Mechanical Vibrations

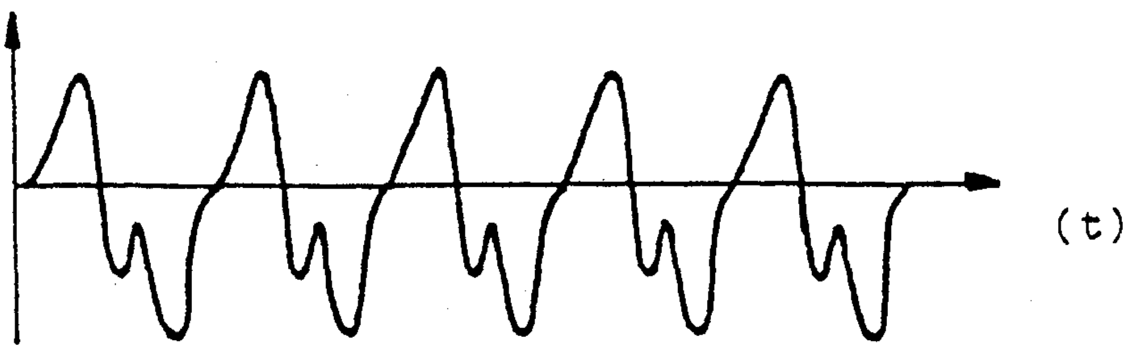
- Why a mechanical vibration analysis?
  - Too high vibration and cyclic stress (fatigue) can occur, even in case pulsation levels are within the allowable levels: mechanical resonances
- What is a vibration?
  - A vibration is a more or less regular movement of a body as a function of *time*
- Examples:
  - 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)

EFRC

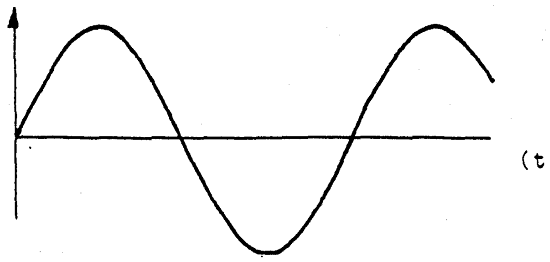
Vibration Forms

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

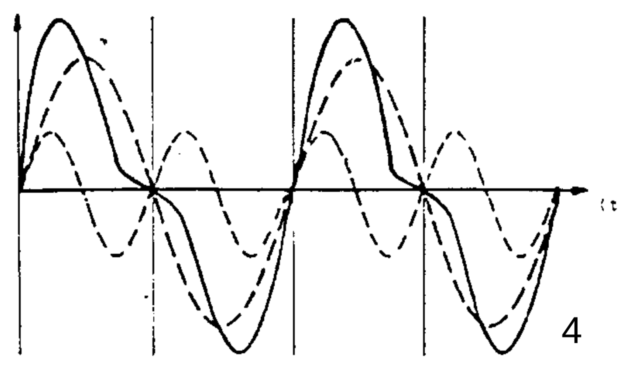
- Periodic vibration:
  - Vibration whereby the amplitude is repeated after a discrete time period
- Examples:
  - Combustion forces in a cylinder
  - Unbalanced loads of reciprocating compressors
  - Pulsation-induced shaking forces



- 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

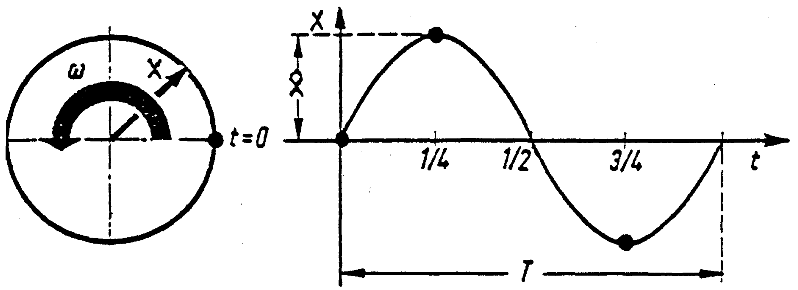


- Two or more summed harmonic vibrations with different frequencies:
  - periodic vibration but not harmonic



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 (ω):
  - $\omega = 2\pi f$

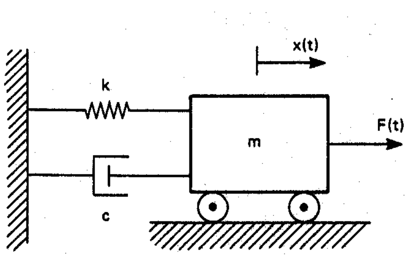


Theory of Mechanical Vibrations

- General equation of motion:

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

Pulsation forces and mechanical loads



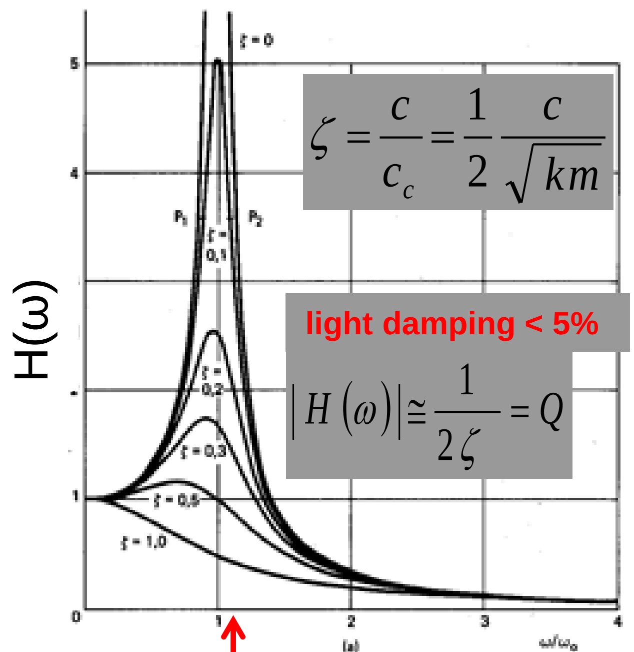
m= mass
c= damping
k= stiffness

$\ddot{x}$ = acceleration
$\dot{x}$ = velocity
x = displacement

- Resonance condition:

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

ω_0 = mechanical natural frequency
ω = excitation frequency ($\omega = 2\pi f$)
$\omega = \omega_0$ at resonance

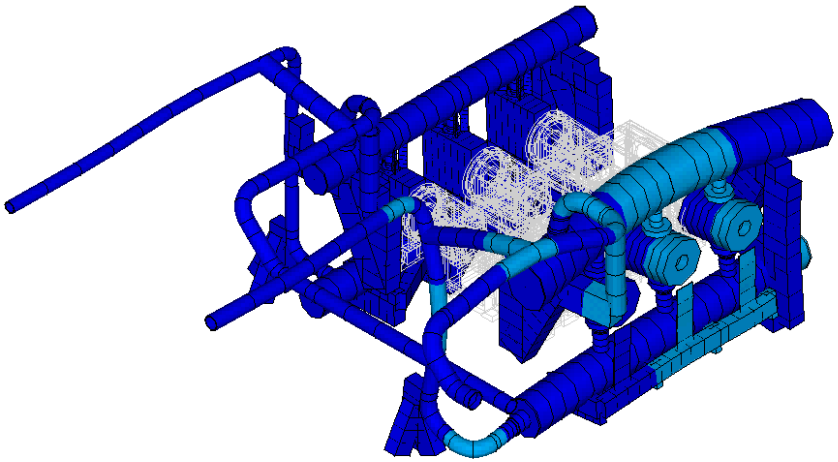


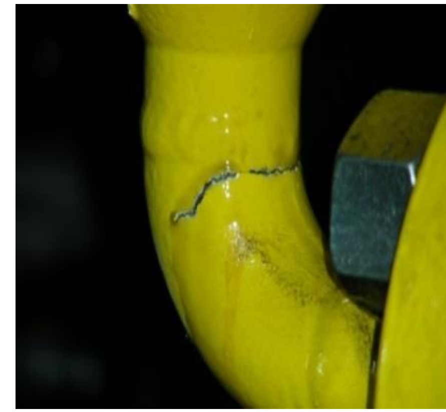
|H (ω)| as a function of ω/ω_0

Resonance

Theory of Mechanical Vibrations

Mode shape at resonance





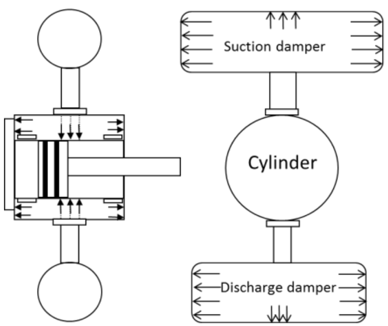
Fatigue failure

Remark:

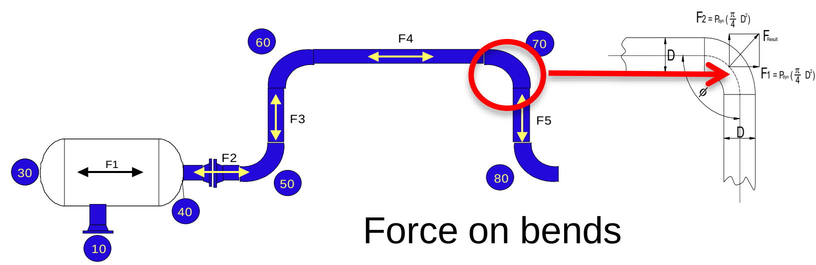
- In many cases the highest vibration levels occur at resonance conditions.
- However, high vibration levels may also be generated by high pulsation-induced shaking forces not occurring at resonance conditions.
- The distinction between both is difficult to determine. In many cases additional supports may only decrease the vibrations but will introduce high local stresses.

Dynamic Forces generated by Reciprocating Compressors

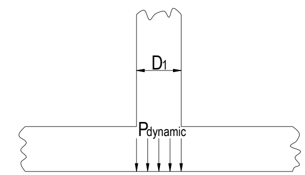
- Pulsation-induced shaking forces (pulsation dampers, cylinders, pipe system, separators, coolers etc.)



Forces on dampers and cylinder

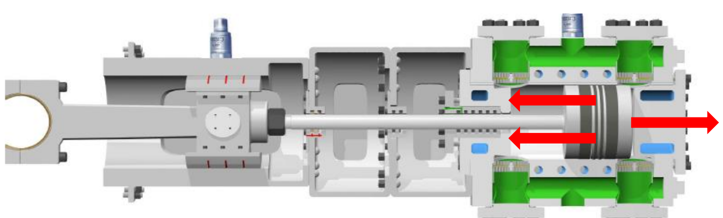


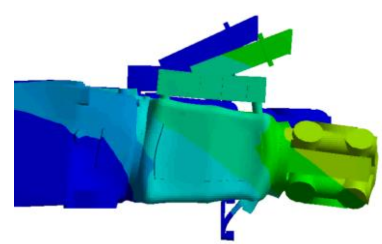
Force on bends



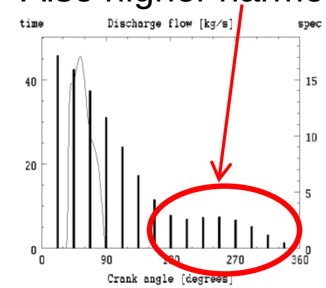
Force on Tee

- Cylinder gas stretching forces
  - MNF of suction dampers and suction piping can be excited
  - Not only forces of 1X but also higher harmonics are important





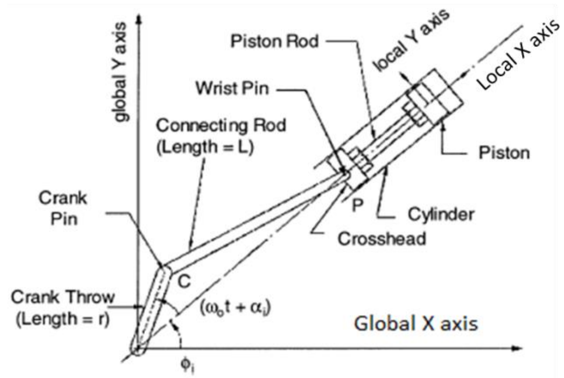
Also higher harmonics



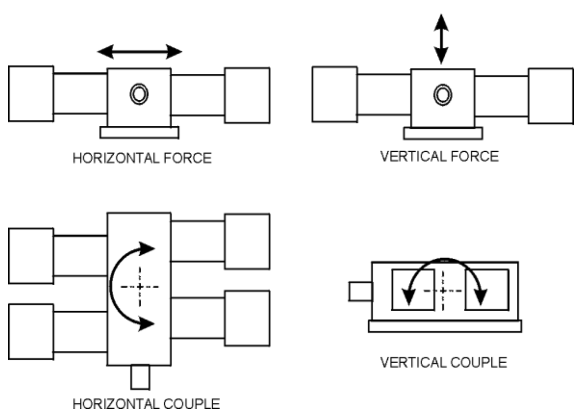
Spectrum of flow pulse

Dynamic Forces generated by Reciprocating Compressors

- Free forces & moments caused by rotating and translating parts:



