ALMA MATER STUDIORUM
UNIVERSITÀ DI BOLOGNA

CHASSIS AND BODY DESIGN MANUFACTURING

DESIGN OF A MOTORBIKE HANDLEBAR BRACKET

Candidate
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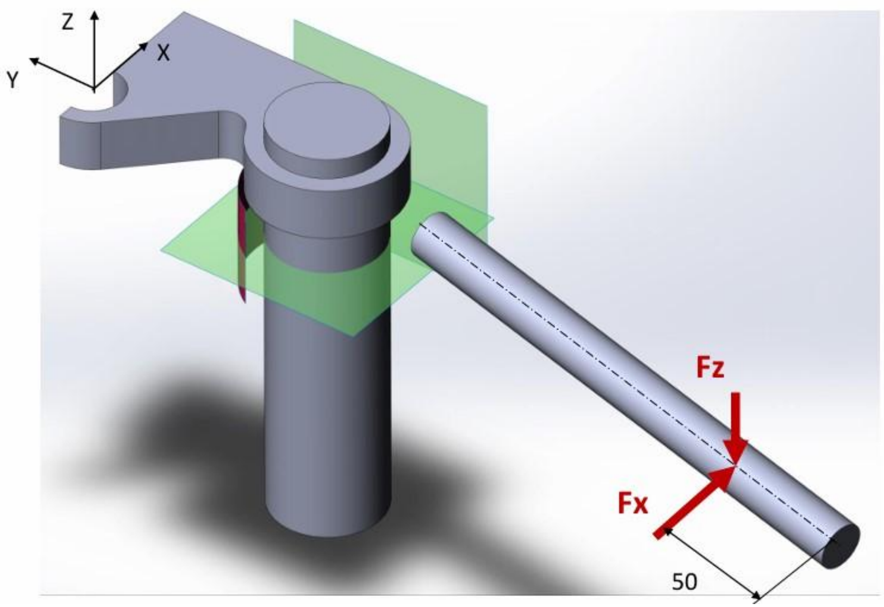
CHASSIS AND BODY DESIGN MANUFACTURING

PROJECT DETAILS

Boundary conditions
The geometry must be contained between the green planes
Fasteners with maximum class of 12.9
10^5 cycles life target



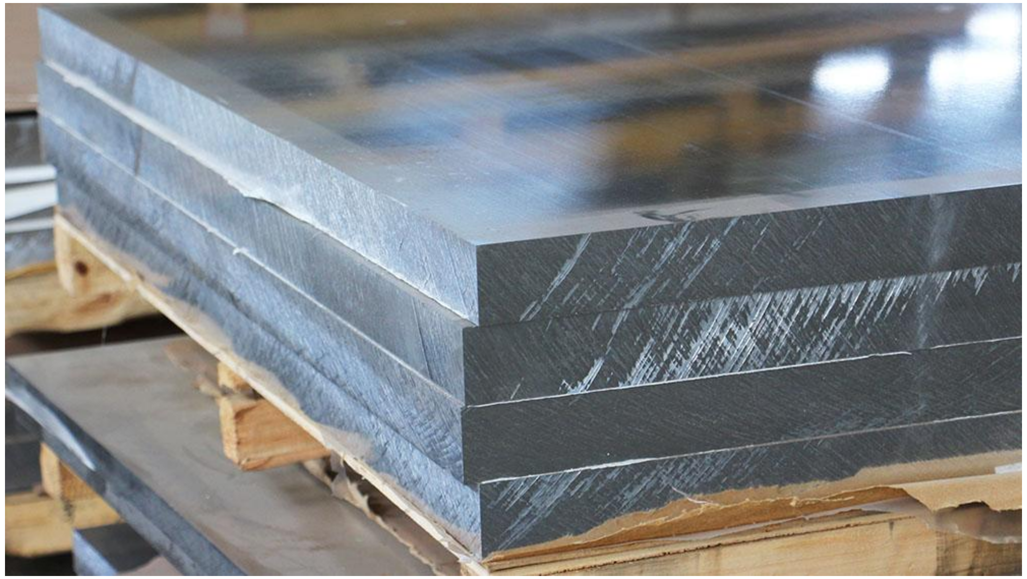
The final **GOAL** is to design the lightest component possible, according to the given limitations.

