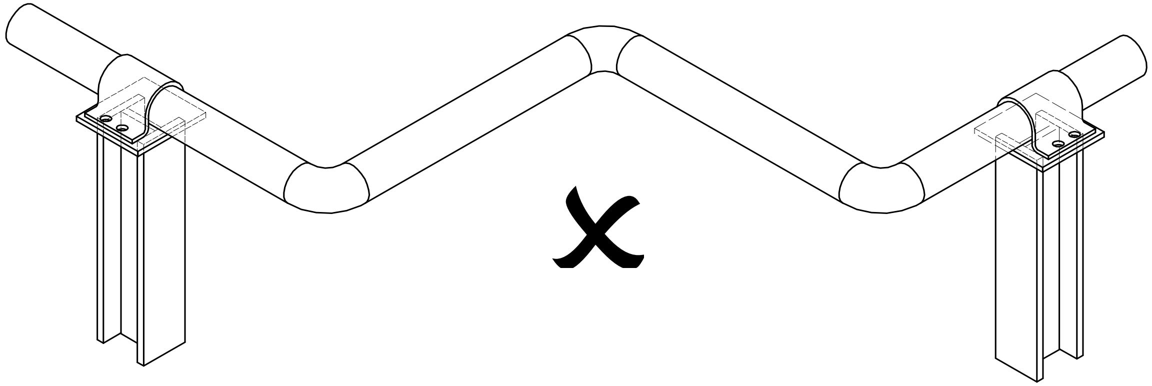
Keep all piping low



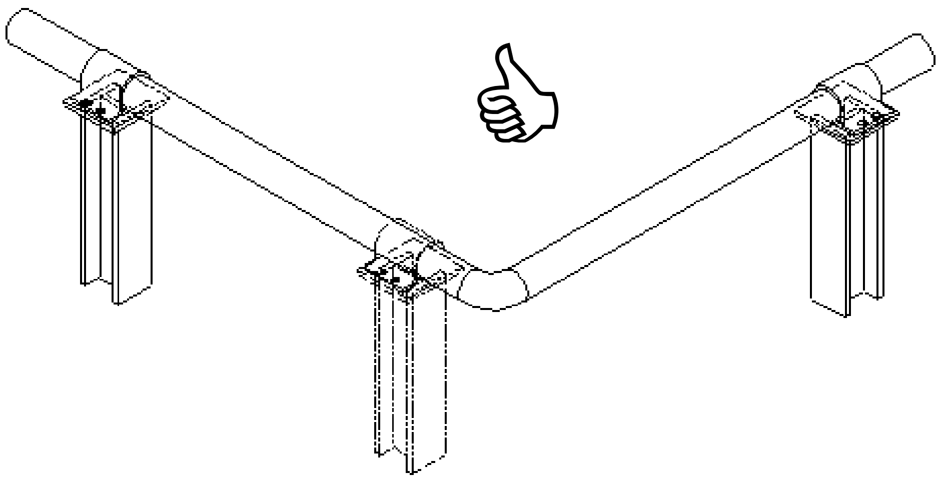
Good Clamps have width
(do not use U-bolts)