Working principle of a crank mechanism



Summary of free forces & moments for multiple cylinders

Inertia force in piston direction:

$$F_x = (m_{rec} + m_{rot}) r\omega_0^2 \cos\omega_0 t + m_{rec}\frac{r^2}{L}\omega^2 \cos 2\omega_0 t$$

1X 2X

Inertia force perpendicular to piston

$$Fy = m_{rot} r\omega_0^2 \sin\omega_0 t$$

1X

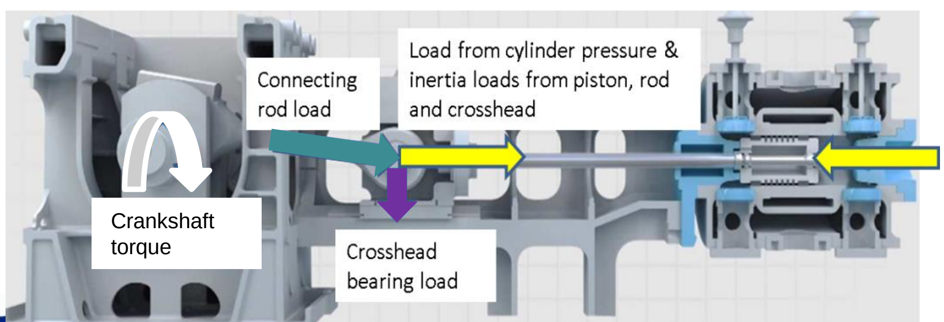
In which:

- F_x : load in piston direction
- $F_{y:}$ load perpendicular to piston
- m_{rec}: reciprocating mass
- m_{rot}: rotating mass
- r: stroke
- L: connection rod length
- $\omega_{0:}$ circular velocity
- t: time

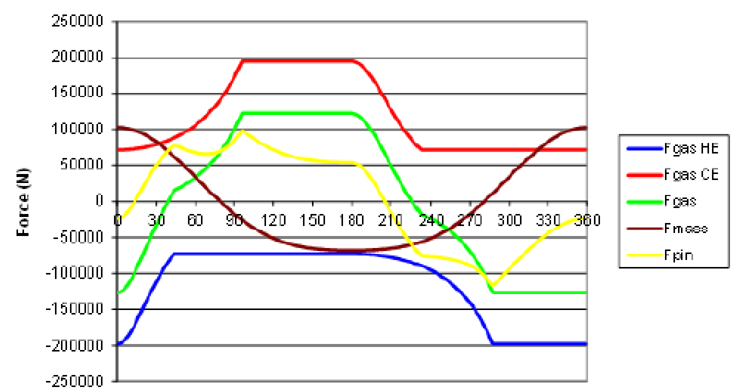
Dynamic Forces generated by Reciprocating Compressors

<u>Vertical</u> crosshead forces

- important when supporting structures in vertical direction are flexible
- created when rotation motion is converted into reciprocating motion
- can be retrieved from crosshead pin load (horizontal direction)

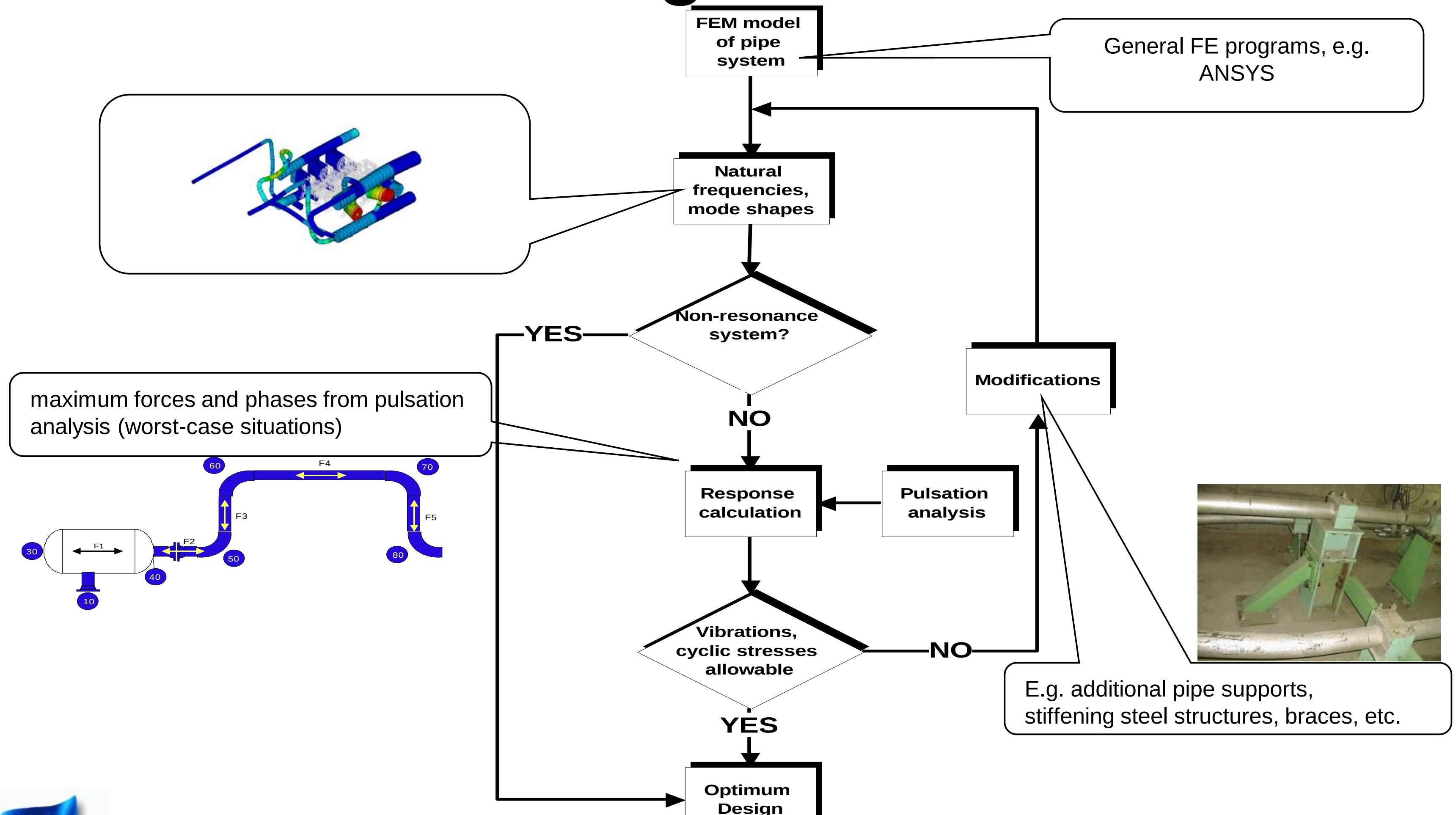


Forces on crosshead



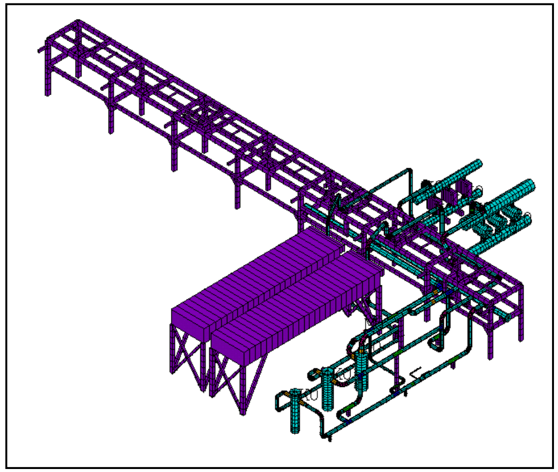
Example of gas & inertia loads on crosshead pin (yellow line indicates total pin load)

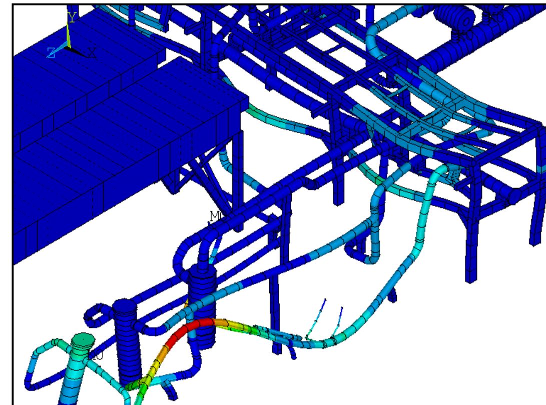
Mechanical response analysis according to API 618



Avoid unacceptable vibrations and fatigue failures

Example of an Underground Gas Storage (UGS) system



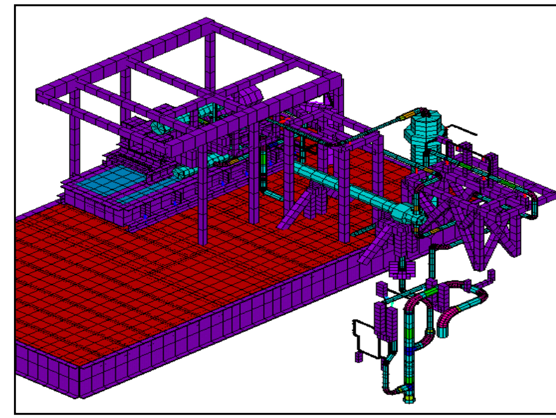


Mode shape at 23.14 Hz

<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

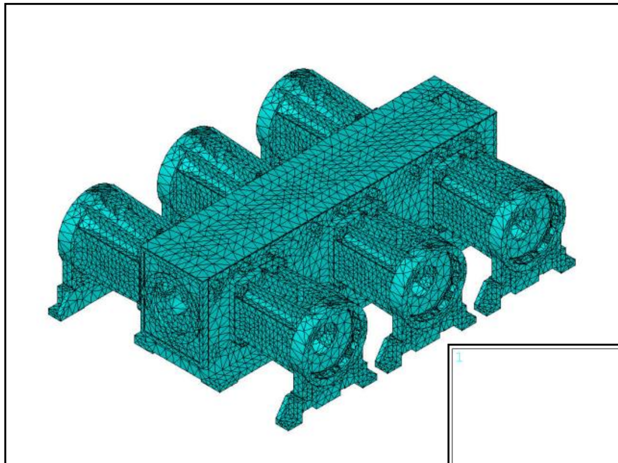
Some examples of off-shore models



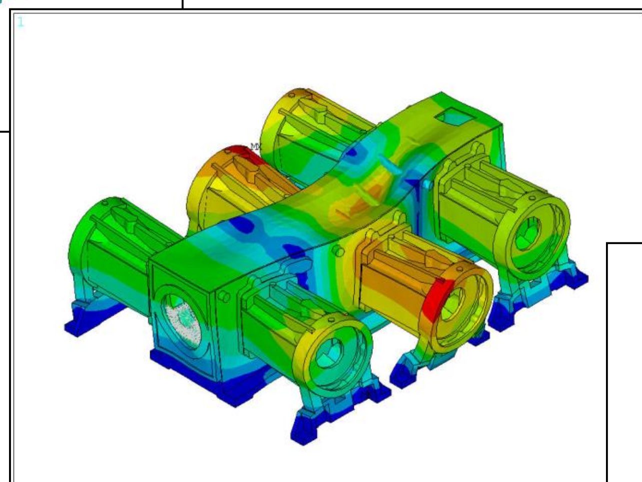
Challenges in offshore systems:

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

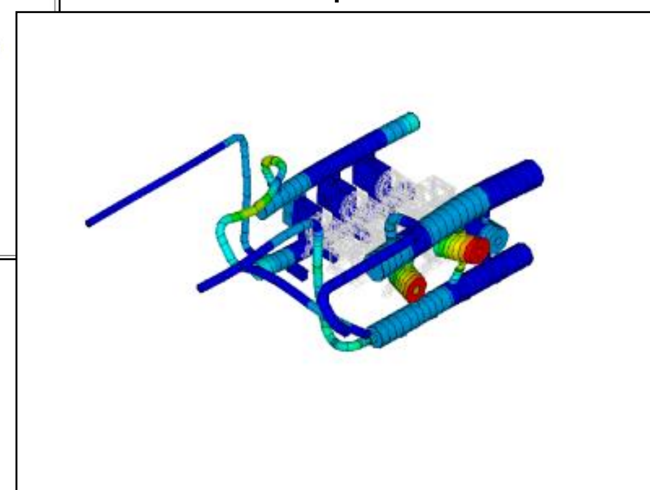
Examples of Compressor Finite Element models