CHASSIS AND BODY DESIGN MANUFACTURING

ASSIGNED MATERIAL

The material used is the alluminum alloy
EN AW 2024 T3

Property	Value
Density	2.77×10^3 kg/m^3
Young's Modulus	73 100 MPa
Poisson's Ratio	0.33
Yield Strength	345 MPa
Ultimate Strength	483 MPa

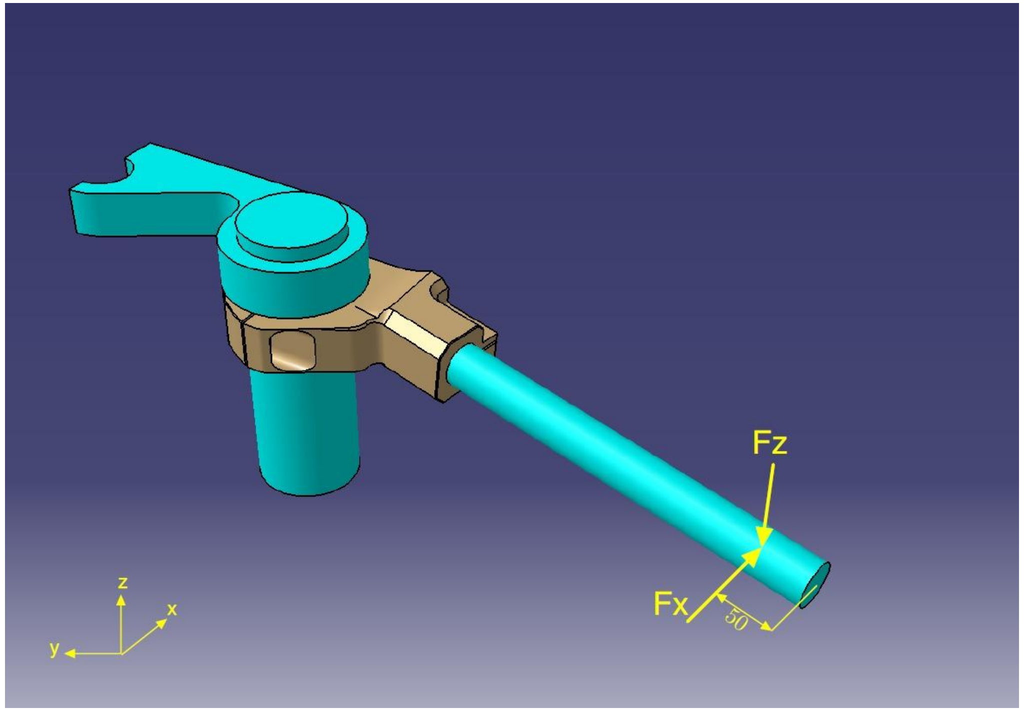


LOADING CONDITIONS

The component is subjected to static and cyclic loads, situated near the extremity of the handlebar.

Load conditions for Maximum Force Analysis	
$F_{z\,max}$	100 N
$F_{x\,max}$	450 N

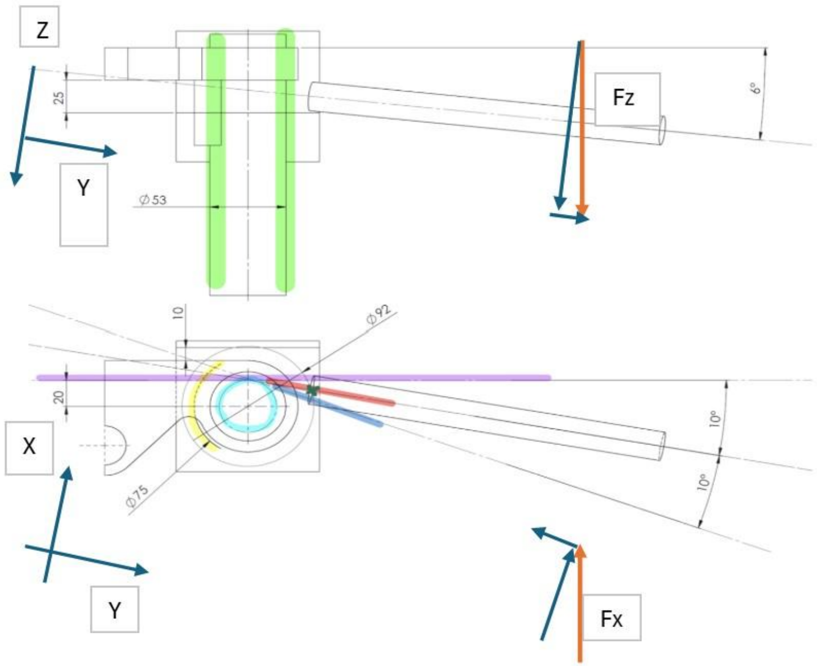
Load conditions for Fatigue Analysis	
F_z	50 N
F_x	±350 N



FORCES RESOLUTION

Tilt Angles
6° downwards
±10° around the vertical axis

Constraint Reactions ZY		Constraint Reactions YX	
Y_a	-10.5 N	Y_a	78.1 N
Z_a	99.5 N	X_a	-443.2 N
$M_{a,zy}$	26 901.8 Nmm	$M_{a,yx}$	-119 875.7 Nmm



BOLT SIZING

- VDI 2230 standard was used to calculate the diameter of the bolt.
- Bolt 10.9 Class, following ISO 4762 was chosen.
- Tightening process done by a torque wrench.