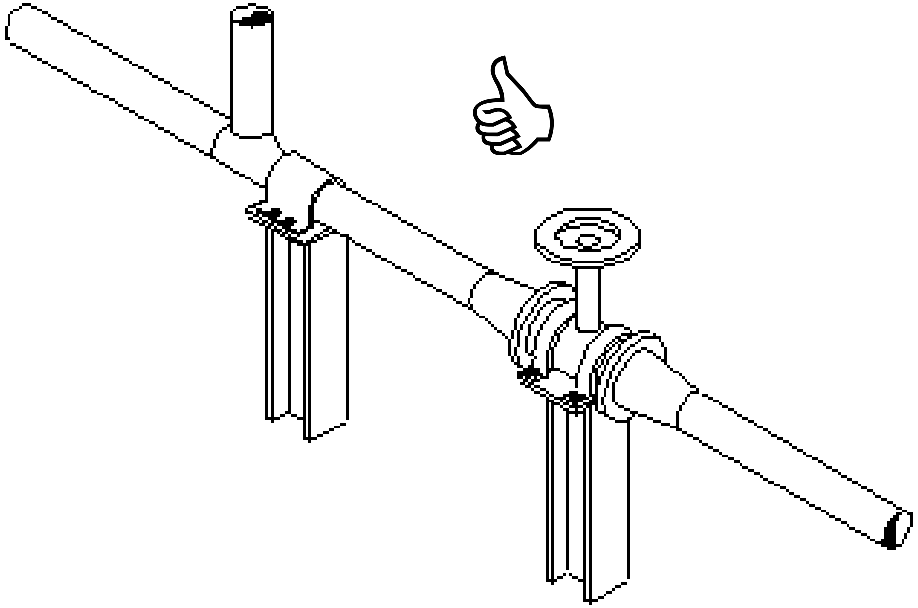
Do’s, Don’ts Design Rules

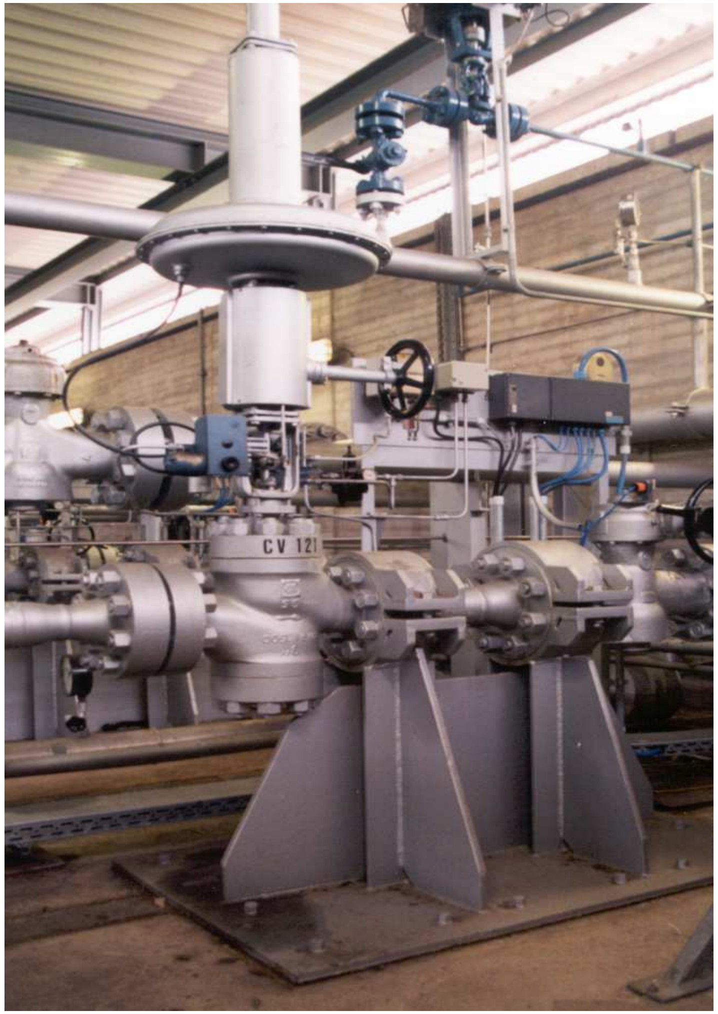


Minimise number of elbows and Tees
and avoid freestanding elbows



Do's, Don'ts Design Rules

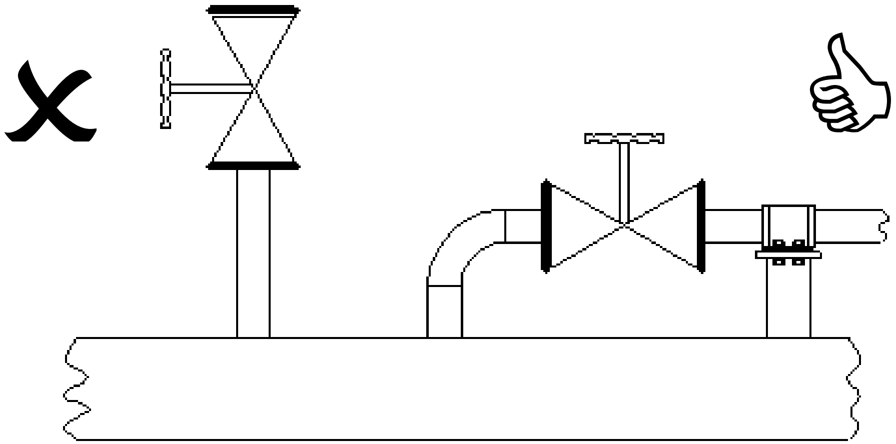


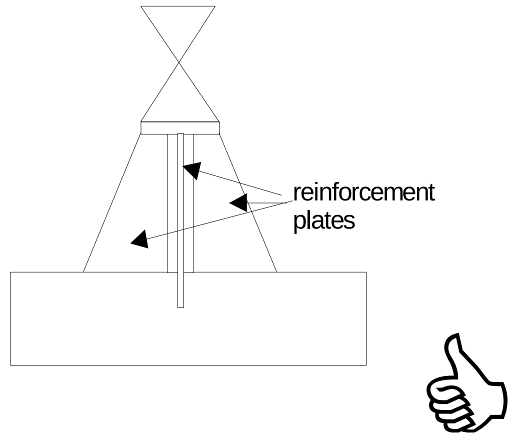


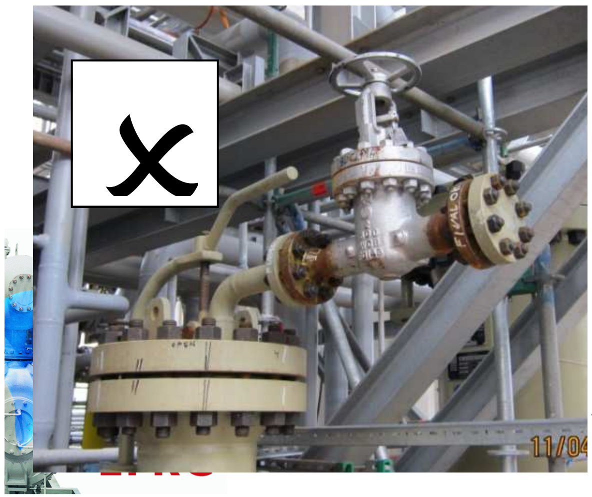
Locate supporting directly under heavy components (valves, flanges etc.) and Tee joints

