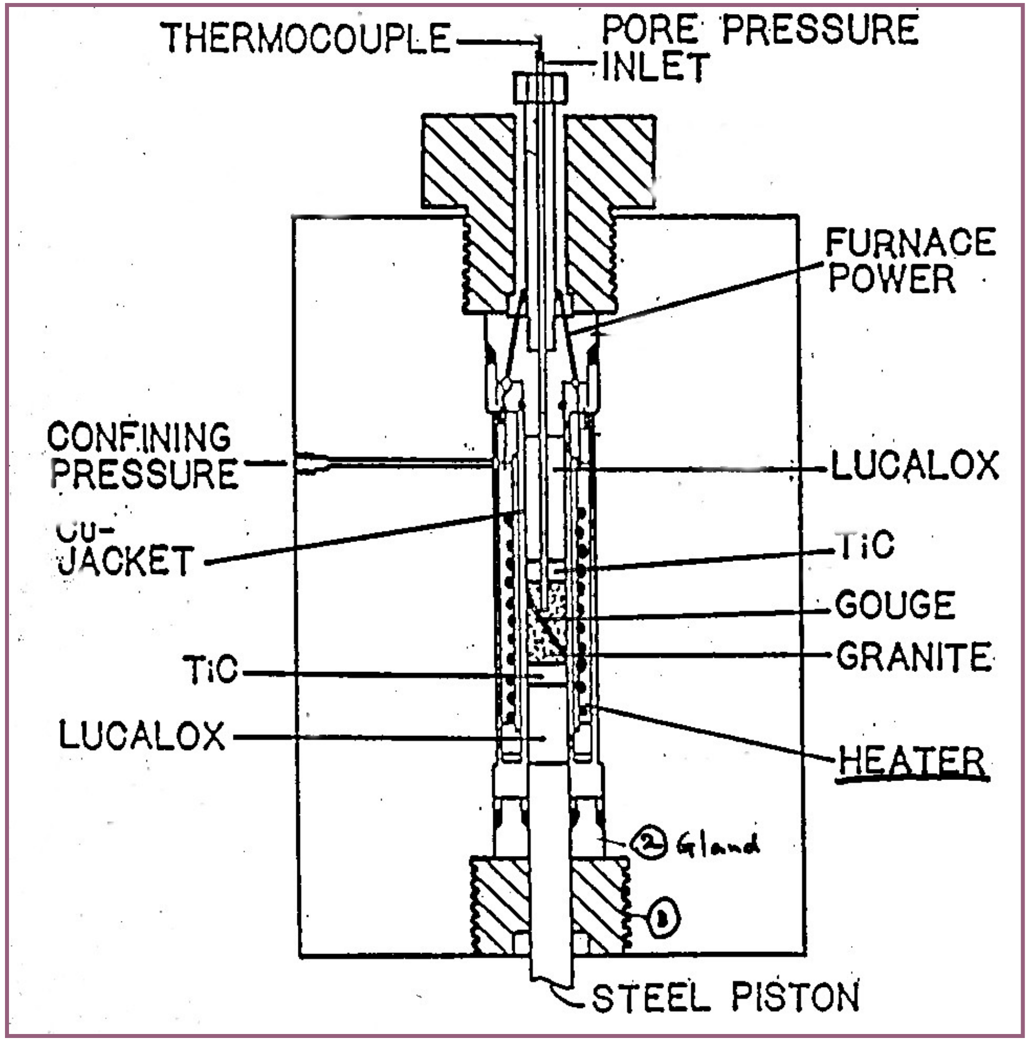
Pressure vessel

Ref :T. **Shimamoto, Seminar design (2013),** Tullis and Tullis, Experimetal deformation techniques (1986), Paterson and Wong Experimental rock déformation: the brittle field (2005), ...