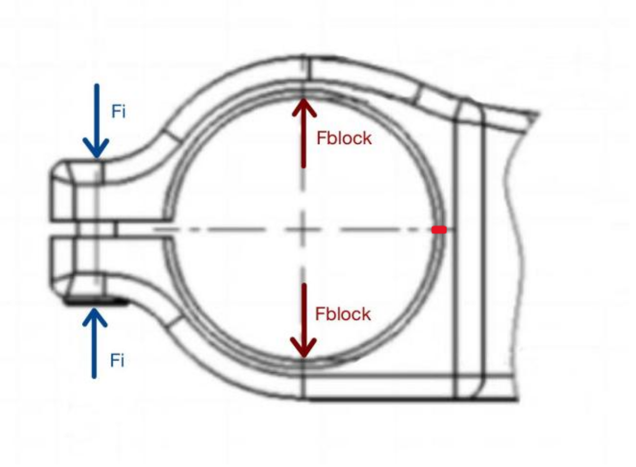
Handlebar BOLT		
F_A	1375,59	N
F_Q	67,69	N
F_Q/μ	173,56	N
Fn	1600,00	N
Fm_{MIN}	4000,00	N
Fm_{MAX}	10000,00	N
d	**6,00**	mm
aA	2,50	-

Fork BOLT		
Fn	2315,97	N
Fq	67,69	N
F_Q/μ	173,56	N
Fn	2500,00	N
Fm_{MIN}	6300,00	N
Fm_{MAX}	10000,00	N
d	**6,00**	mm
aA	1,59	-

CHASSIS AND BODY DESIGN MANUFACTURING

COUPLING PRESSURE BETWEEN HANDLEBAR BRACKET AND BAR

The screw must not only keep the part fixed in place to prevent detachment, but it must also resist bending moments due to the forces on the handlebar.



Results		
F_{block}	3876.6 N	
F_i	1375.5 N	
Design Tightening Torque	3939.1 Nmm	
σ_{eq}	144.8 MPa	$< S_p$ =810 MPa
$A_{tdesign}$	5.16 mm^2	$< A_{tselected}$ = 20.12 mm
σ_{eq_o}	141.2 MPa	

For a more realisic result we assumed a weight of the pilot of 70 kg and a CS of 1,5.

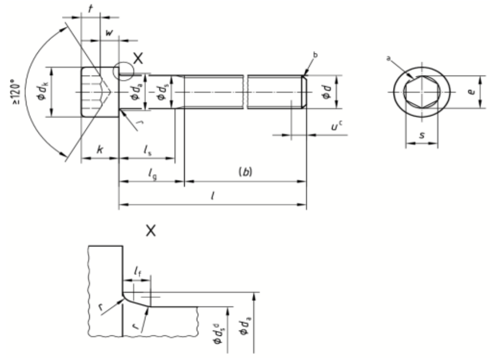
COUPLING PRESSURE BETWEEN HANDLEBAR BRACKET AND FORK

Results		
F_{block}	10749.5 N	
F_i	2315.9 N	
Design Tightening Torque	7 395.2 Nmm	
σ_{eq}	243.7 MPa	$<S_p$ =810 MPa
$A_{tdesign}$	8.18 mm^2	$<A_{tselected}$ = 20.12 mm
σ_{eq_o}	158.3 MPa	

The methodology used for calculating the various parameters are the ones of the previous case.