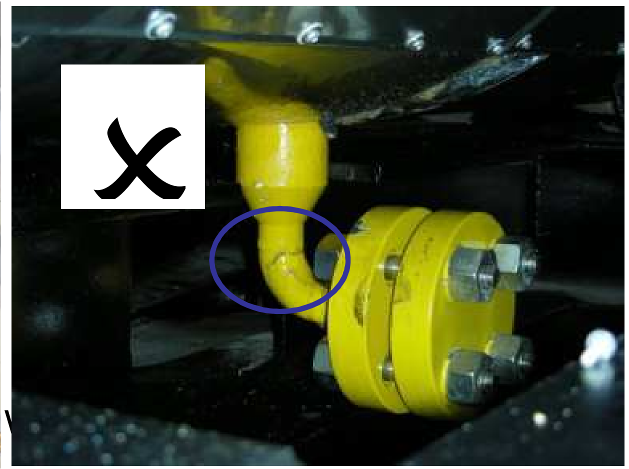
Do's, Don'ts Design Rules

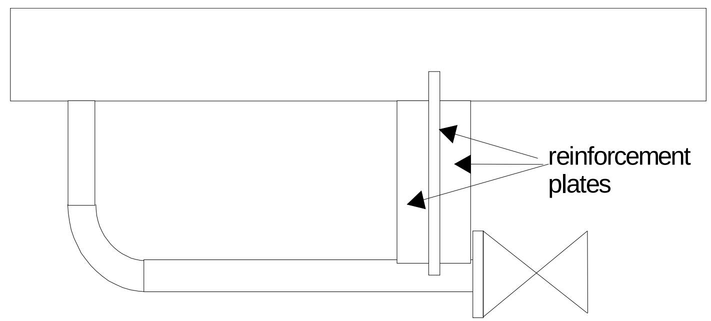
Brace small lines (drains, purge lines)
Do not use unreinforced branch connections
Apply two-plane bracing of small bore side branches back to main pipe