Mode shape at 156.7 Hz



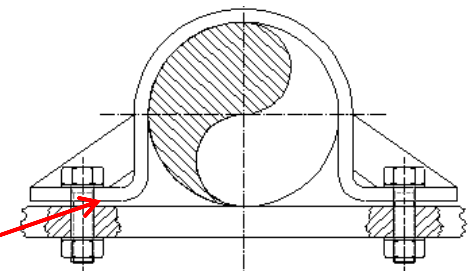
Mode shape at 94.7 Hz

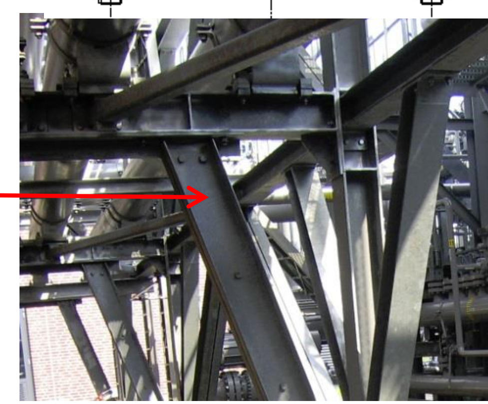


Mitigation Vibrations & Cyclic Stress Levels

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

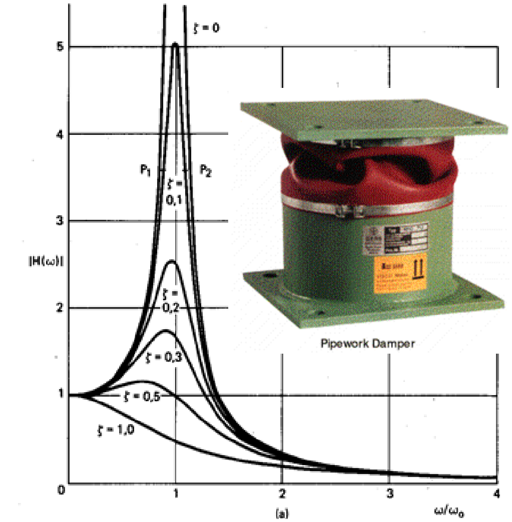
$$\omega_0 = \sqrt{k / m}$$





  - 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

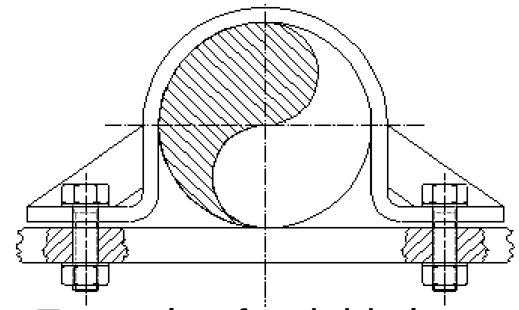


- Damping:
  - Most effective at resonance conditions
  - Viscous dampers (allow thermal motions)
  - Most effective for frequencies < 50 Hz

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

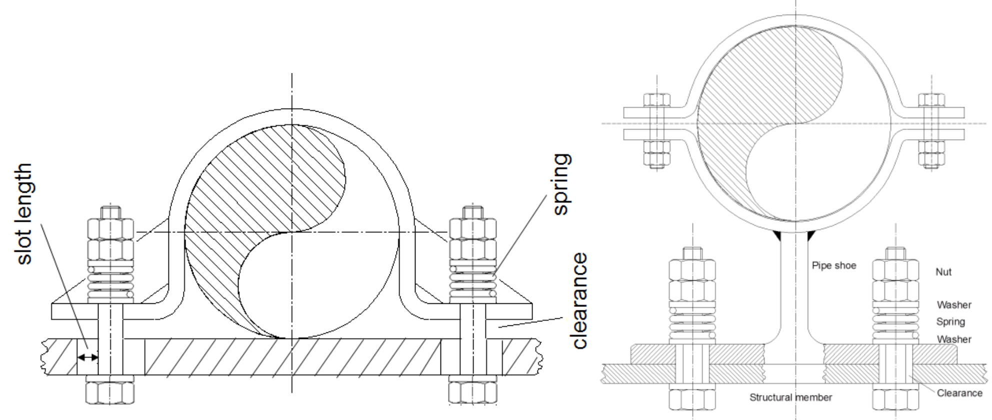
Pipe Supports

- Rigid clamps are required for systems which are subjected to vibrations
- Disadvantage of additional pipe clamps:
  - Introduction of too high expansion stresses



Example of a rigid clamp

- Solution:
  - spring hold down supports



Examples of spring hold down supports

- Required spring preload:

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

F_w= pulsation-induced reaction force

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

API 618 Standard

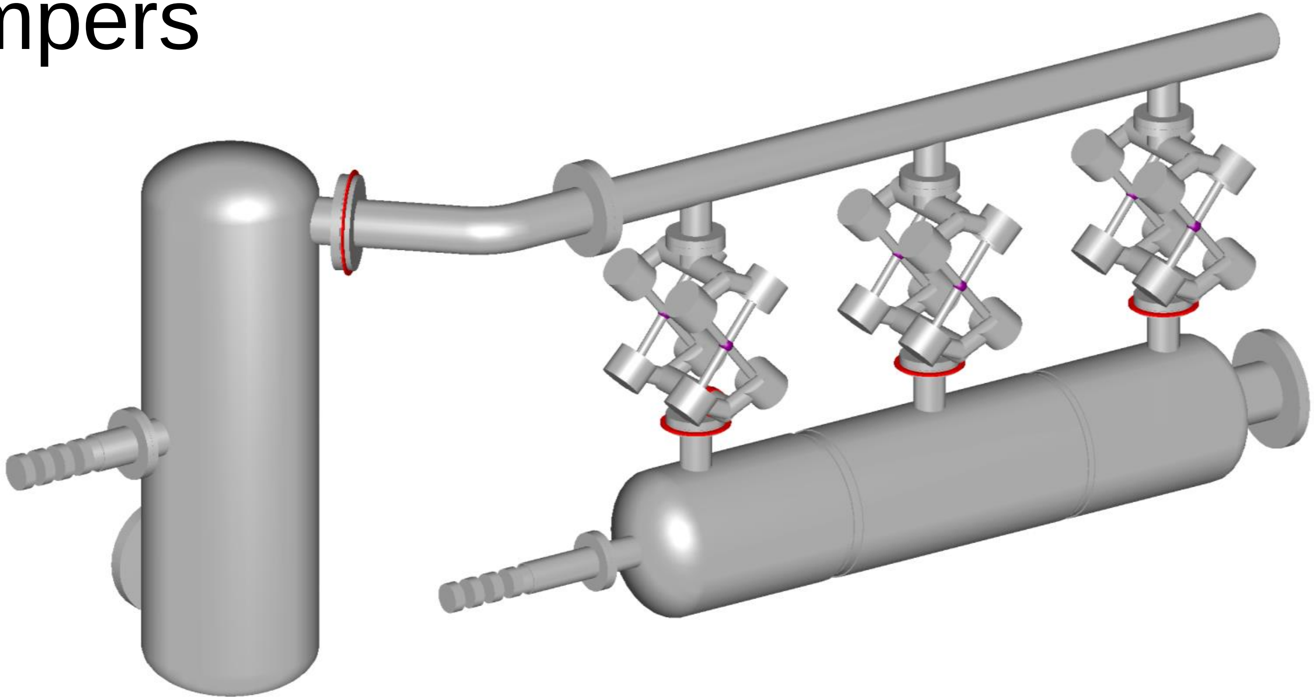
- Specifies allowable pulsation, vibration & cyclic stress levels
- Stipulates a design approach

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

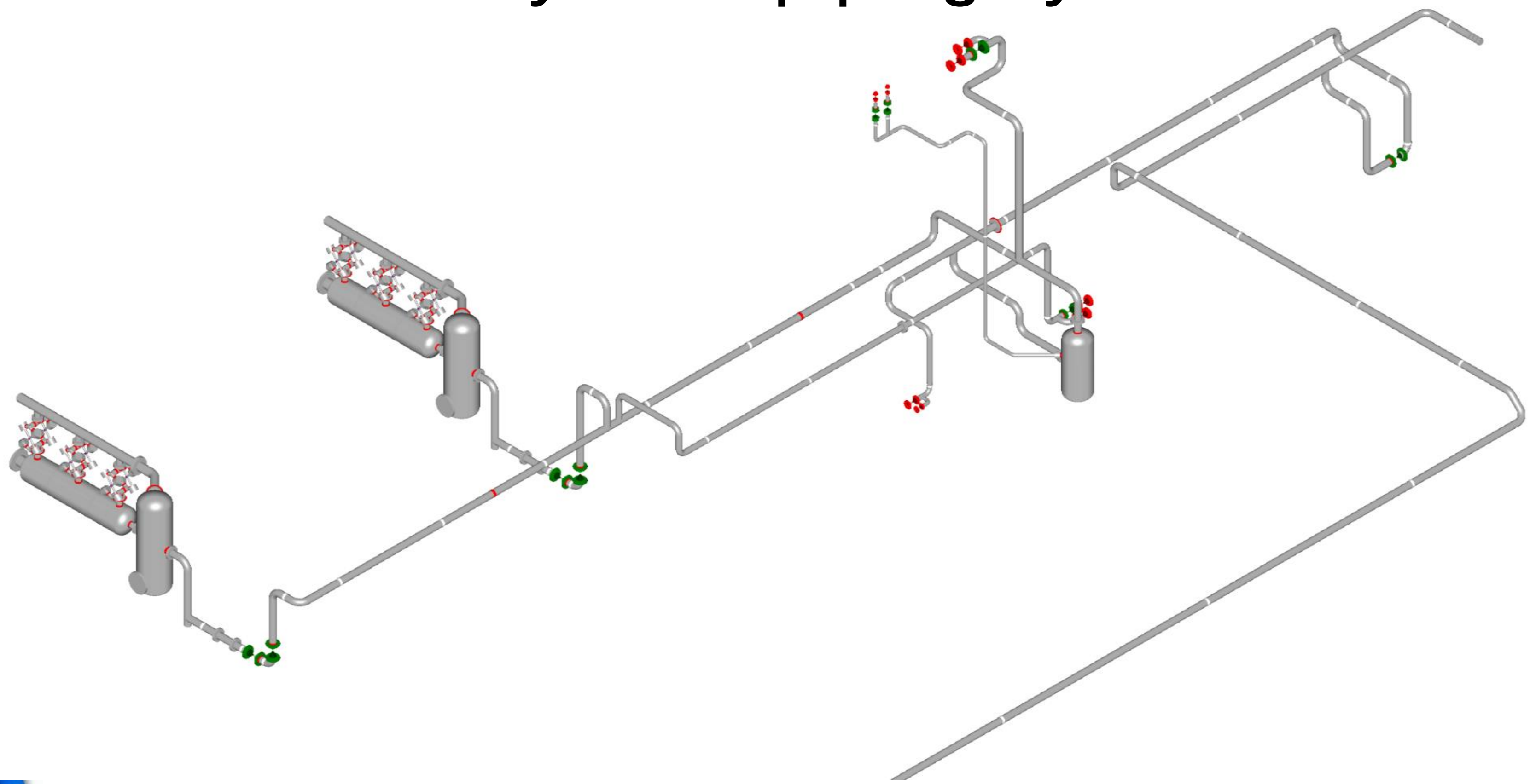
Design approach 3

1) Acoustic evaluation & design of pulsation dampers



Design approach 3

2) Pulsation analysis of piping system



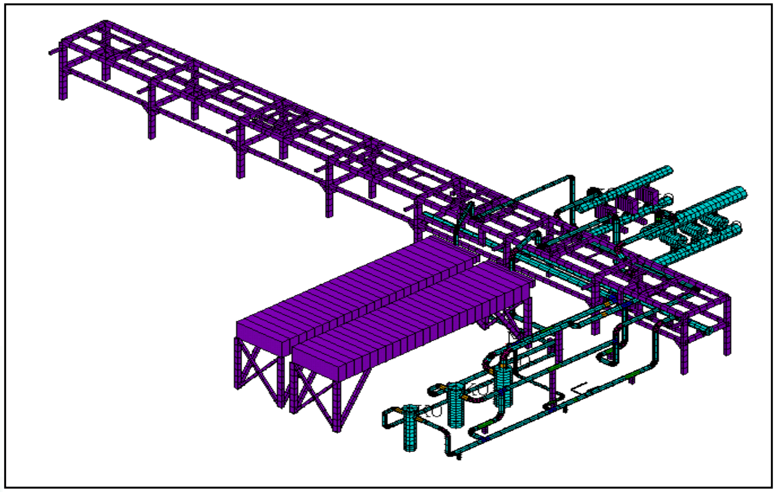
Design approach 3

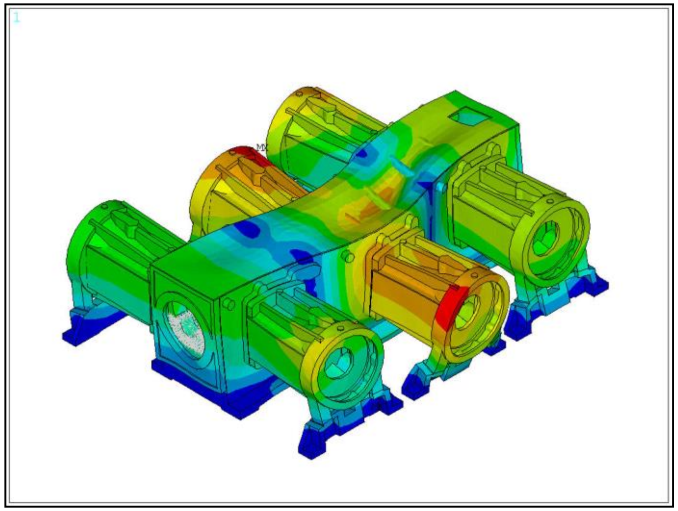
3) Forced mechanical response analysis

Vibrations and cyclic stresses of:

– Piping

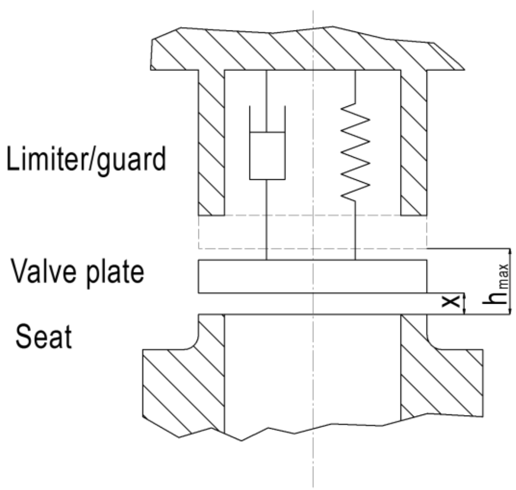
– Compressor manifold

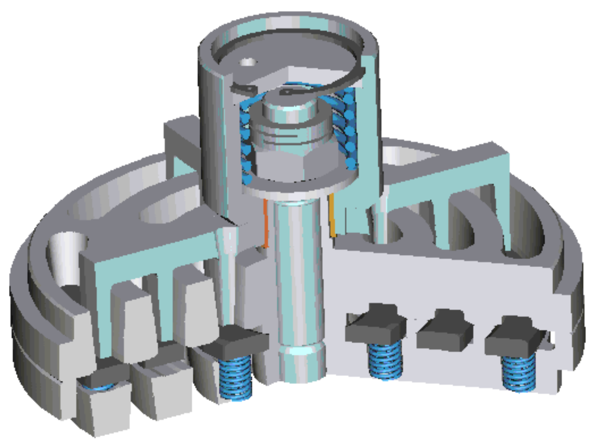


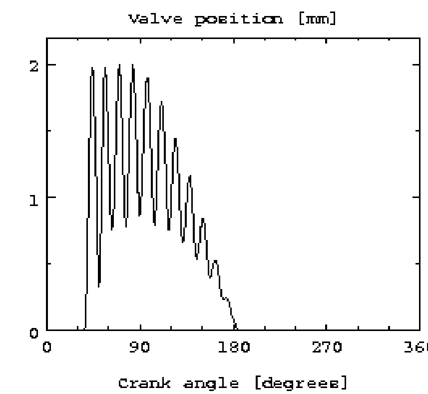


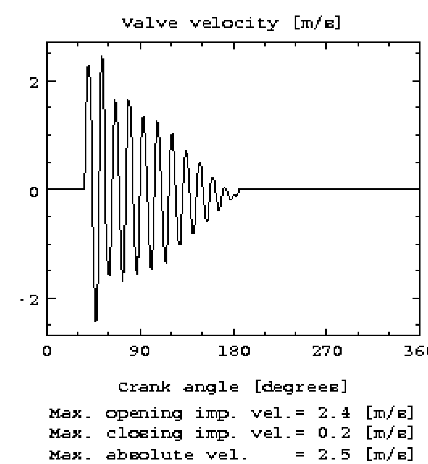
Design approach 3

4) Analysis of compressor valve dynamic behaviour









Thank you for your attention!

Questions ?

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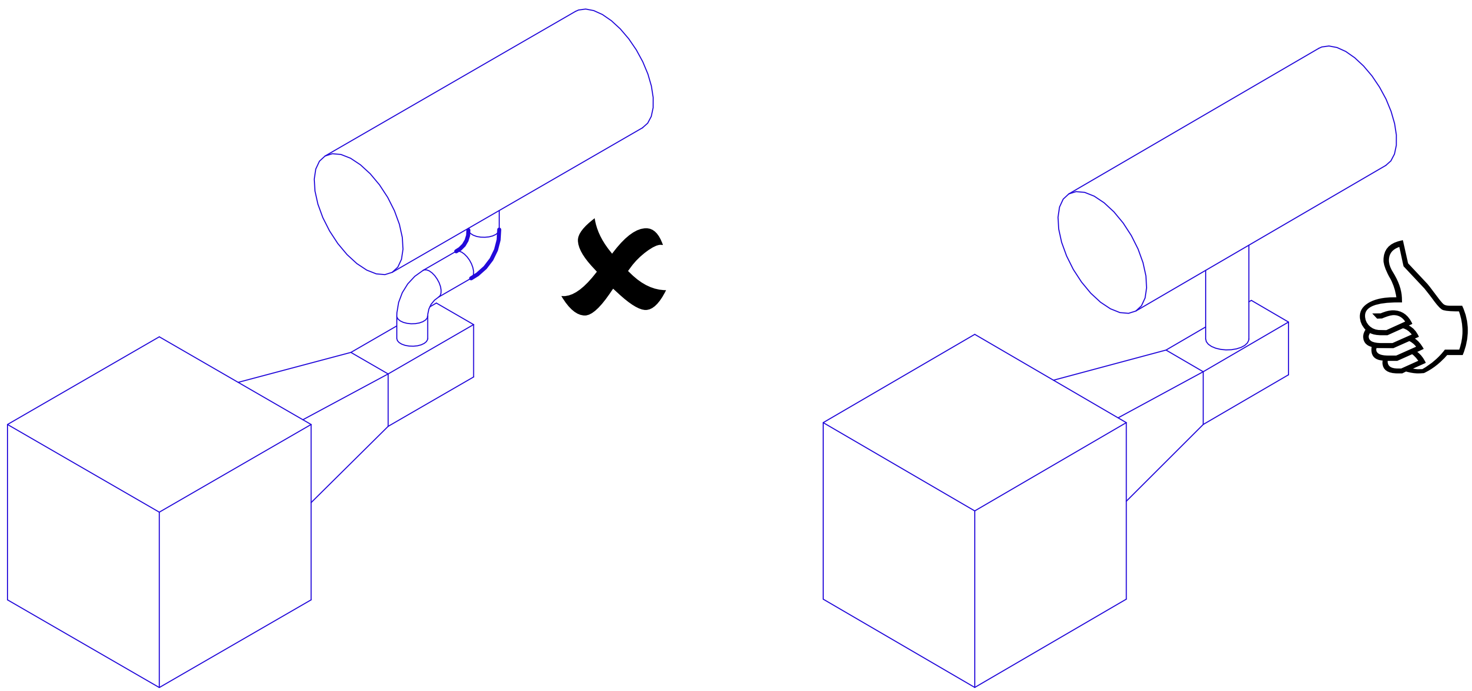
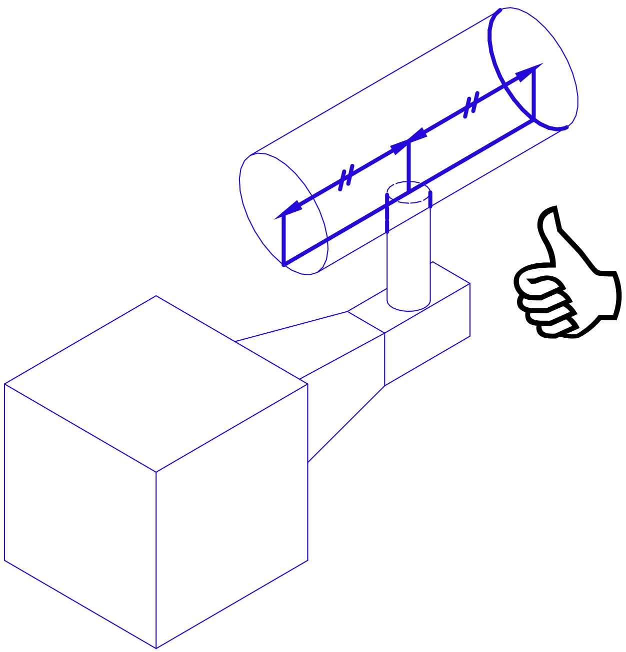
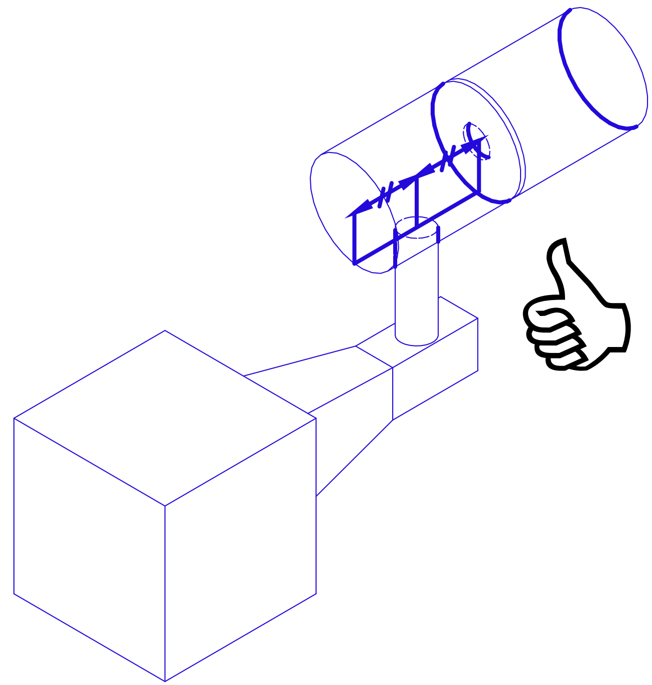
Encore: basic design rules (do's and dont's)