(*MIT group*; D. E. Moore et al., 1989, JSG, 11: 329-341)

A pressure vessel for gas rig

(1) Pressure vessel

(3) Nut

Most vessel failure started from the first few threads of nuts. (L. M. Logan)

(4) Piston & spacers

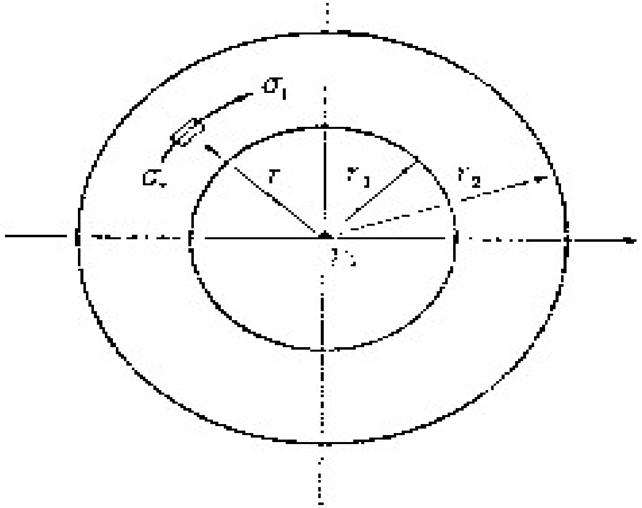
--- *split piston!!*

(5) Furnace & T/C

(6) Specimen & jacket

Stresses for an internally pressurized cylinder

Stresses for an internally pressurized cylinder



σ_t : tensile stress

σ_r : radial stress

σ_z : axial stress

$$\sigma_r = -\frac{P_1}{k^2-1}\left\{\left(\frac{r_2}{r}\right)^2 - 1\right\} \text{ kg/cm}^2$$

$$\sigma_t = \frac{P_1}{k^2-1}\left\{\left(\frac{r_2}{r}\right)^2 + 1\right\} \text{ kg/cm}^2$$

$$\sigma_z = \frac{P_1}{k^2-1} \text{ kg/cm}^2$$

The maximum tensile stress at the inner wall

$$\sigma_t^{max} = \left[\frac{(\kappa^2+1)}{(\kappa^2-1)}\right] P_i$$

with $\kappa = r_o/r_\iota$

r_o, r_ι : external & internal diameters

Example:
for $r_o/r_\iota = 3$, $\sigma_t^{max} = 1.25\ P_i$

(25 % increase in σ_t^{max} *due to* P_i)

Stresses for an internally pressurized cylinder

The maximum tensile Stress at the inner wall (most dangerous!)

$$\sigma_t^{max} = \left[\frac{(\kappa^2 + 1)}{(\kappa^2 - 1)}\right] P_i$$

with $\kappa = r_o/r_i$

$r_o/r_i = 2, \quad \sigma_t^{max} = 1.67\, P_i$

$r_o/r_i = 3, \quad \sigma_t^{max} = 1.25\, P_i$

$r_o/r_i = 4, \quad \sigma_t^{max} = 1.13\, P_i$

$r_o/r_i = 5, \quad \sigma_t^{max} = 1.08\, P_i$

$r_o/r_i = 10, \quad \sigma_t^{max} = 1.02\, P_i$

(1) Select specimen size and determine r_i.

(2) Select material for vessel

Example AISI E4347 ---

Tensile strength = ca. 1 GPa

(3) Select outer diameter r_o and determine pressure limit with safety factor of 4.

Example: for $r_o/r_i = 3$

$P_i^{max} = (1\ GPa/4)/1.25$

$= 200\ MPa$

This simple method gives about two-times safety for internal yielding.