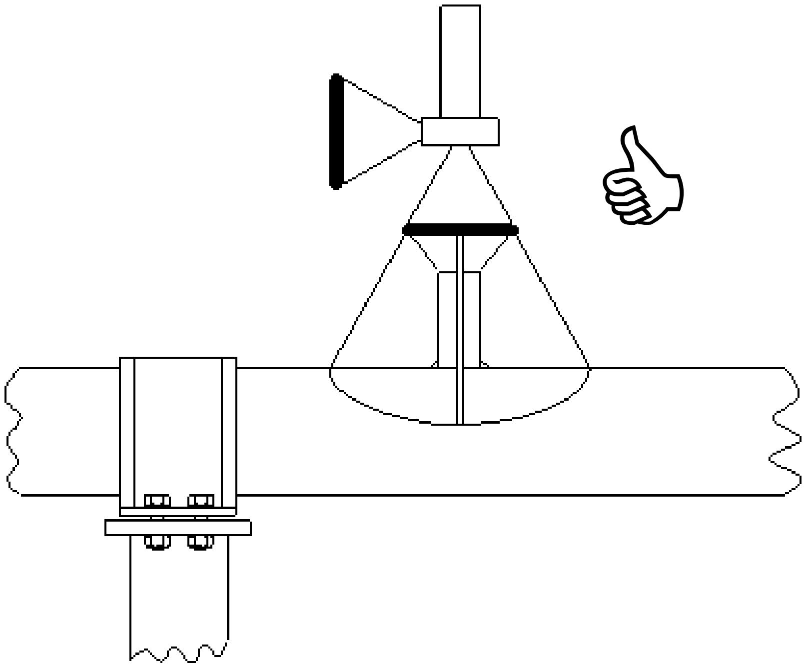


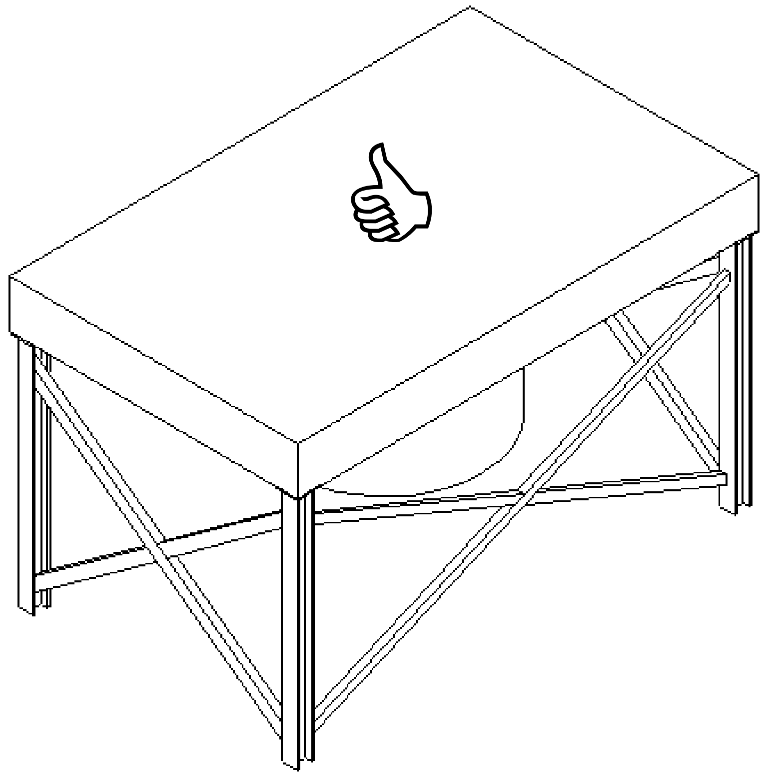




Do's, Don'ts Design Rules

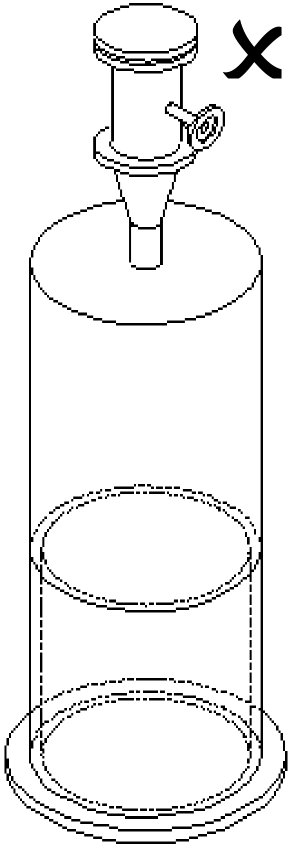
Keep relief valves close to main line – Apply X-bracing in air cooler frames

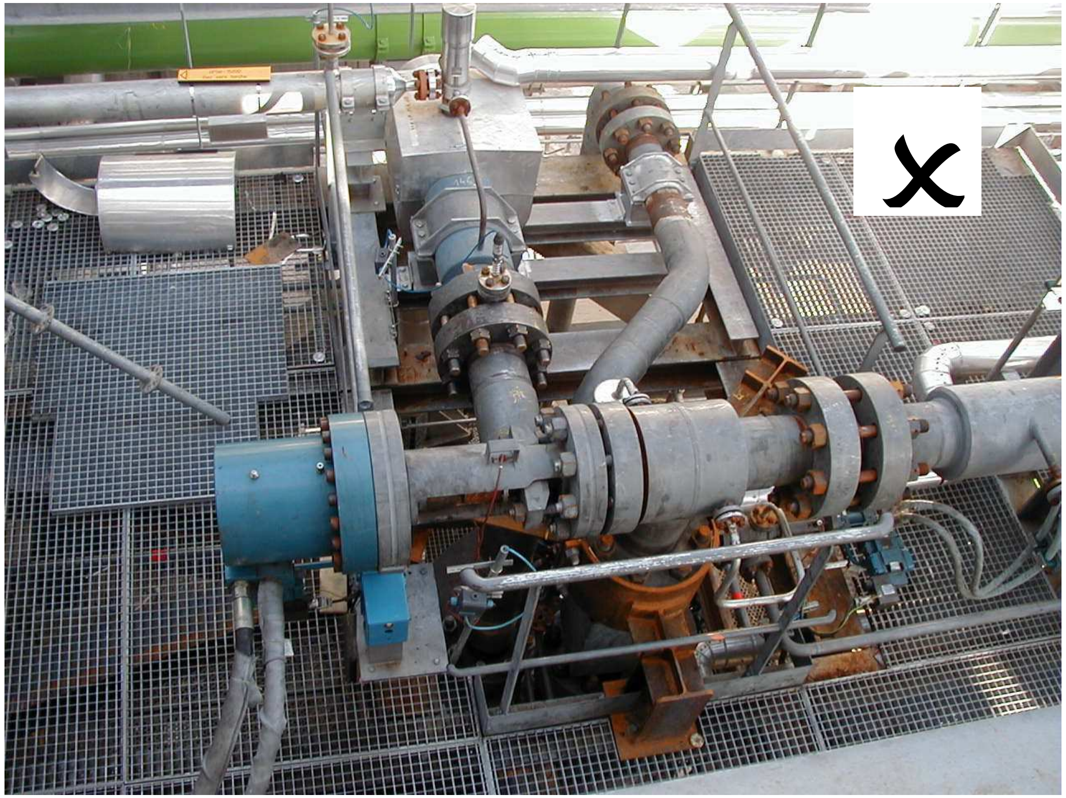




Do's, Don'ts Design Rules

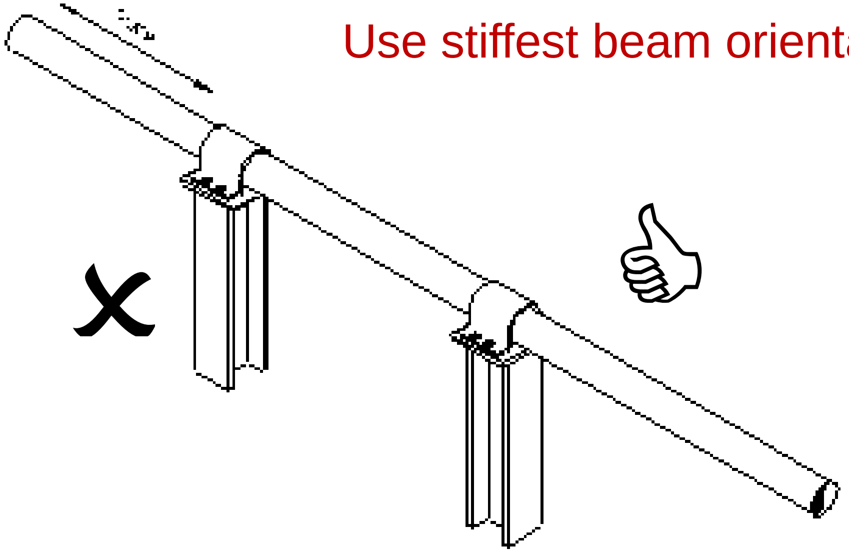
Avoid heavy valves at high elevation (also on top of separators)