CALCULATION

Coupling Pressure between Handlebar and Fork			
Ultime Tensile Strength S_u	1000,00	MPa	
Yield Strength S_y	900,00	MPa	
Resistent Area A_t	20,12	mm^2	
Resistent Diameter d_t	5,06	mm	
Metric Nominal Diameter d	6,00	mm	M6
Friction coefficient of thread μ_{th}	0,39	-	
Friction coefficient of Bolt head μ_b	0,39	-	
Screw head diameter d_k	10,00	mm	ISO 4762-2004
Underhead Diameter d_w	9,38	mm	
Screw Pitch p	1,00	-	ISO 4762-2004
Pitch Diameter d_2 min	5,21	mm	ISO 965-2
Pitch Diameter d_2 max	5,32	mm	ISO 965-2
d_2	5,35	mm	
Min Major Diameter d_{min}	5,97	mm	ISO 965-2
Max Major Diameter d_{max}	5,79	mm	ISO 965-2
Average Major Diameter d_{mean}	5,88	mm	
Torsional Modulus W_t	12,73	mm^3	
Preload on each Bolt F_i	2315,97	N	
F_{block}	10749,58	N	
Number of Bolts	2,00	-	
Design Tightening Torque	7395,16	Nmm	
Sigma under tightening condition σ_{eq}	243,75	MPa	$< S_p = v{*}S_y$
Stress Ratio SR	2,86	Steel-All	
$A_{t\_minimum}$	8,18	mm^2	$< A_{t\_selected}$
Sigma under operating condition σ_{eq_o}	158,35	MPa	
Underhead Loading R_n	100,00	N	
Coefficient of Torsional Relaxation k_t	0,50	-	



$$A_{t\_design} \cong \frac{SR \cdot F_i}{S_p}$$

$$SR = \frac{\sigma_{eq}}{\sigma_{axial}} = \frac{S_p \cdot A_t}{F_i}$$

$$A_{t\_selected} \geq A_{t\_design}$$

Design assessment under tightening condition

$$\sigma_{eq} = \sqrt{\left(\frac{F_i}{A_t}\right)^2 + 3 \cdot \left(\frac{M_{shank}}{W_t}\right)^2} = \sqrt{F_i^2 + 48 \cdot \left(F_i \cdot \frac{(0,16 \cdot p + 0,58 \cdot \mu_{th} \cdot d_2)}{d_t}\right)^2} \cdot \frac{1}{A_t} \leq \nu R_{p0.2} = S_p$$

Design assessment under operating condition Screw

During operation, the screw may undergo additional axial loads due to applied external loads R_n while the shank torsion is reportedly lower than during tightening. German standard VDI 2230 and Bickford report that such a decrease in torsional stress can be as high as 50% when R_n and R_t are static, whereas a 100% reduction can take place in the case of dynamic loading. This reduction is due to mutual torsional sliding occurring between the screw head and underhead.

$$\sigma_{eq-o} = \sqrt{\left(\frac{F_b}{A_t}\right)^2 + 3 \cdot \left(\frac{M_{shank}}{W_t}\right)^2} \cong \sqrt{(F_i + R_n \cdot 0,25)^2 + 48 \cdot \left(F_i \cdot k_t \cdot \frac{(0,16 \cdot p + 0,58 \cdot \mu_{th} \cdot d_2)}{d_t}\right)^2} \cdot \frac{1}{A_t}$$

UNDERHEAD LOADING $k_T \geq 0,5$

k_t coefficient of torsional relaxation of the screw ($0 \leq k_t \leq 0,9$)…see next slides

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CALCULATION

Stiffness Calculations Handlebar_Fork Bolt		
Shank length	2,10	mm
Nominal Diameter	6,00	mm
An	28,27	mm^2
Engage length	5,00	mm
Not engaged length	10,00	mm
Underhead Diameter	9,38	mm
Eal2024	73100,00	MPa
Ebolt	210000,00	MPa
Head Stiff.	89064151,73	N/mm
Shank Stiff.	2827433,39	N/mm
Thread Stiff.	81409933,27	N/mm
Engaged Thread Stiff.	8140993326,60	N/mm
Al Stiff engaged	31002807,10	N/mm
Equivalent Compliance	3,97283E-07	mm/N
Equivalent Bolt Stiffness	2517094,27	N/mm
Kplate	2197372,51	N/mm
Kplate/Kbolt	0,87	-
LoadFactor	0,47	-