Keep cylinder connection of pulsation damper as short as possible

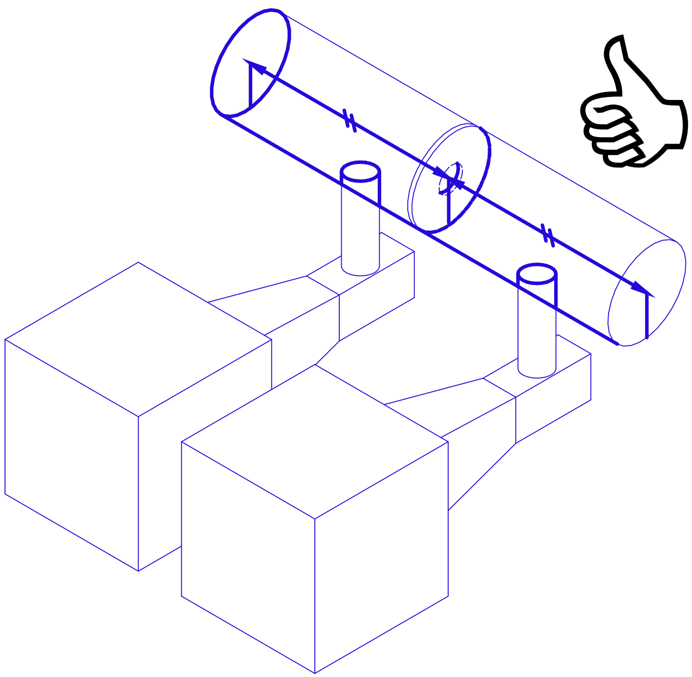


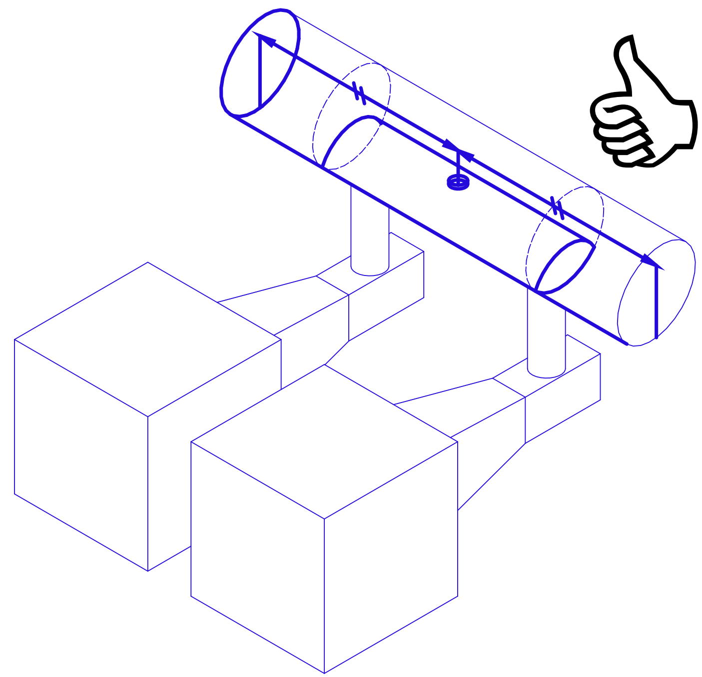
Symmetric layout of cylinder connection in the damper compartment



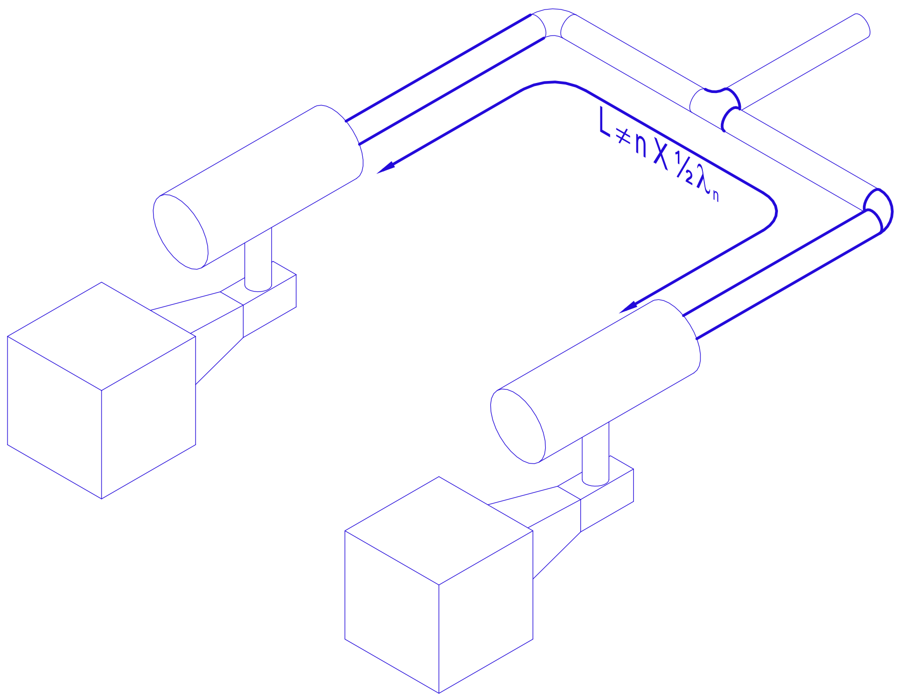


Manifold damper: Baffle plate or extended cylinder connection

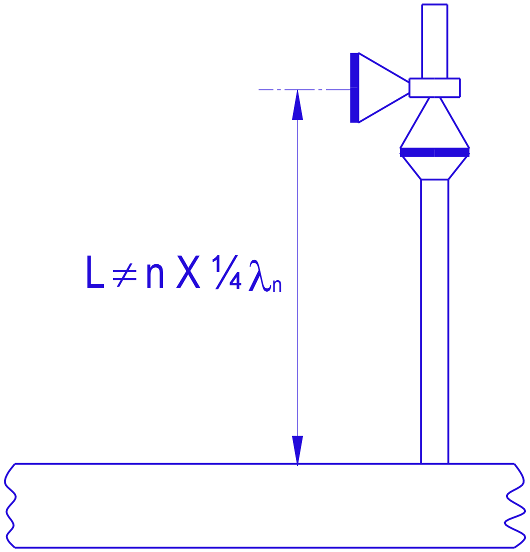




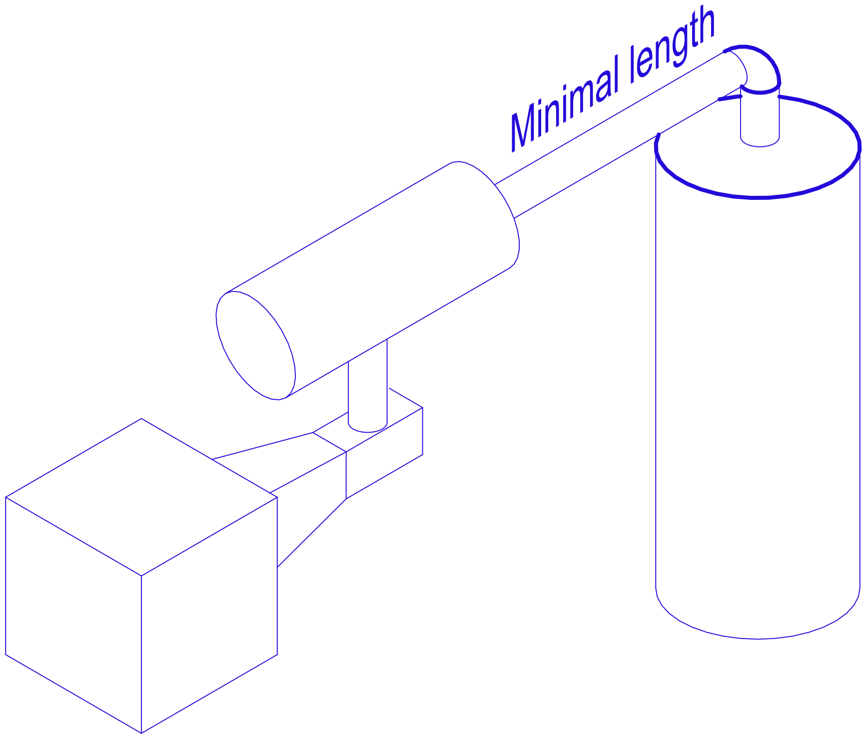
Avoid standing waves and Helmholtz resonance between vessels



Avoid standing waves in closed side branches

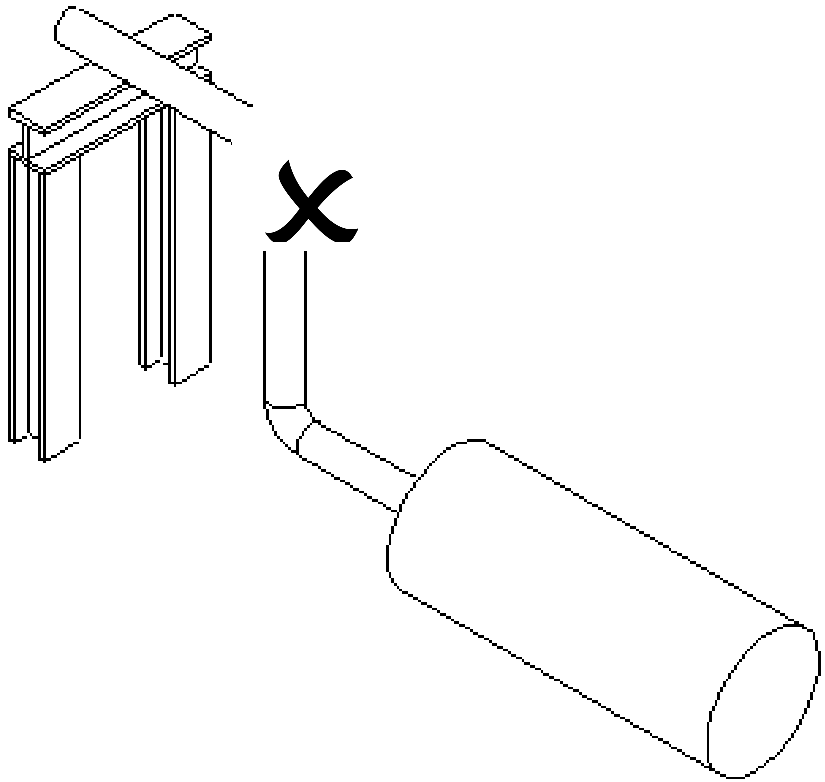


Minimize length between large vessel and damper

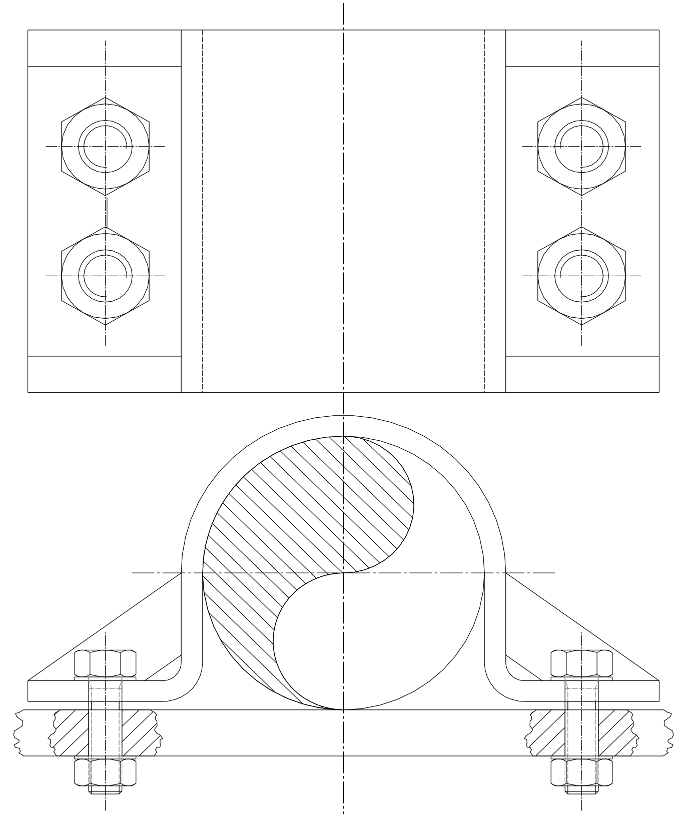


Piping layout, clamps

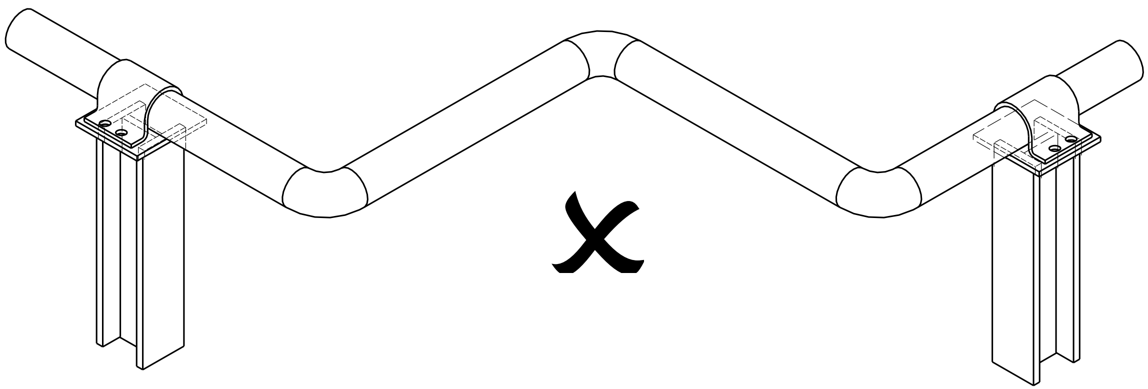
Keep all piping low



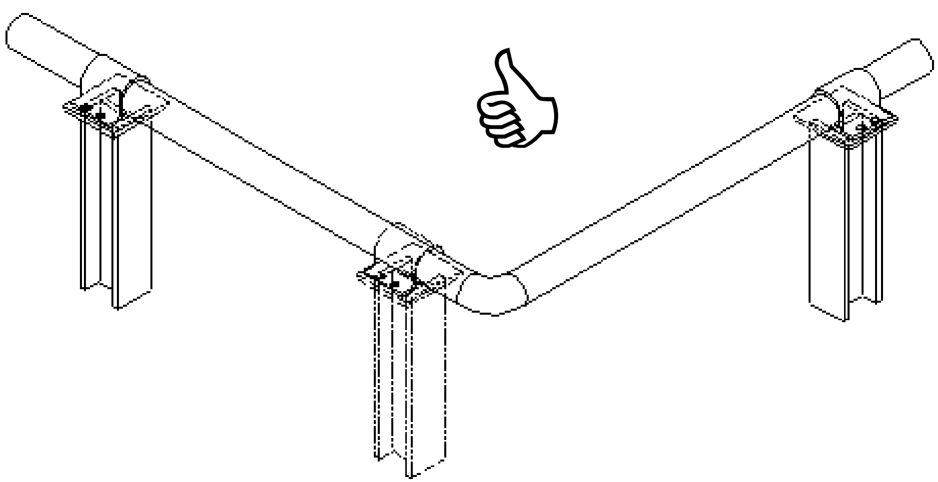
Good Clamps have width (do not use U-bolts)