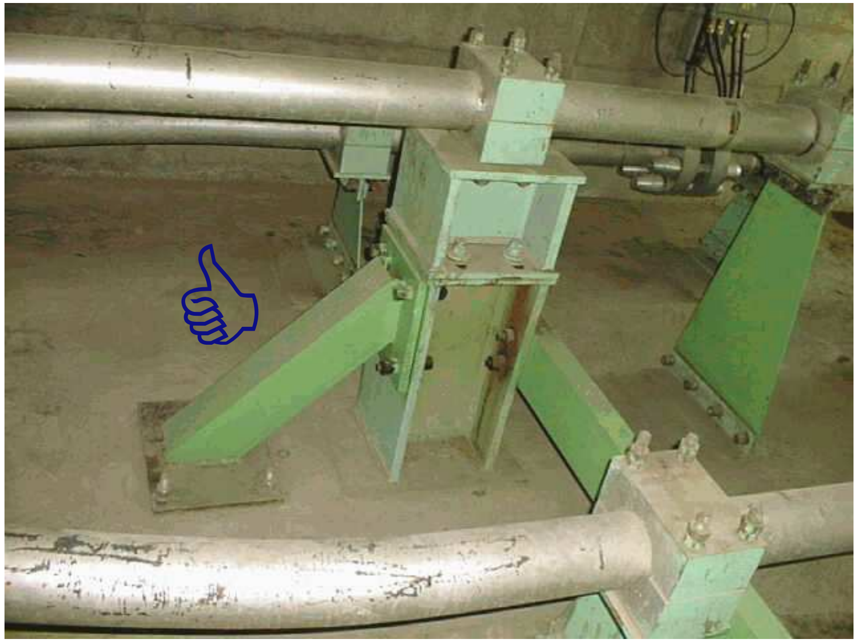




Do's, Don'ts Design Rules

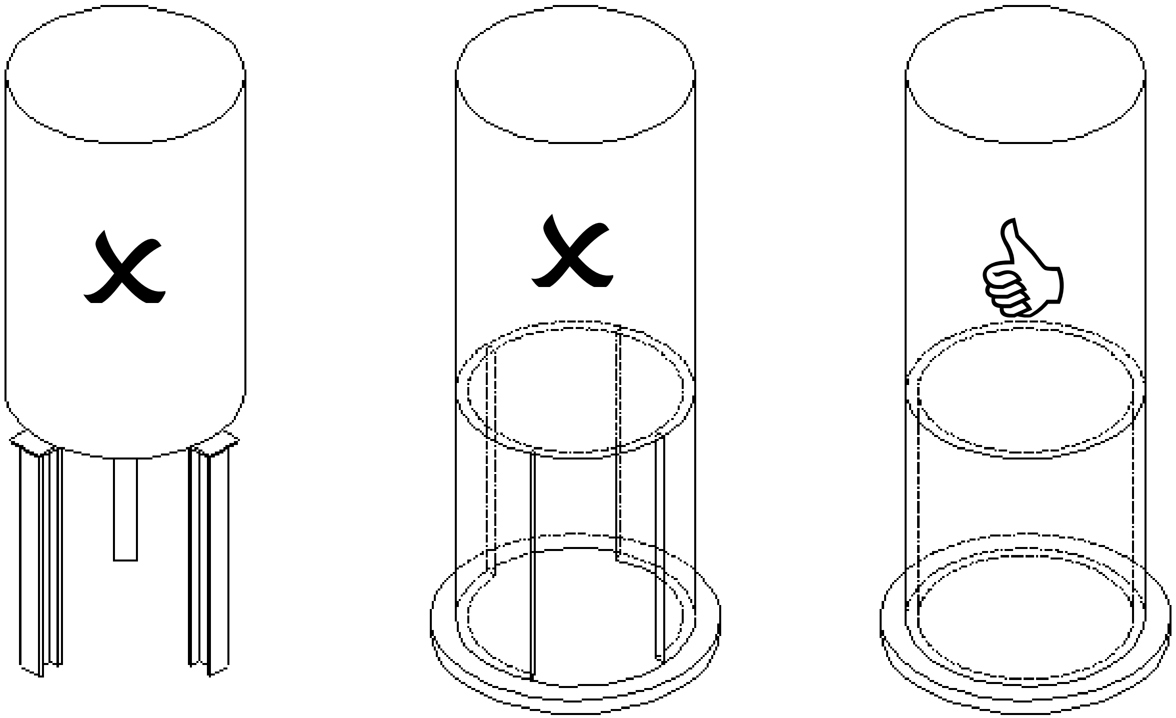
Use stiffest beam orientation in the force direction





