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MECHANICAL ANALYSIS OF THE LIFTING POINTS FOR THE **VE**RTEX **LO**CATOR (VELO) STAND

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Abstract

The purpose of the mechanical calculation is to investigate the stress and displacements in one of the lifting points of the VELO STAND occur by the weight of the VELO DETECTOR. These lifting points have to comply with the CERN SAFETY CODE [EDMS 335726]. The numerical analysis was done by the *IDEAS* finite element analysis software.

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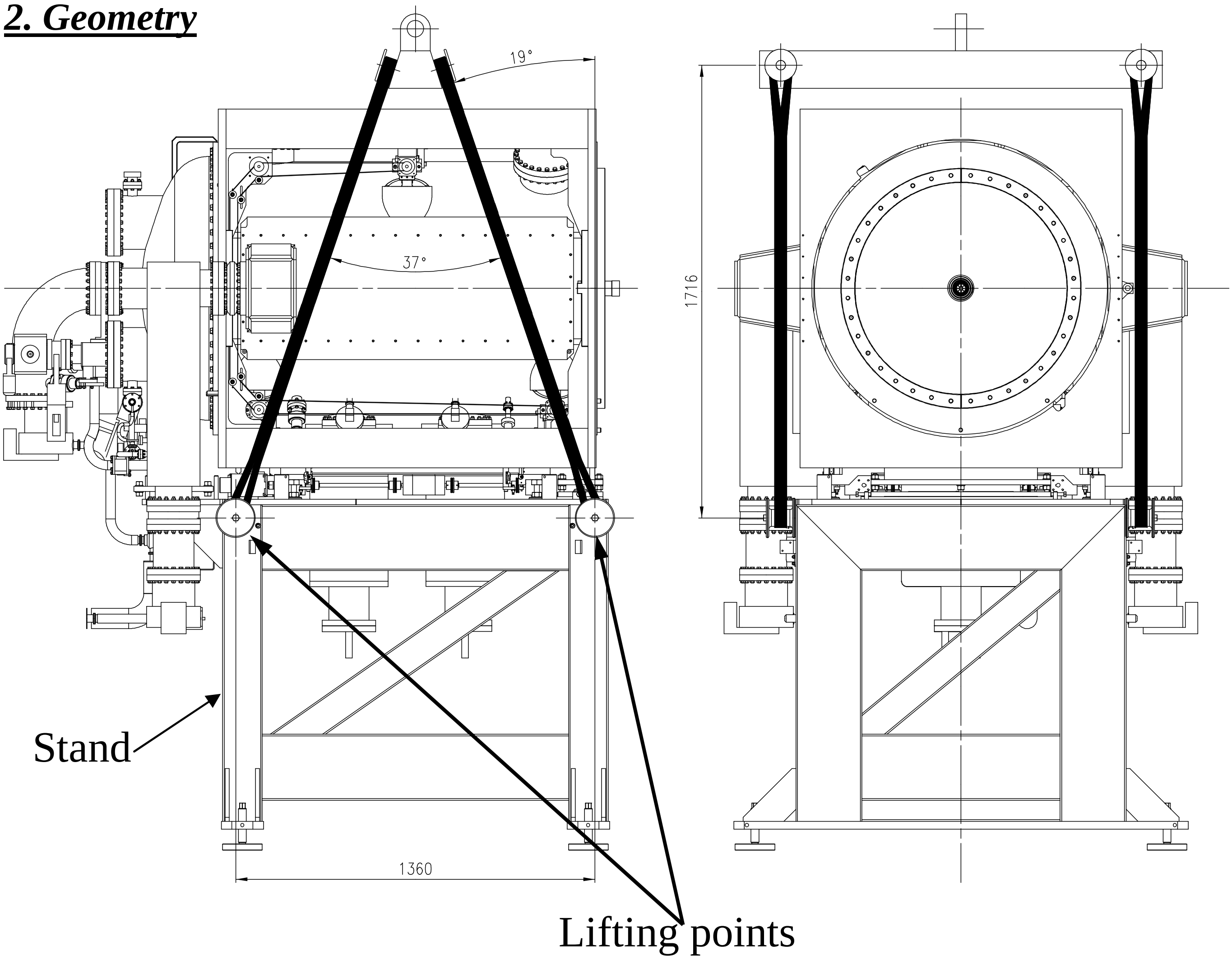
1. Introduction