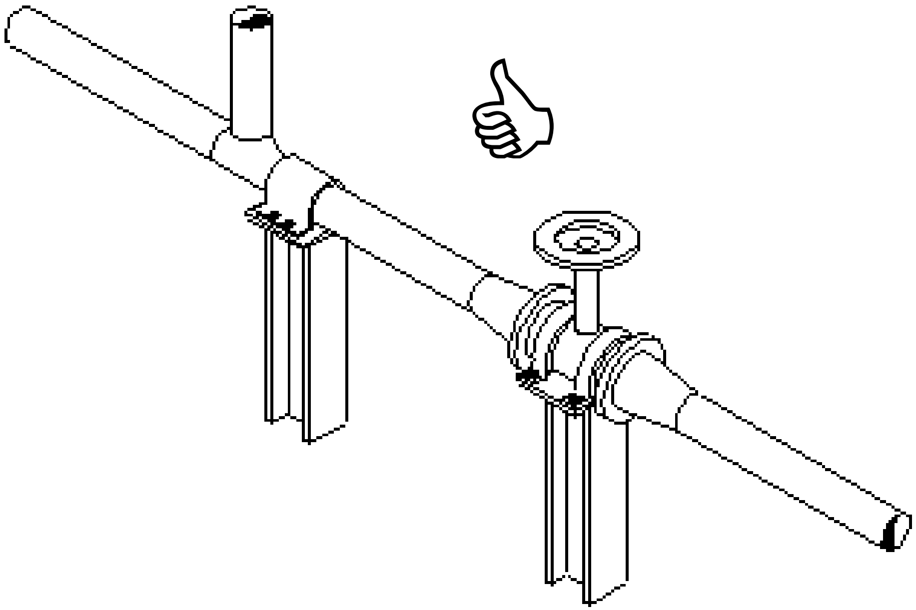
Do’s, Don’ts Design Rules

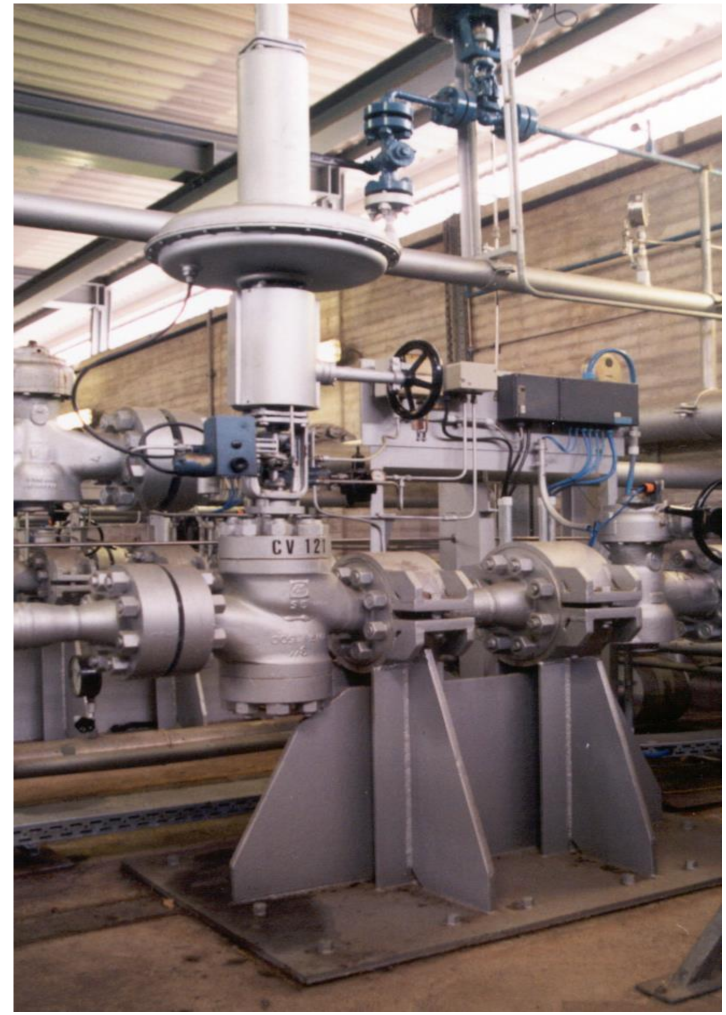


Minimise number of elbows and Tees
and avoid freestanding elbows



Do’s, Don’ts Design Rules





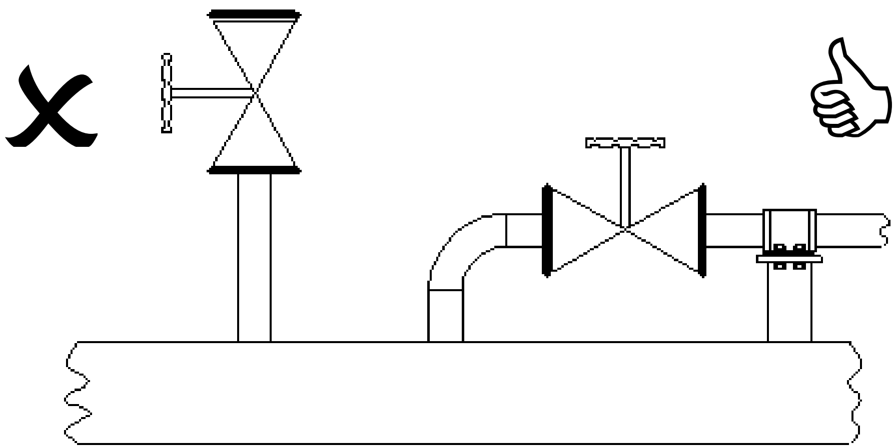
Locate supporting directly under heavy components (valves, flanges etc.) and Tee joints

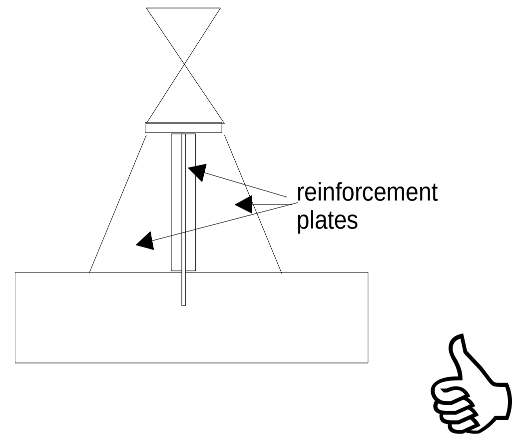
Do’s, Don’ts Design Rules

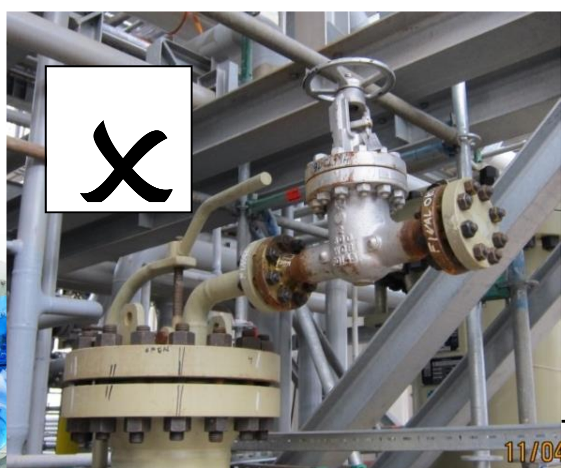
Brace small lines (drains, purge lines)

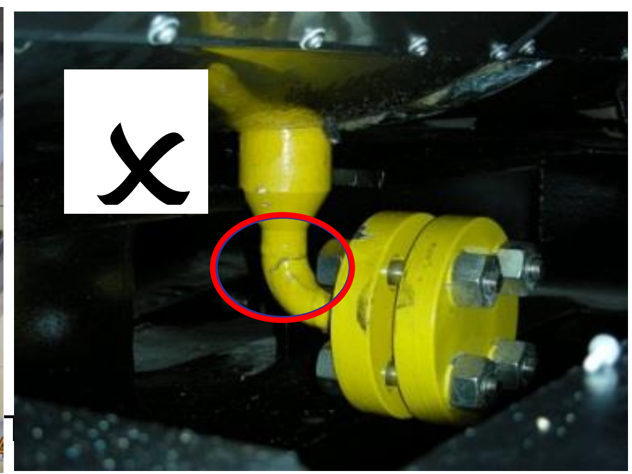
Do not use unreinforced branch connections

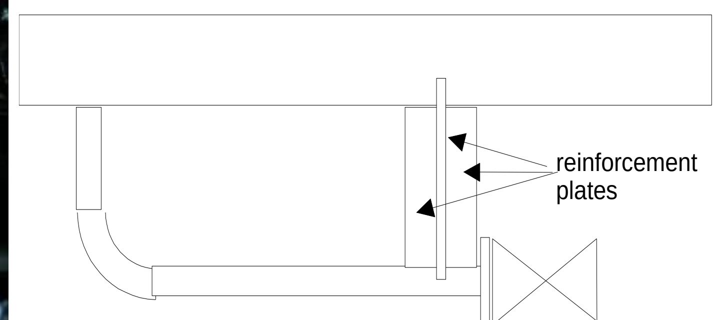
Apply two-plane bracing of small bore side branches back to main pipe