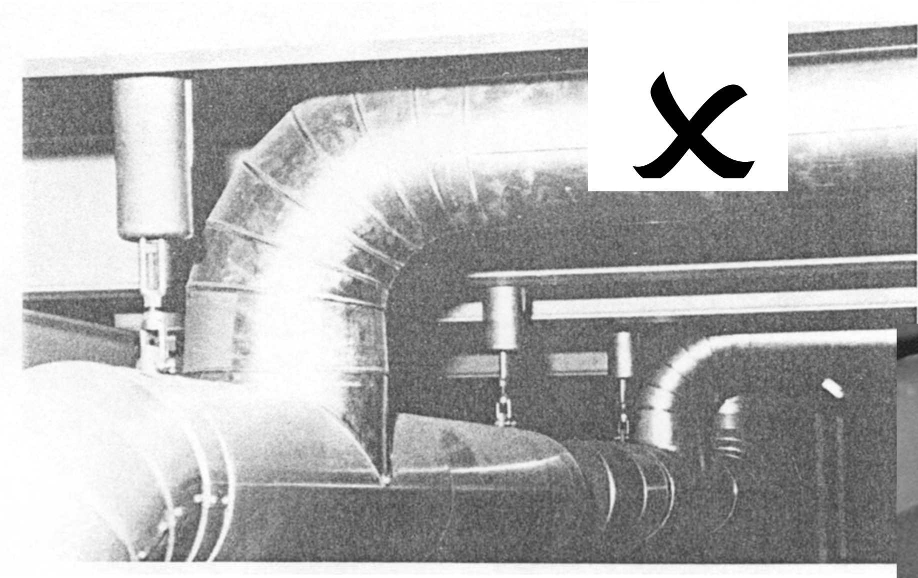
Do's, Don'ts Design Rules

Use full skirts in vessel supporting

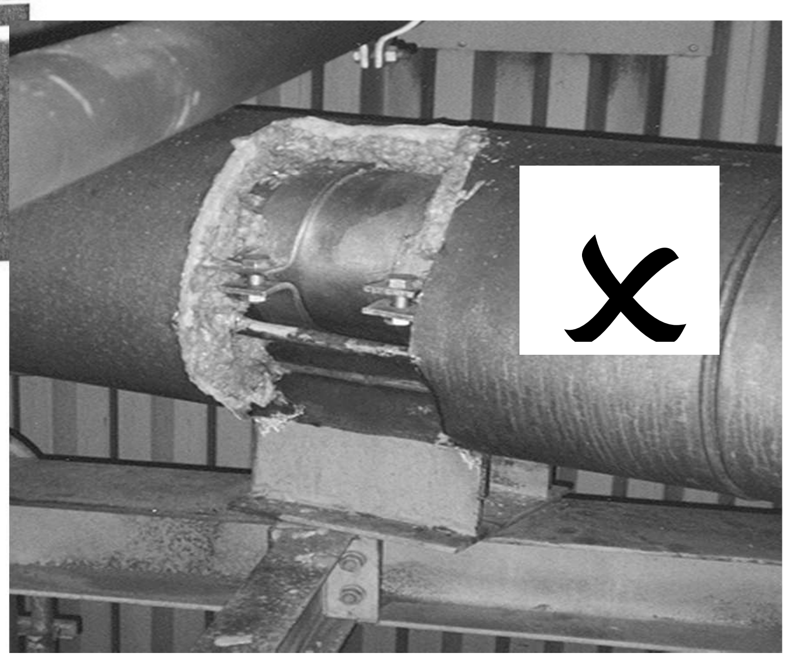


Do’s, Don’ts Design Rules

Avoid rod and constant load hangers

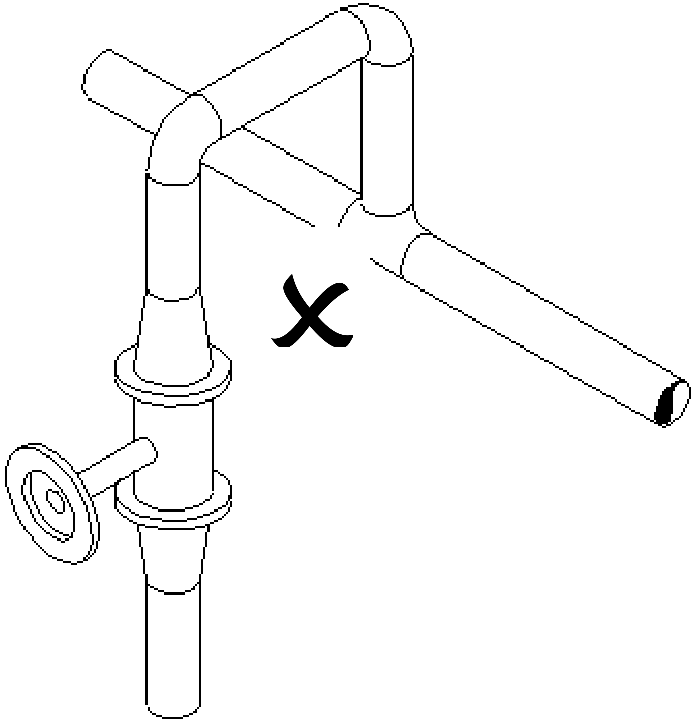


Avoid weight only supports

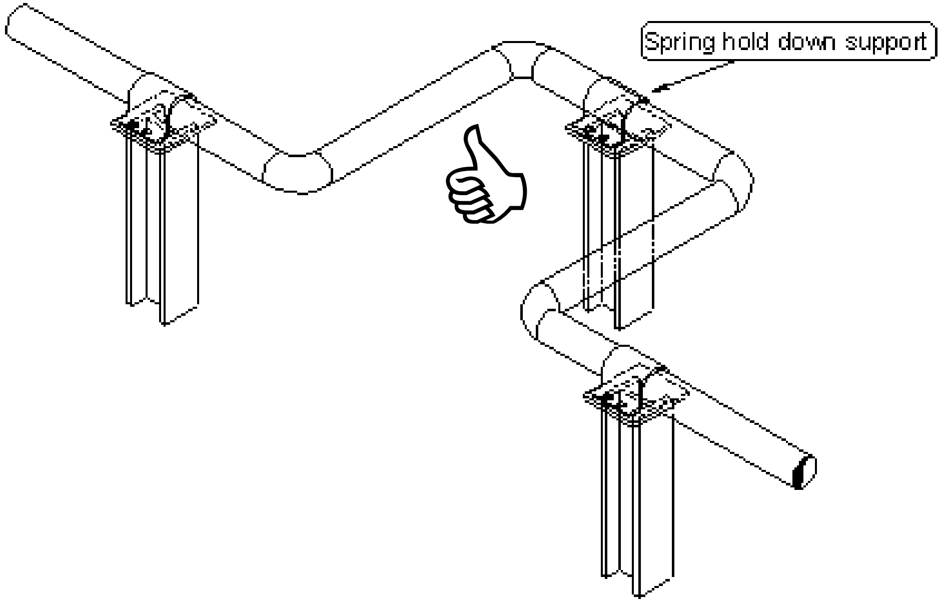