Internally and Externally Pressurized Cylinder (THICK WALL)

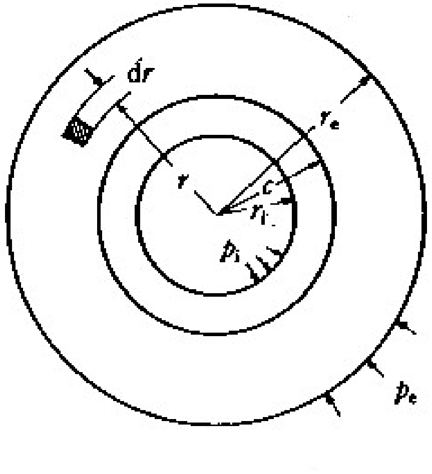
(Tesile stresses are taken as positive.)

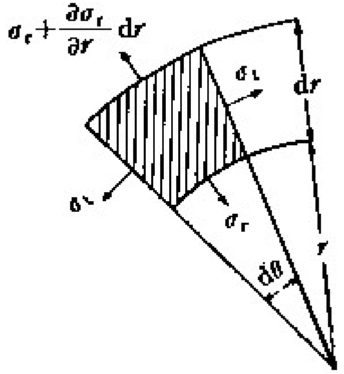
$$\sigma_t = \frac{-p_e K^2 + p_i}{K^2 - 1} - \frac{(p_e - p_i) K^2}{K^2 - 1} \left(\frac{r_i}{r} \right)^2$$

$$\sigma_r = \frac{-p_e K^2 + p_i}{K^2 - 1} + \frac{(p_e - p_i) K^2}{K^2 - 1} \left(\frac{r_i}{r} \right)^2$$

$$\sigma_z = \frac{1}{K^2 - 1} p_i \qquad \sigma_z =$$

(with confined ends) (with free ends)





Internally and Externally Pressurized Cylinder

(Tensile stresses are taken as positive.)

$$\sigma_t = \frac{-p_e K^2 + p_i}{K^2 - 1} - \frac{(p_e - p_i) K^2}{K^2 - 1}\left(\frac{r_i}{r}\right)^2$$

$$\sigma_r = \frac{-p_e K^2 + p_i}{K^2 - 1} + \frac{(p_e - p_i) K^2}{K^2 - 1}\left(\frac{r_i}{r}\right)^2$$

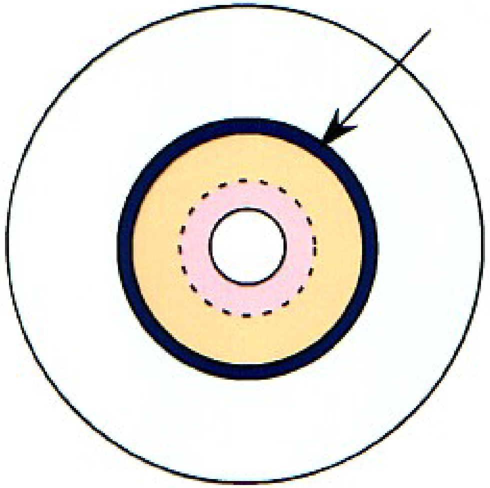
$$\sigma_z = \frac{1}{K^2 - 1} p_i \qquad \sigma_z = 0$$

(with confined ends) (with free ends)

$$\boxed{\sigma_t^{max} = \left[\frac{(\kappa^2 + 1)}{(\kappa^2 - 1)}\right] p_i - \left[\frac{2\kappa^2}{(\kappa^2 - 1)}\right] p_e}$$

For $\kappa = 3$, $\sigma_t^{max} = 1.25\, p_i - 2.25\, p_e$

Externally pressurized double-walled pressure vessel



P_{oil} = ca. 500 MPa can make it possible to sustain p_i > 1 GPa.