The VELO will be installes as a pre-assembly in cavern UX85 at point 8. The total mass of the detector, including the stand, is approximately 2,600 kg. On the top corners of the stand are 4 removable lifting points which allow us to lift the complete detector into position. The stand and the lifting rods are made out of AISI 304 stainless steel.
These lifting points have to comply with the CERN SAFETY CODE [EDMS 335726].

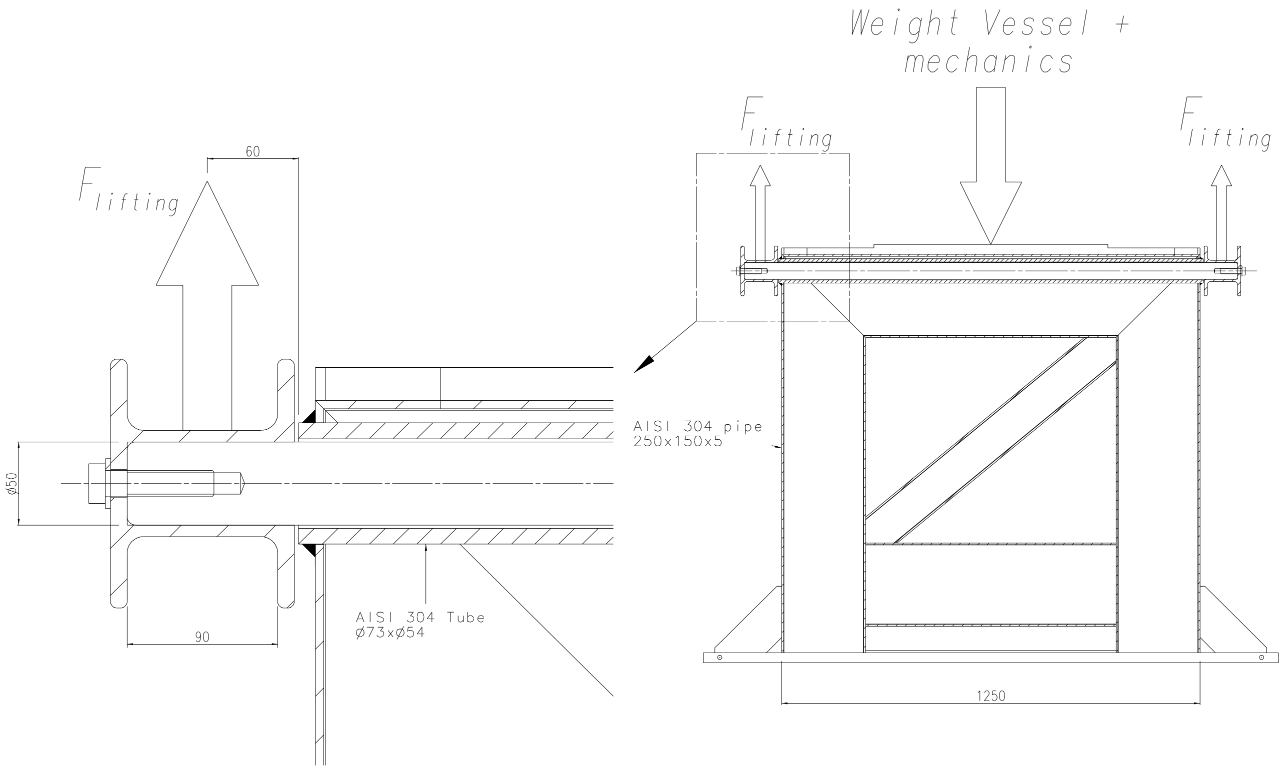
2. Geometry



3. Material properties

Material:	AISI 304
Young's modules:	210 [GPa]
Poisons ratio:	0.33
Yield Strength:	180 [MPa]
Ultimate Strength:	460 [MPa]
Density:	7.85 [g/cm3]