Do’s, Don’ts Design Rules

Avoid unsupported overhanging weight

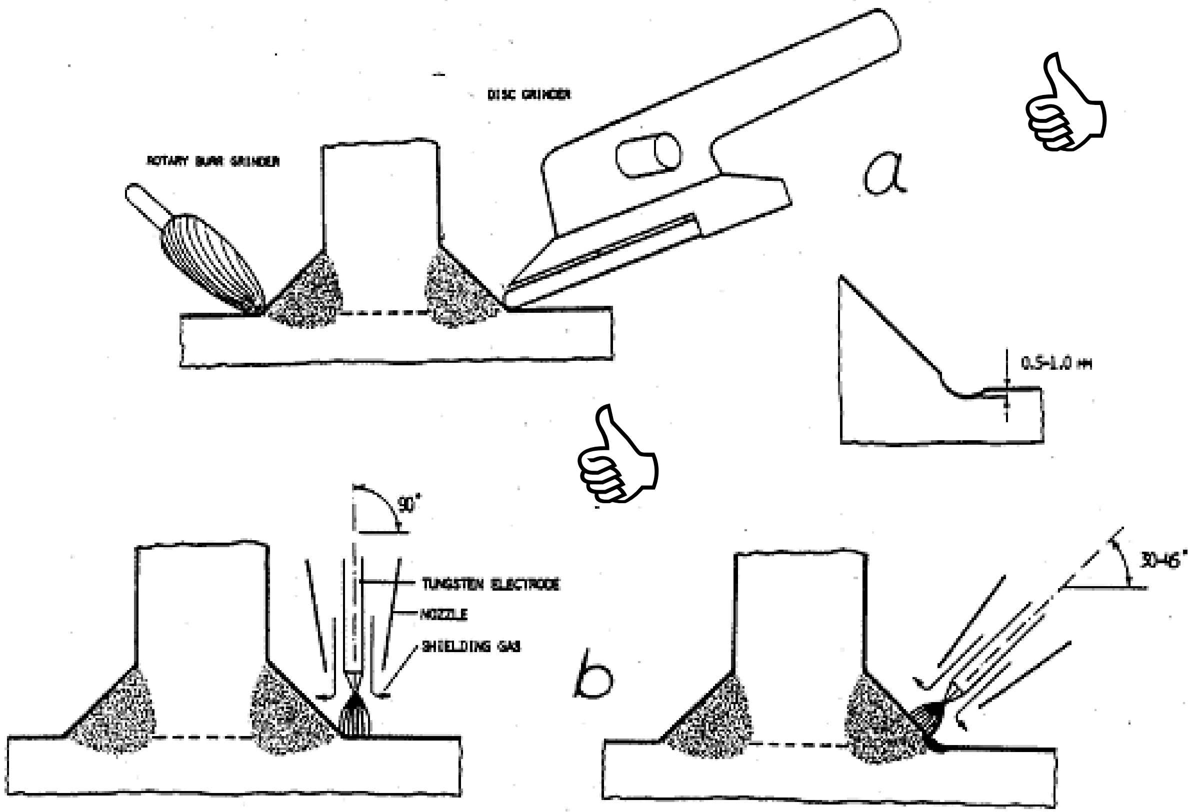


Avoid unsupported expansion loops



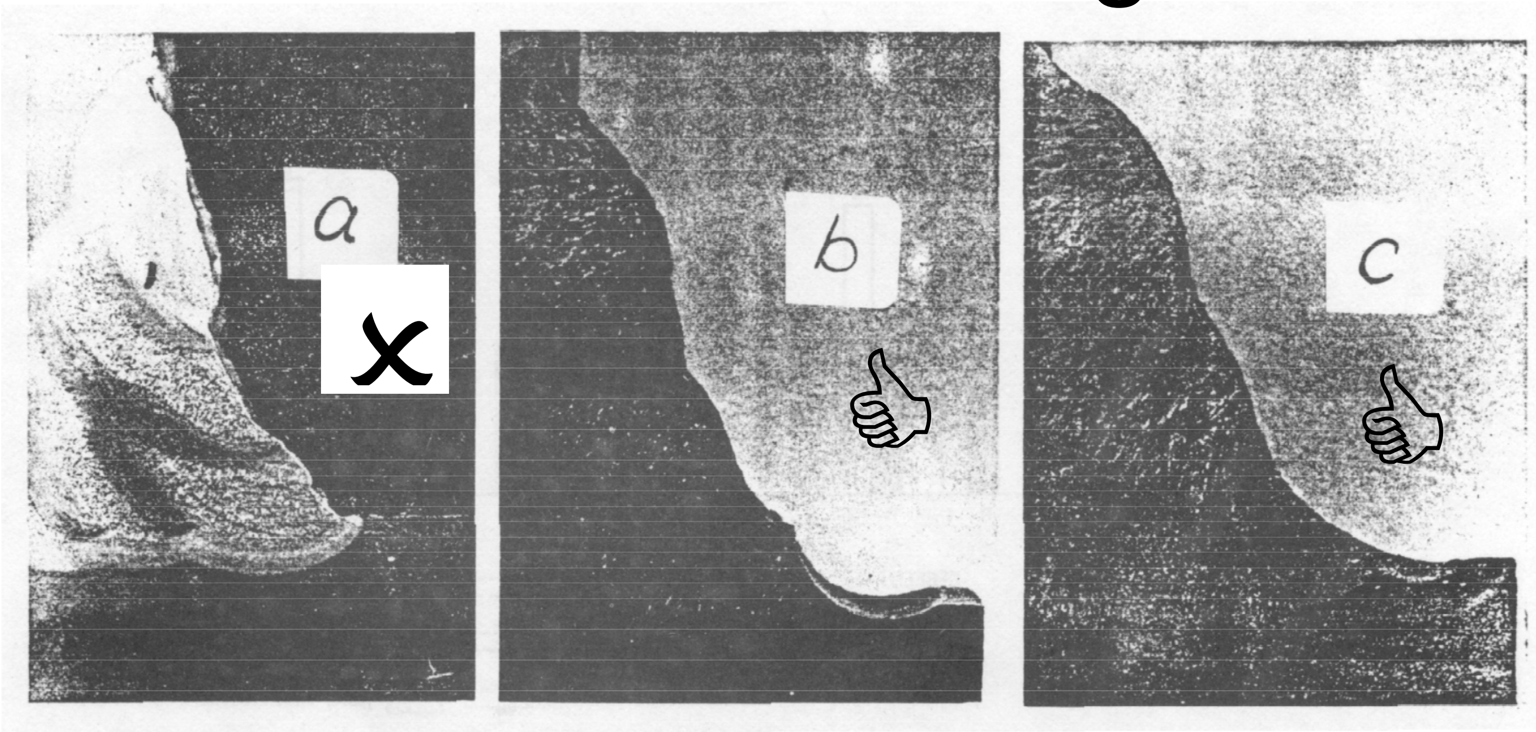
Do’s, Don’ts Design Rules

Welds:
a) avoid weld imperfections
b) apply full penetration welds