Plastic Deformation of Pressure Vessel

Tresca criterion:

$$\sigma_1 - \sigma_3 = 2\tau_y = \sigma_y$$

von Mines Criterion:

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2\sigma_y^2$$

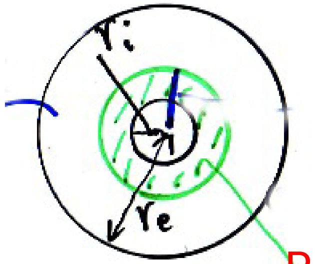
Initial yielding (Tresca):

$$p_y = \frac{\sigma_y}{2}\left(1 - \frac{1}{K^2}\right)$$

Total yielding (Tresca):

$$p_{fy} = \sigma_y \ln K$$

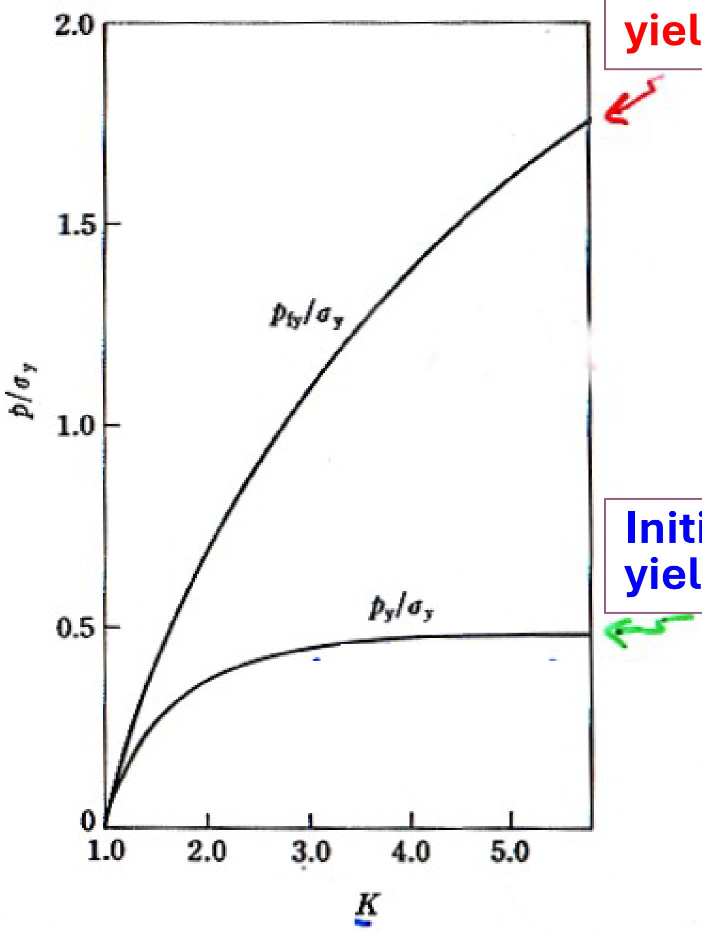
Elastic



Plastically deformed zone

Solid-pressure medium rig is used close to this limit!

Total yielding

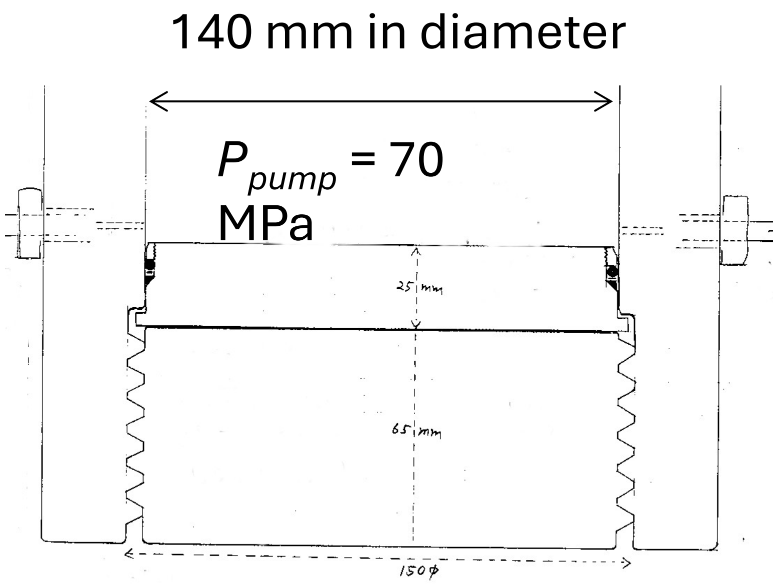


Initial yielding

Heard's used it
But NOT EU

Designing Nuts

'I saw more than 10 failed pressure vessels. In most cases, failure started from the first few threads of Nuts. It is best to design nuts so only one thread sustains forces acting on it. (*J. M. Logan*)'



Area = 154 cm^2, F = 108 tons

(1) Determine the wall thickness of the nut. In many cases, this is taken too small!