4. Load

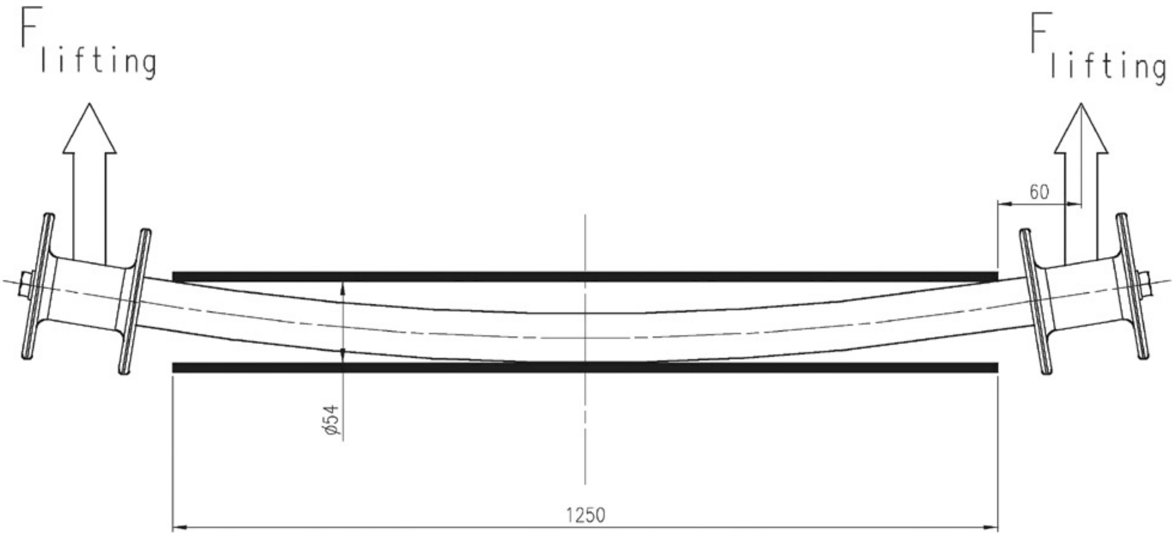


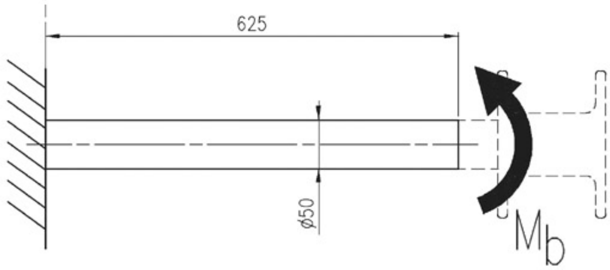
Safety constrains:	Load lifted by only two points (4 points available)		
	Safety factor = **2.4**		
Load(s):	Weight 2 Detectors	=	4,000 N
	Weight Vacuum Vessel	=	10,000 N
	Weight Stand	=	6,000 N
	Weight mechanics	=	6,000 N
		------------------ +	
	Total weight VELO	=	26,000 N

$F_{Lifting}$ = (26,000 [N] /2) cos 19º * 2.4 = ***33,000 N***

5. Lifting Bar

5.1Calculation





Lifting Force:	$F_{lifting} = 33{,}000\ N$
Bending moment:	$M_b = F\ lifting * 60\ [mm] = 1{,}980{,}000\ Nmm$
Moment of inertia:	$I = \pi * d^4 / 64 = 306{,}796\ mm^4$
Moment of resistance:	$W_b = \pi * d^3 /32 = 12{,}272\ mm^3$
Crossing surface rod:	$A = \pi * d^2 / 4 = 1963.5\ mm^2$