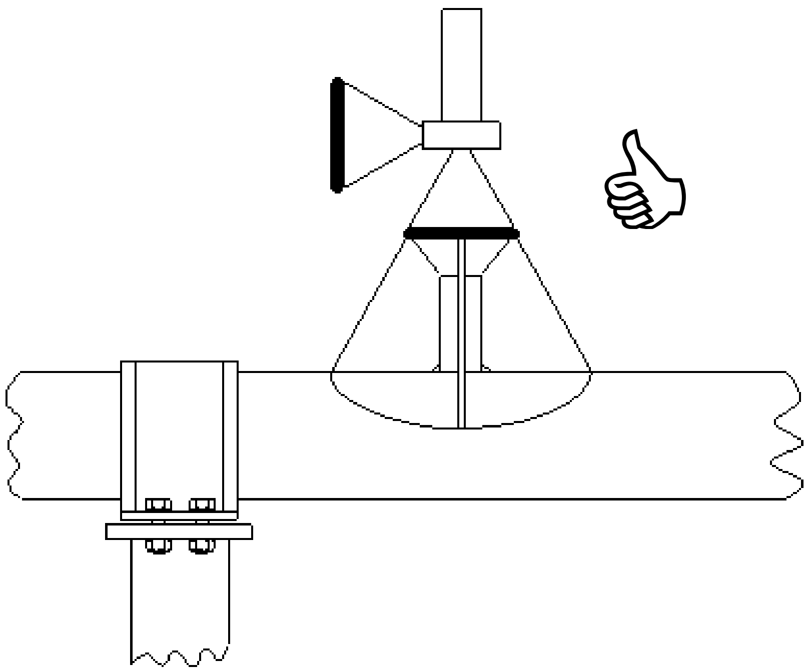


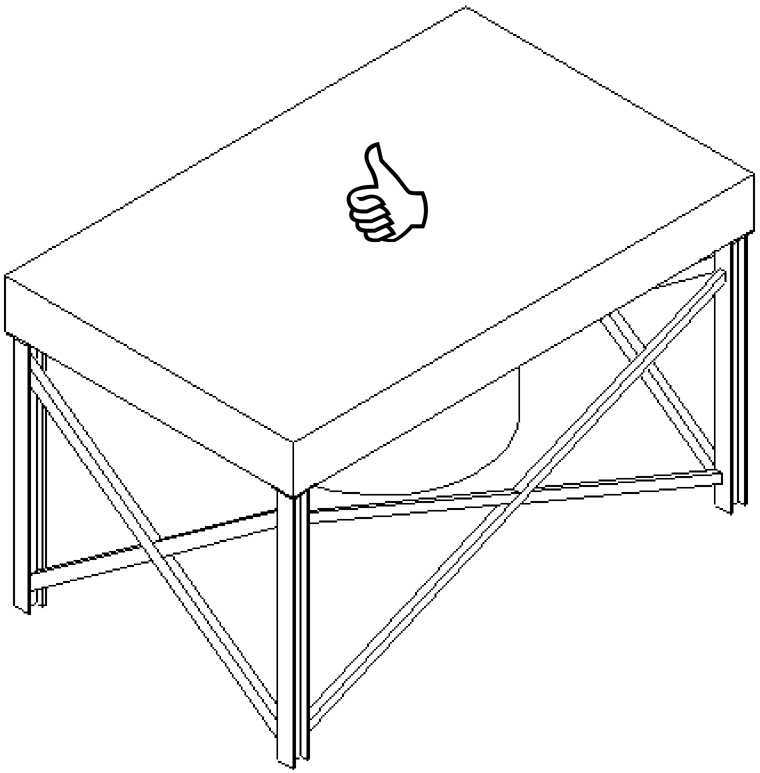




Do's, Don'ts Design Rules

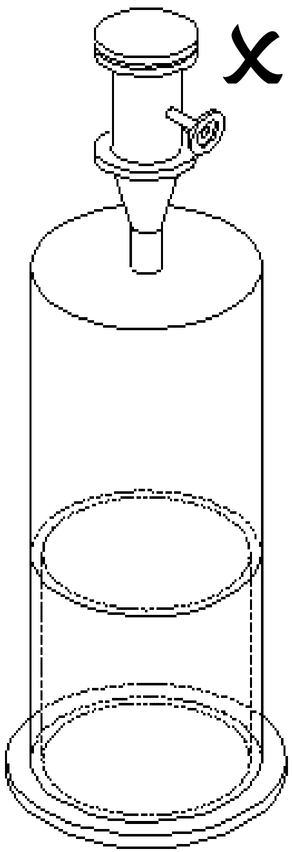
Keep relief valves close to main line – Apply X-bracing in air cooler frames

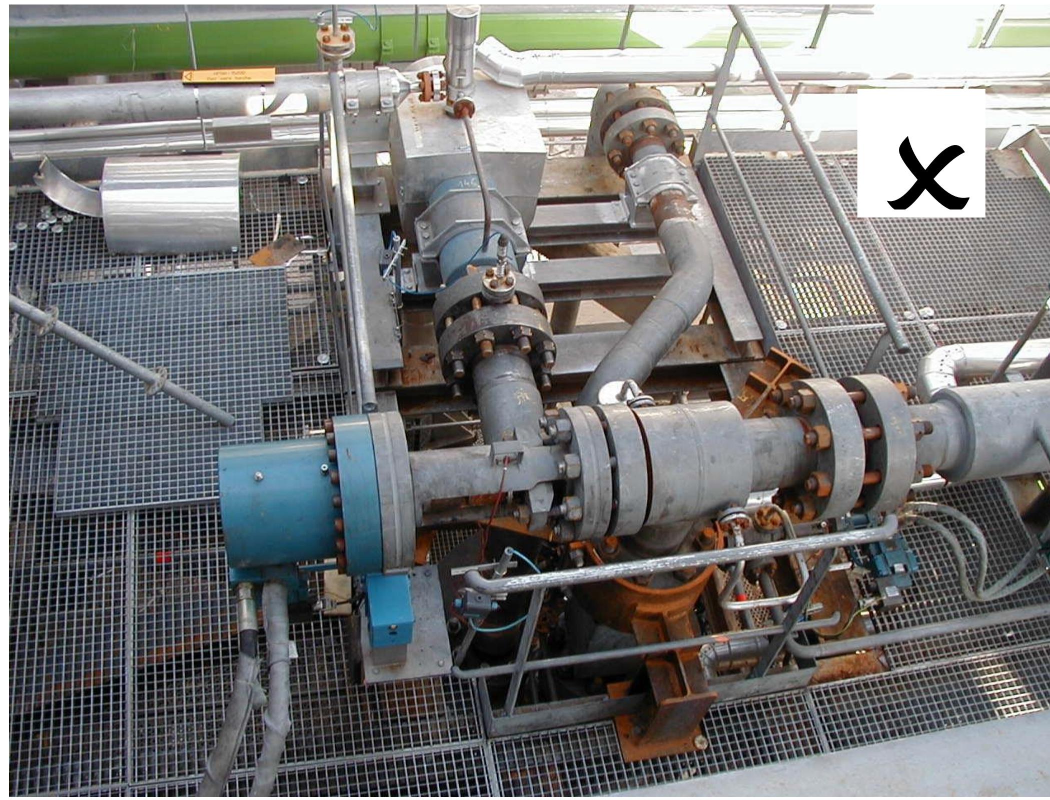




Do’s, Don’ts Design Rules

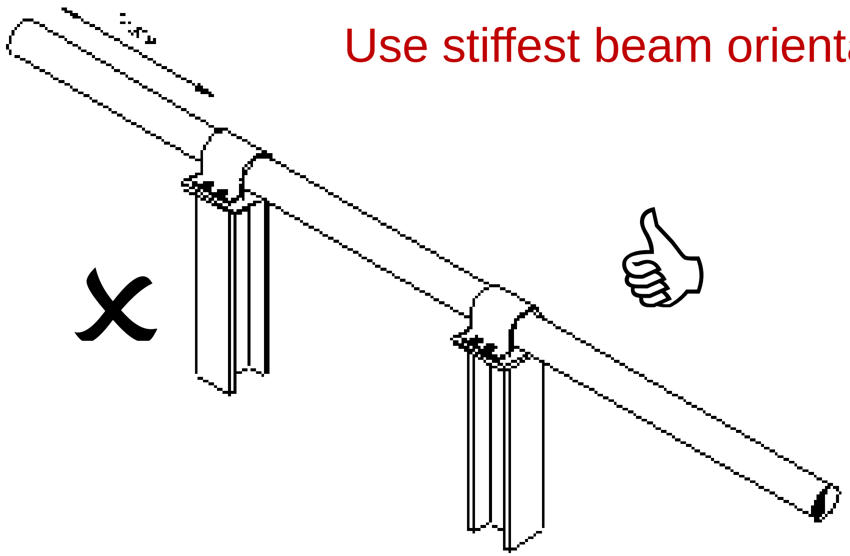
Avoid heavy valves at high elevation (also on top of separators)

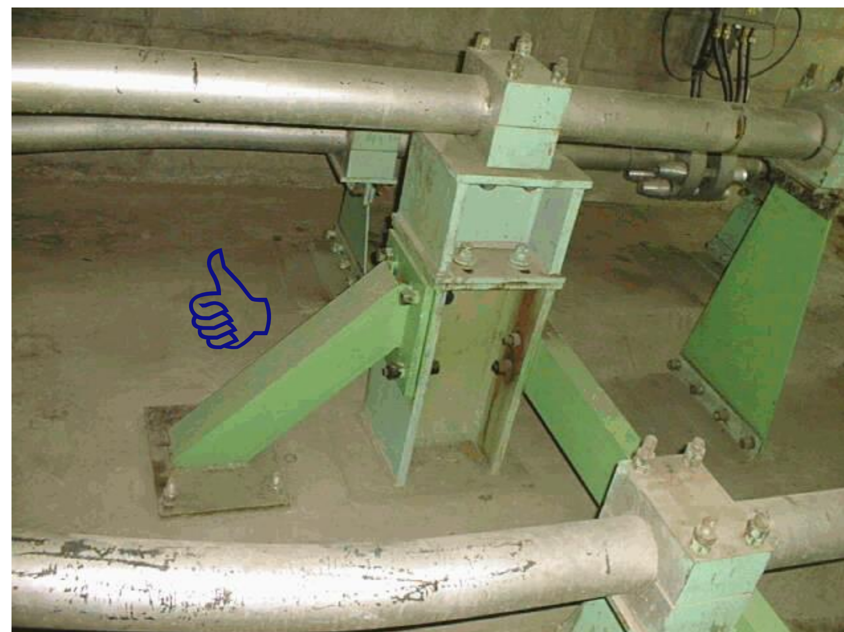




Do’s, Don’ts Design Rules

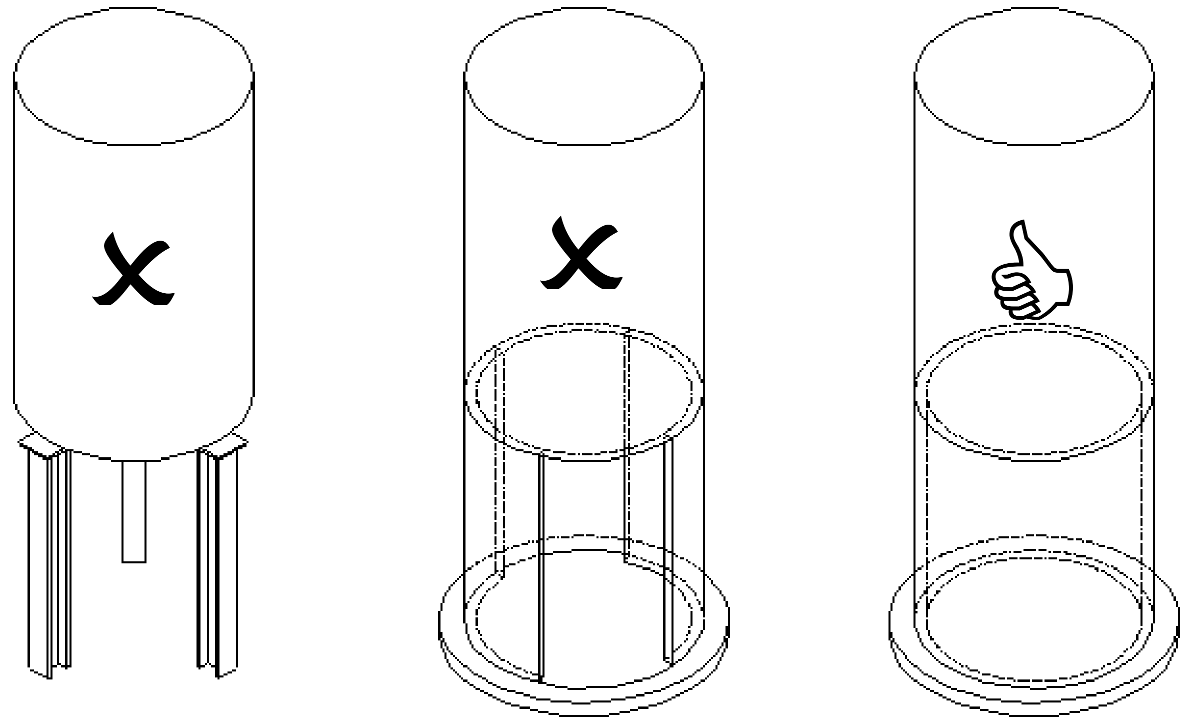
Use stiffest beam orientation in the force direction