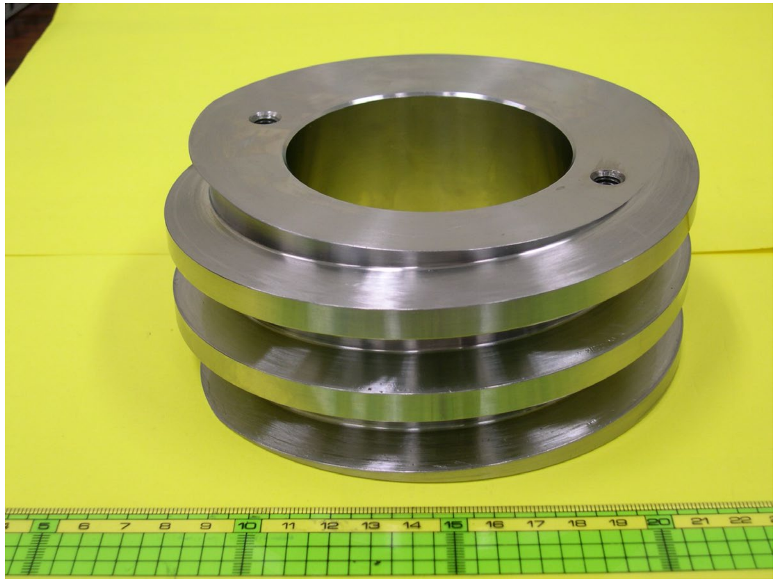
(2) Calculate force acting on nut.

(3) Determine the base width of nut with 4 times safety to support the force with one thread. Trapezoidal threads are better!