Deflection (max):	$f_{max} = (M_b * l^2)/2 * E * I) = 6.2\ mm$
Bending stress:	$\sigma_b = M_b / W_b = 161\ N/mm^2$
Shear stress:	$\tau = F_{lifting} / A = 16.6\ N/mm^2$

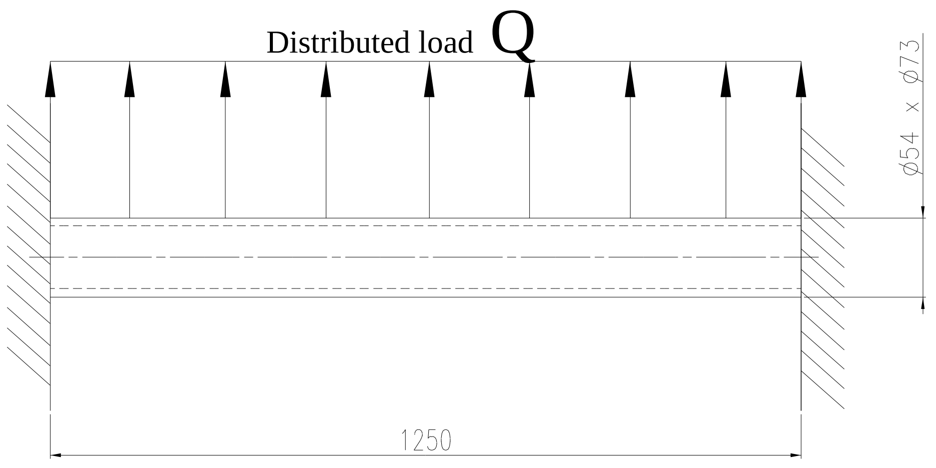
Combined stress:
(according Huber and Hencky)

$$\sigma_v = \sqrt{\sigma_b^2 + 3 * \tau^2} = 163\ N/mm^2$$

6. Lifting Point Stand

6.1 Calculation

The most worse situation is taken to define stresses in the inner tube of the lifting point:



Lifting Force: $F_{lifting} = 33{,}000\ N$

Distributed load: $Q = F_{lifting} / 1250 = 26.4\ N/mm$

Crossing surface tube: $A = \pi * d^2 / 4 = 1{,}895\ mm^2$

Moment of resistance: $W_b = \pi (D^4 - d^3) / 32 * d = 26{,}756\ mm^3$

Bending moment: $M_b = Q * l^2 / 12 = 3{,}437{,}500\ Nmm$

Bending stress: $\sigma_b = M_b / W_b = 128\ N/mm^2$

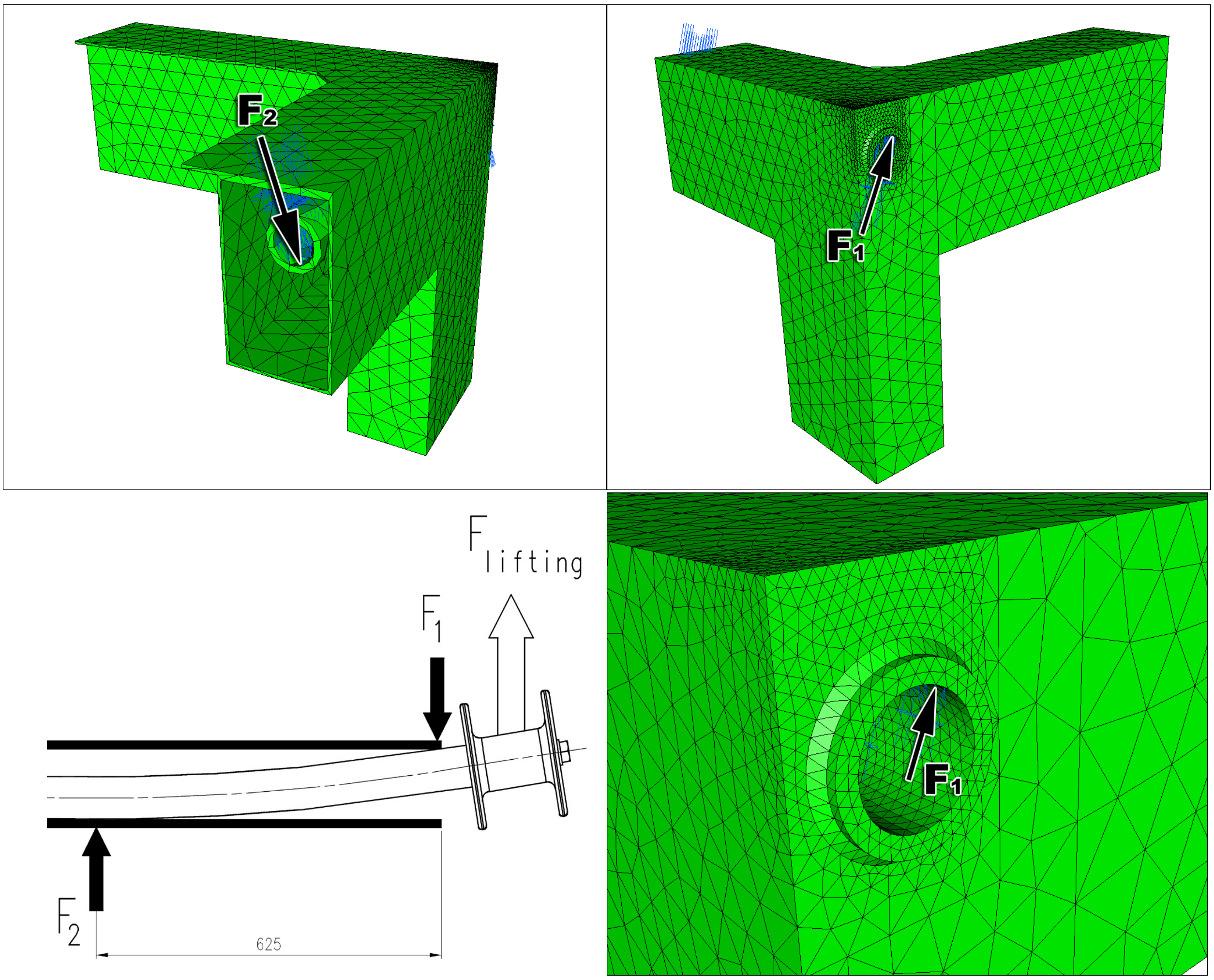
Shear stress: $\tau = F_{lifting} / A = 17.4\ N/mm^2$

Combined stress:
(according Huber and Hencky)

$$\sigma_v = \sqrt{\sigma_b^2 + 3 * \tau^2} = 131\ N/mm^2$$

Drawing of the FEA model

6.2 FEA model



Mesh type(s): 3D Solid parabolic tetrahedron

Load(s):

$$\boldsymbol{F}_1 = \boldsymbol{F}_{lifting} = 33{,}000\ N$$

$\boldsymbol{F}_2$ = *reaction force from the lifting rod* 1)