c) avoid sharp corners (grind welds)



a) cutting
b) dressing

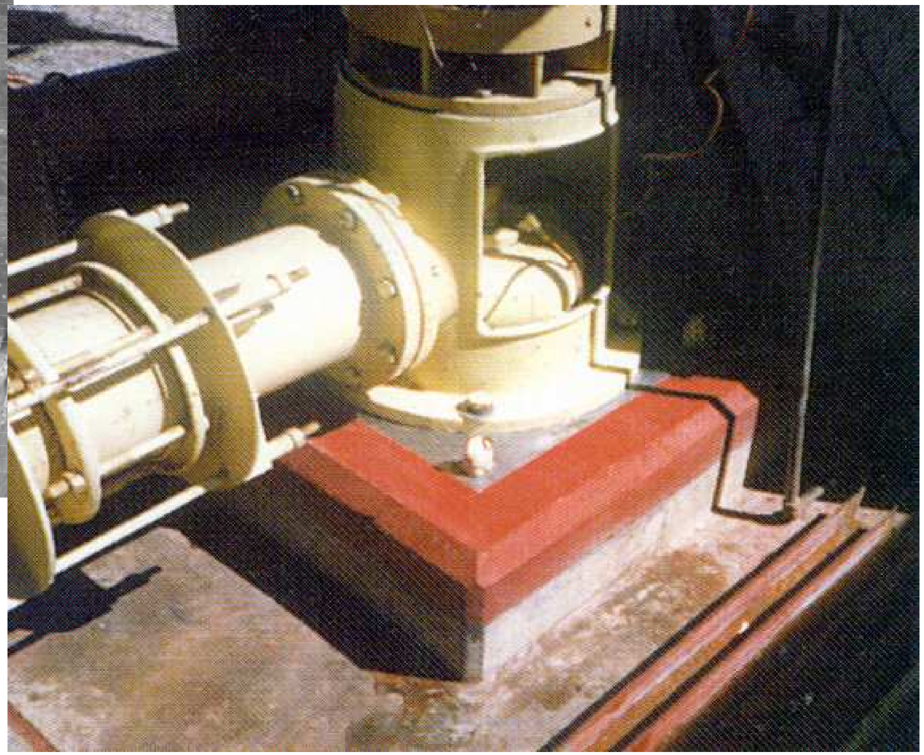
Do's, Don'ts Design Rules



a) as welded
b) after blurr grinding
c) after TIG dressing

Do's, Don'ts Design Rules

Use adequate grouting (epoxy resins)



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