$$\frac{108 \text{ tons}}{44\,W} = 10/4 \text{ Kb}$$

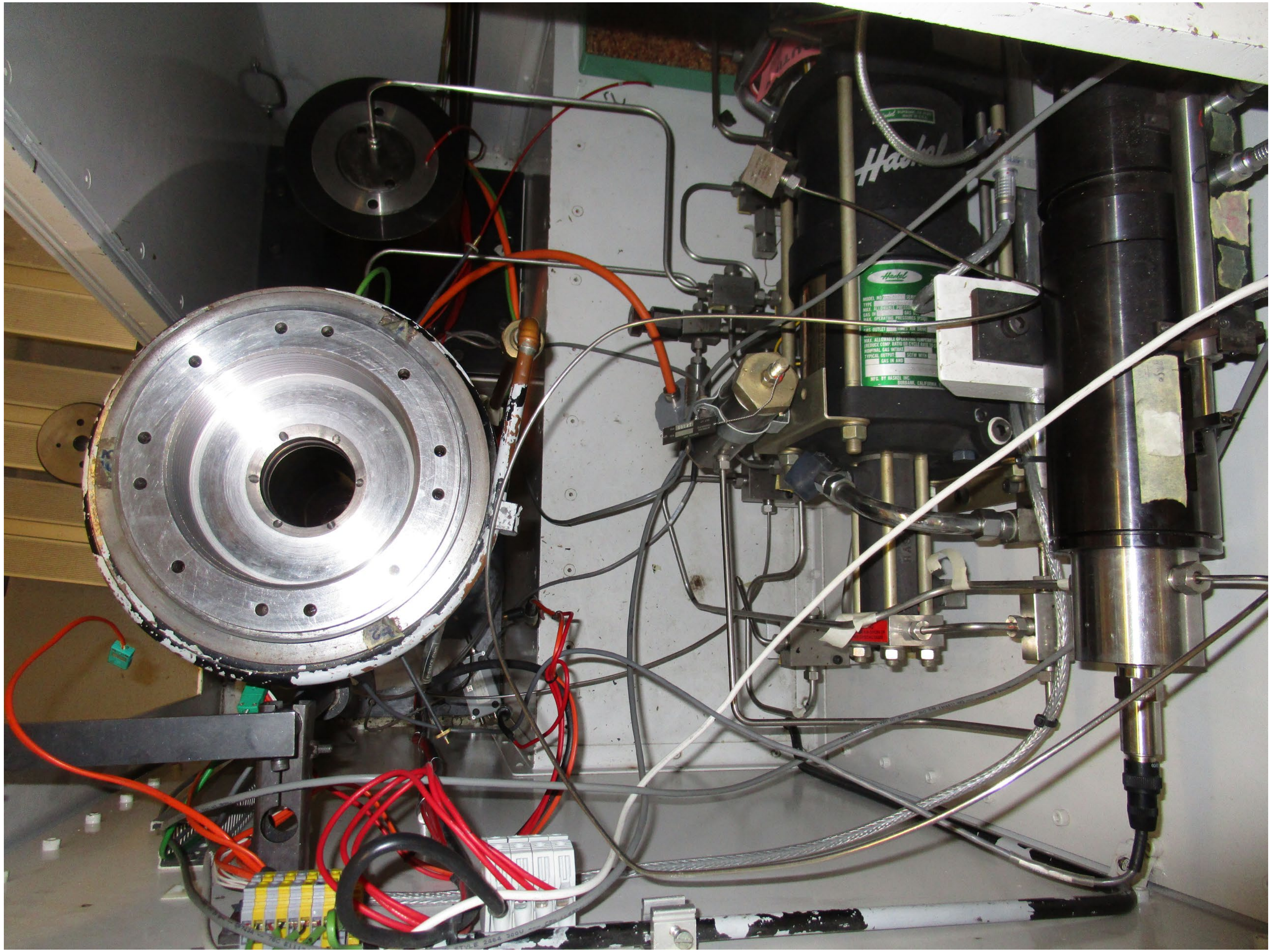
Circumference = 14 x 3.14 = 44cm
Take shear strength as 1 GPa = 10 Kb.
Then with 4 times safety the width of thread *W* has as to be 0.98 cm = 9.8 mm.

Upper nut can sustain
about
100 ton force.



Be careful for violent fracturing experiments!

The assembly is ideal for gas rig.