Moment of inertia: $\boldsymbol{I} = \pi * d^4 / 64 = 306{,}796\ mm^4$

Deflection: $\Delta f = 6.2 - 4 = 2.2\ mm$

$$\boldsymbol{F}_2 = (3 * E * I * \Delta f) / l^3 = 1742\ N$$

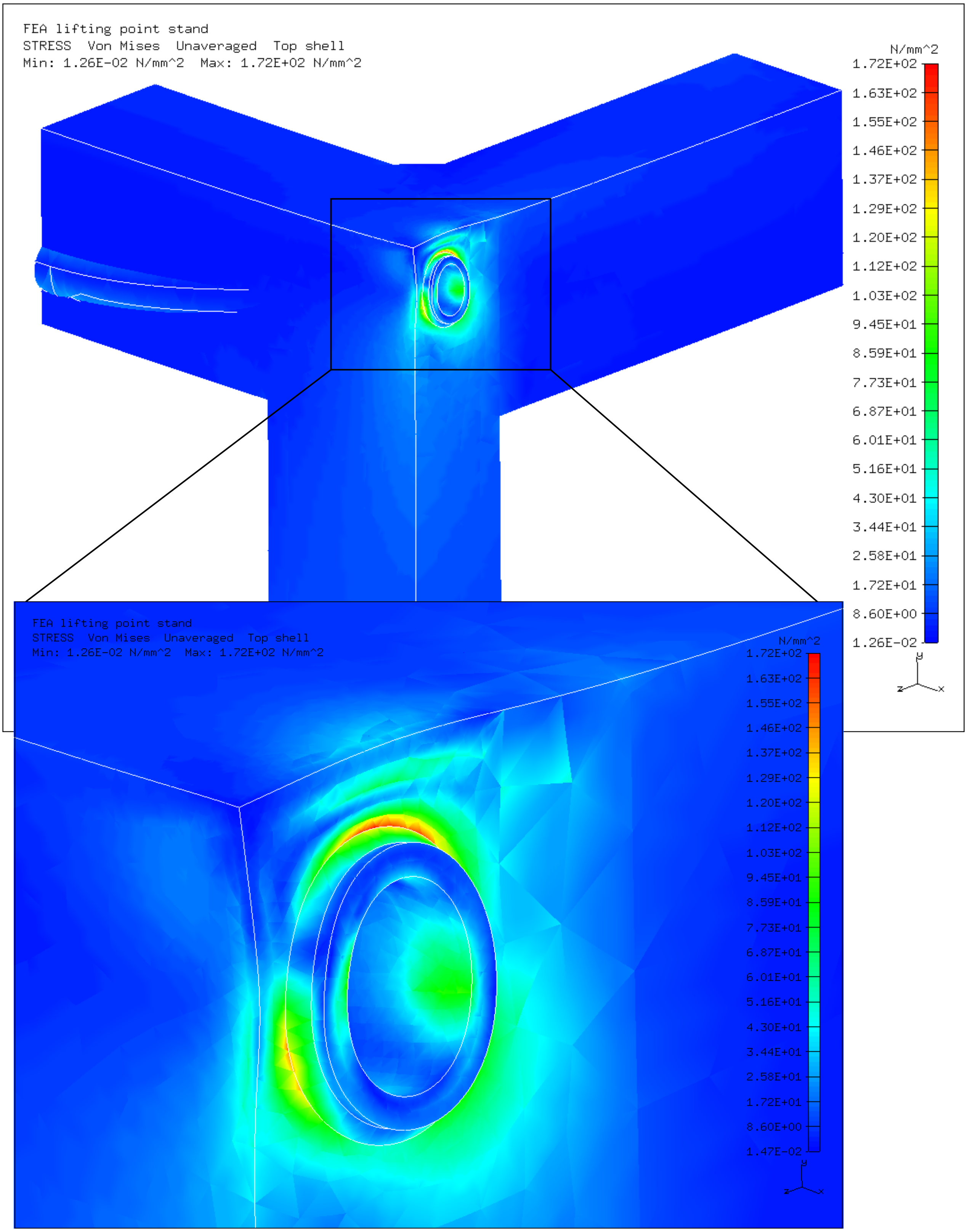
Type of Solution: Linear Statics

Units: Length [mm]; Force [N]; Stress/Pressure [Mpa]