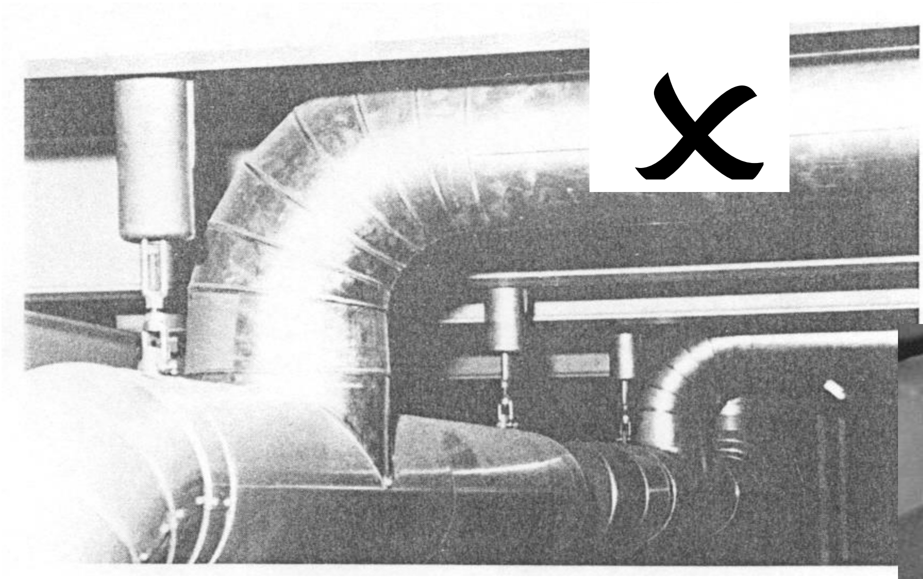
Do’s, Don’ts Design Rules

Use full skirts in vessel supporting

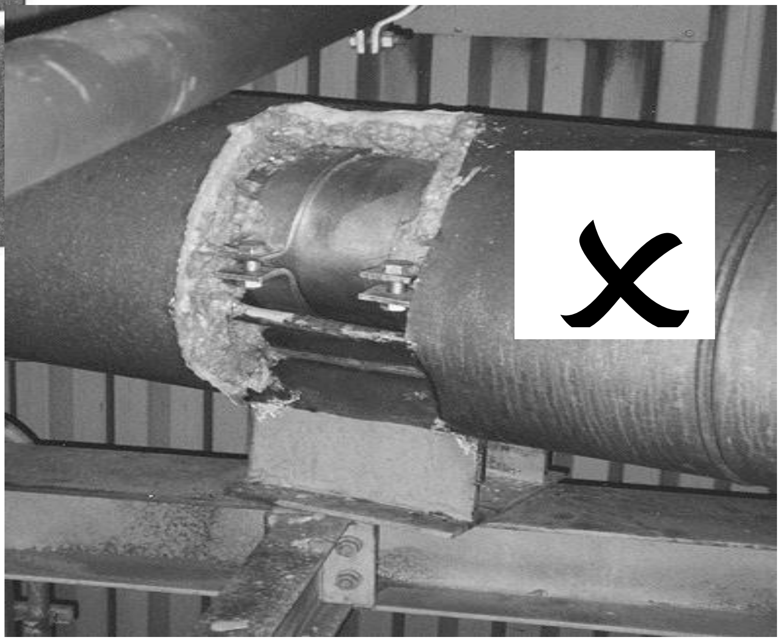


Do’s, Don’ts Design Rules

Avoid rod and constant load hangers

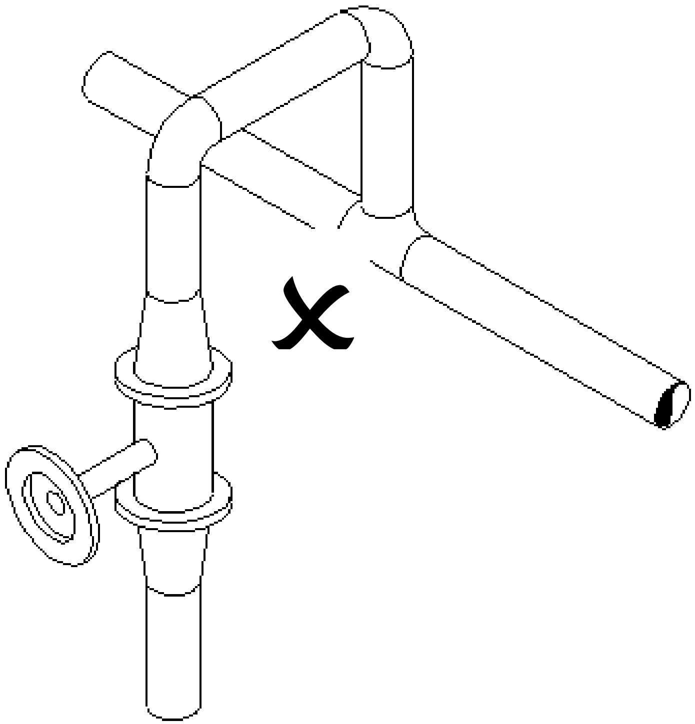


Avoid weight only supports

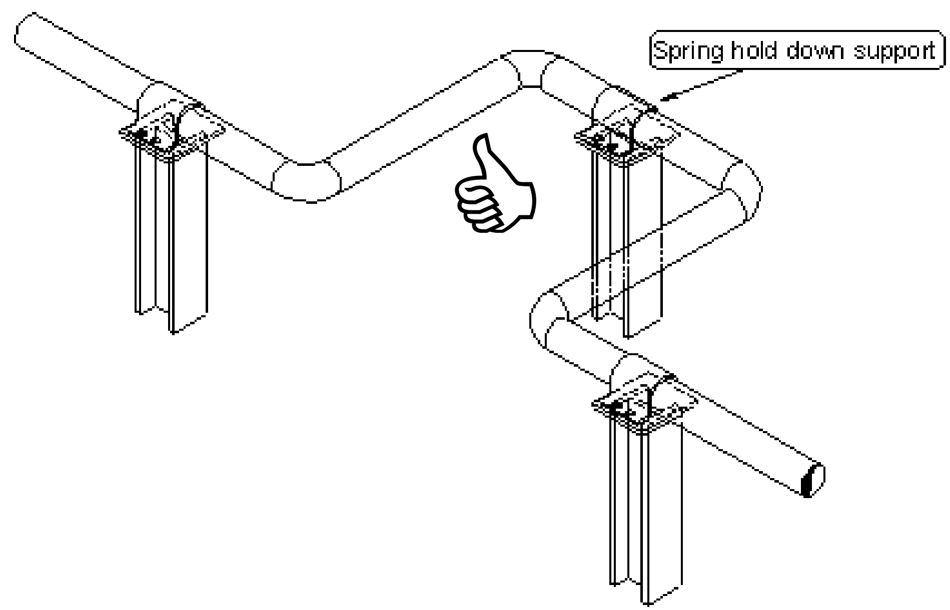


Do’s, Don’ts Design Rules

Avoid unsupported overhanging weight



Avoid unsupported expansion loops



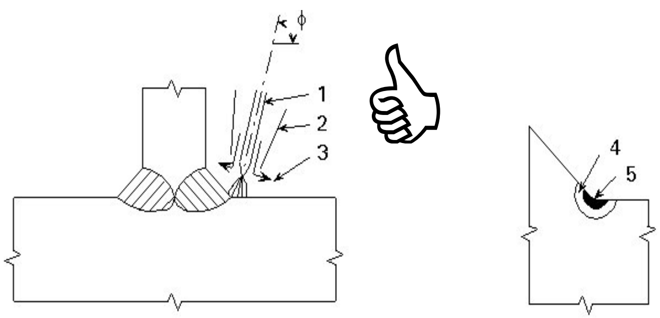
Do's, Don'ts Engineering Rules

Welds:

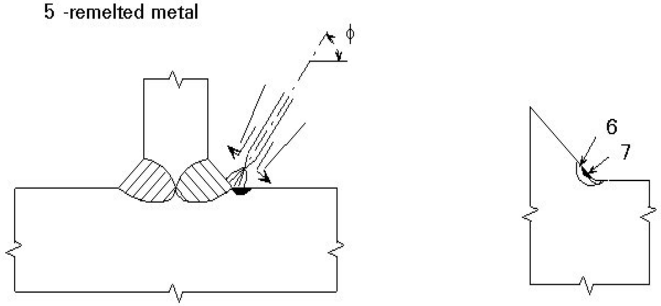
a) avoid weld imperfections

b) apply full penetration welds

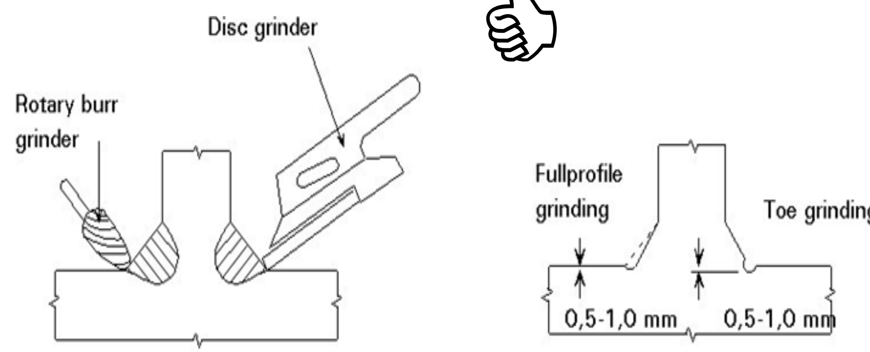
c) avoid sharp corners (grind welds)



(a) Single run
1 -tungsten electrode, 2 - nozzel, 3 -shielding, 4 -HAZ
5 -remelted metal

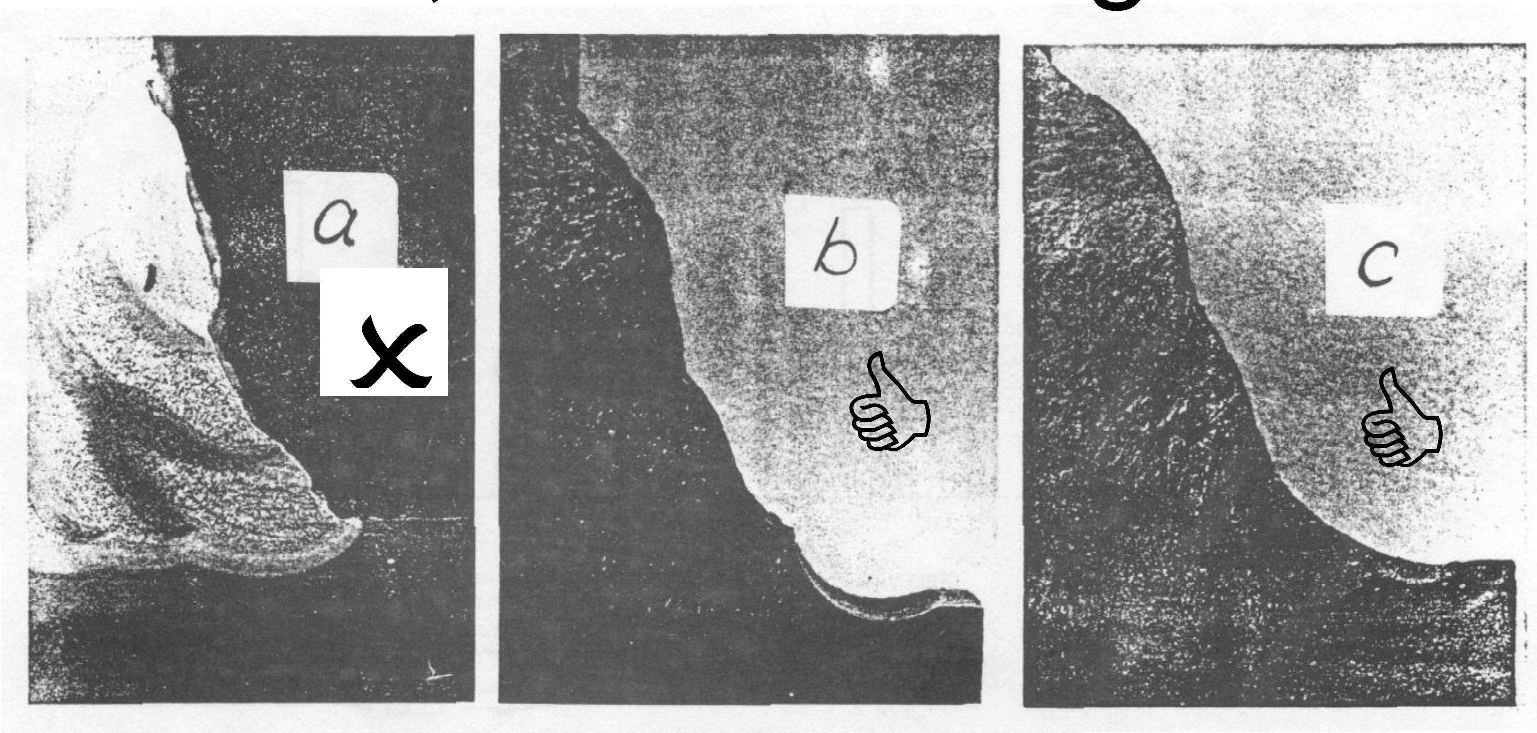


(b) Double run
6 -remelted metal from second TIG run, 7 -HAZ from run ref (8)



(a) 1 -Rotary burr grinding, 2 -Disc grinding (b) 3 -Full profile grinder, 4 -Toe grinding

Do's, Don'ts Design Rules



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

Do’s, Don’ts Design Rules

Use adequate grouting (epoxy resins)