Paterson machine Montpellier

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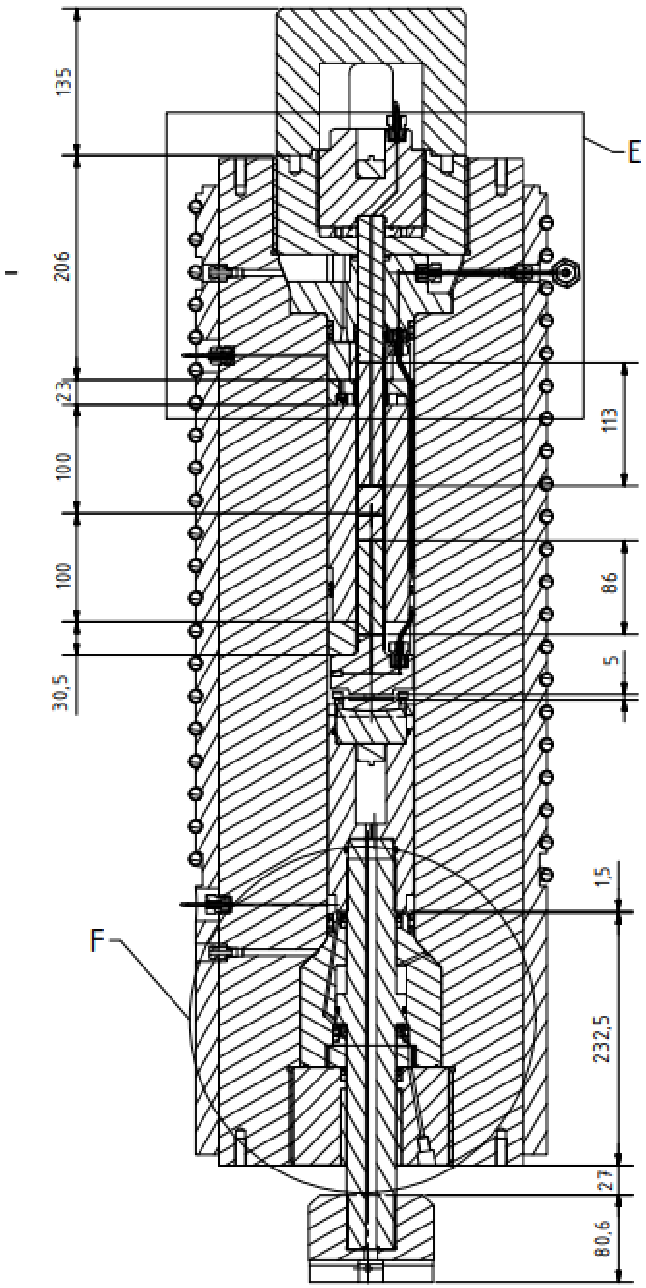

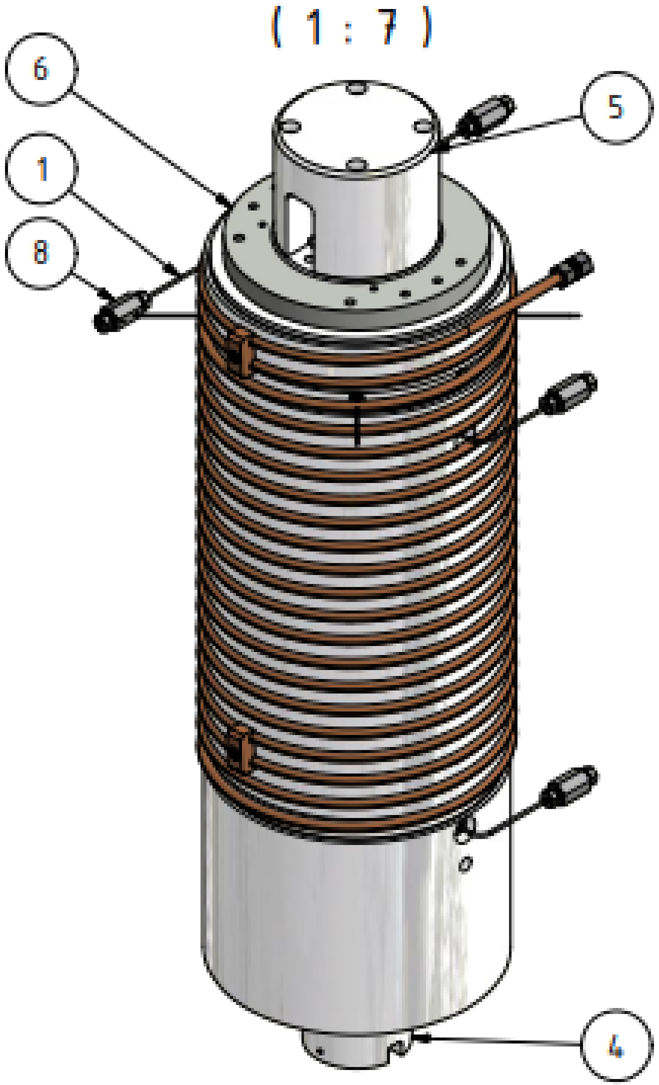

Target EPFL