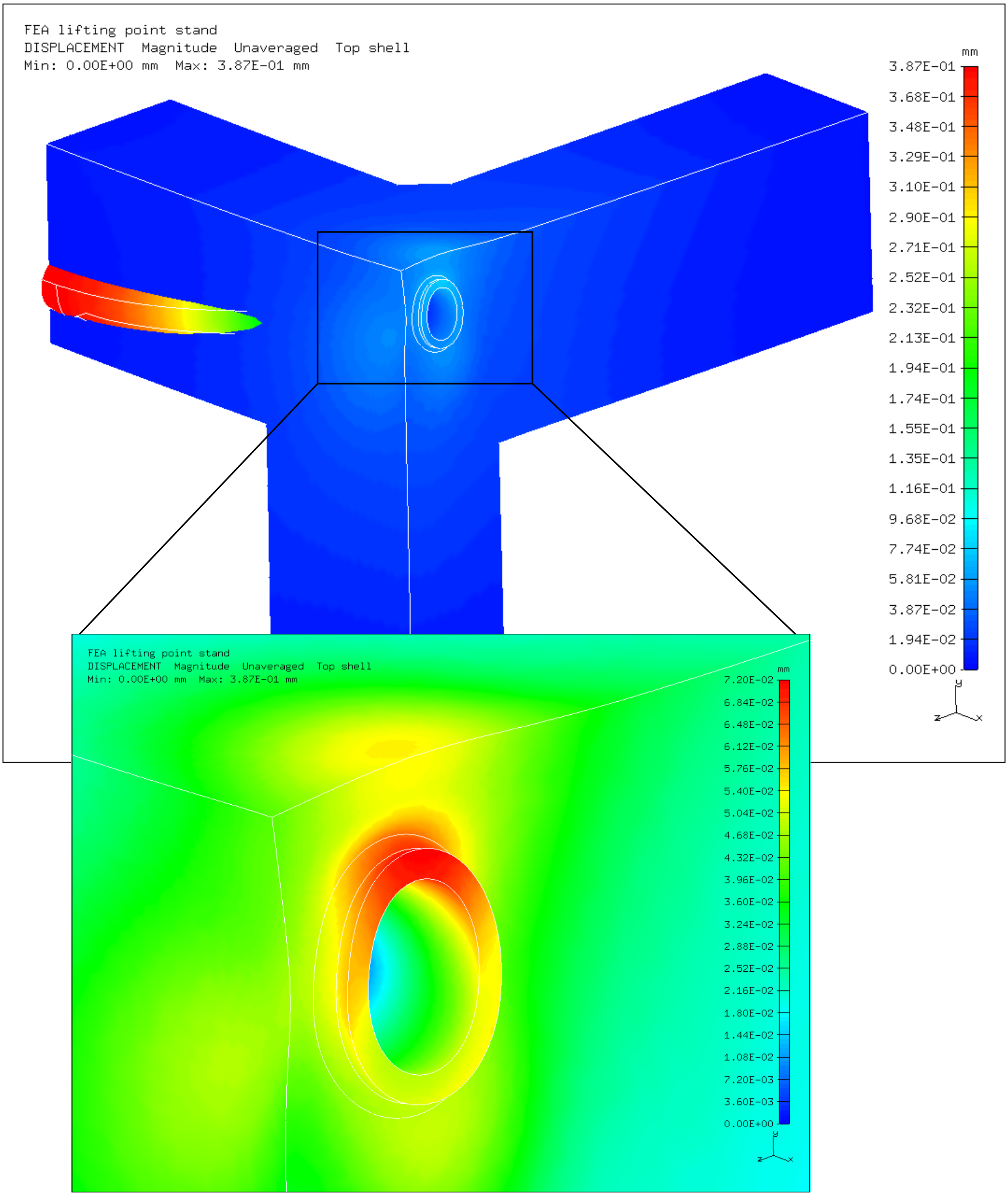
1) Because of the safety constrains, this lifting rod will be deflecting (=6.2mm) more than the free space (=4 mm) in between, so it will cause a force [F_2] on the tube

6.3 FEA Results

STRESS



DISPLACEMENT



6. Conclusion

Max Stress of 172 MPa[1)] is below the yield strength (180 MPa), so still elastic.
A max displacement of 0.38 mm is no problem.

1) Not that for the load, the safety factors are taken into account