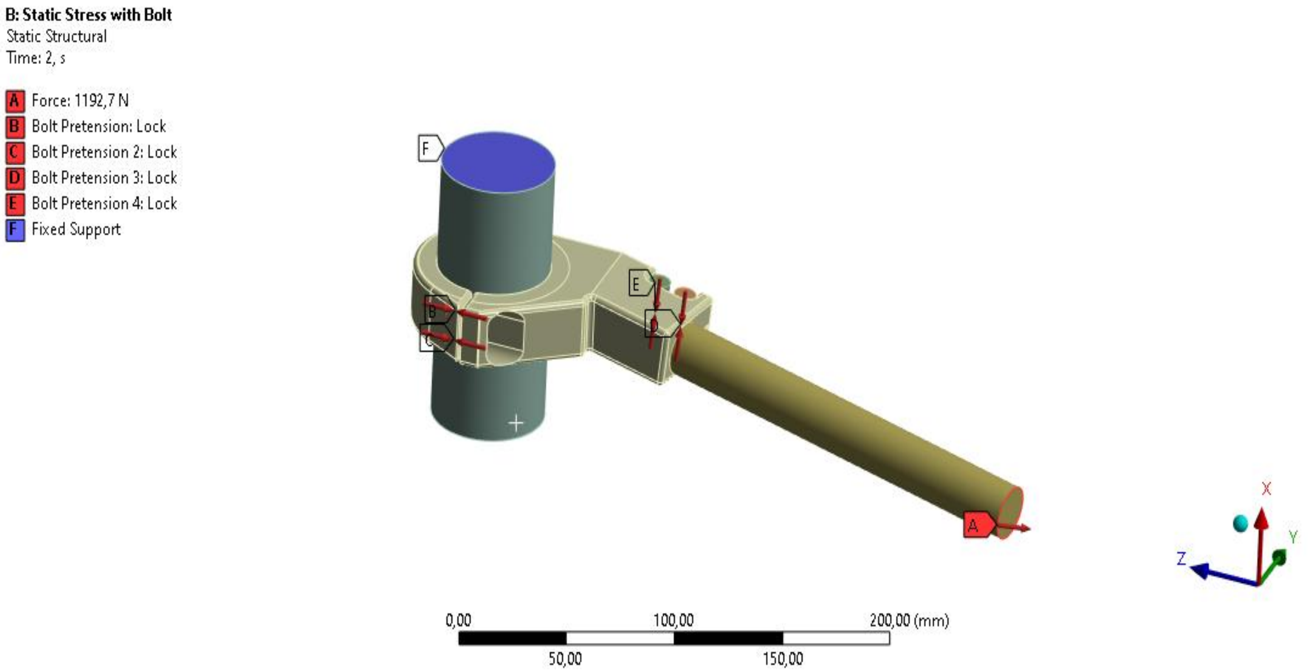
Minimum Engaged length Bolt Handlebar_Bar		
Rs	1,391	-
C1	1,000	-
C3	0,897	-
Tau_bm	272,600	MPa
Pitch	1,000	-
Rm	470,000	Mpa
d	6,000	mm
D1	5,5	mm
D2	5,153	mm
As	20,123	mm^2
m_eff	2,875	mm

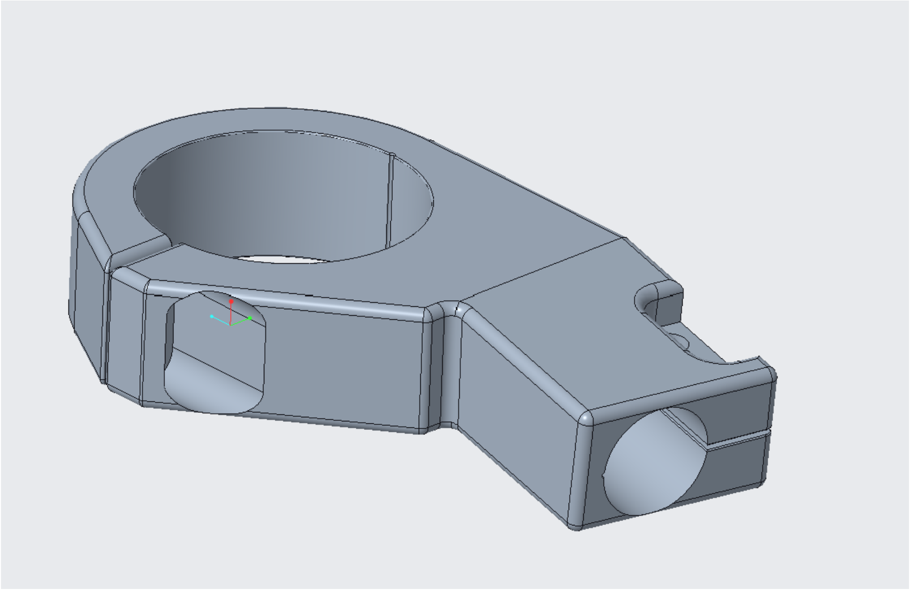
Surface Pressure					
Handlebar_Bar			Handlebar_Fork		
Underhead Diameter dw	9,38	mm	Underhead Diameter	9,38	mm
Coupling Diameter ds	3,82	mm	Coupling Diameter	5,82	mm
Underhead Area	24,28	mm^2	Underhead Area	9,95	mm^2
Underhead Pressure	56,66	MPa	Underhead Pressure	232,67	MPa

Following the VDI 2230 standard, bolt assessment was conducted.

M6 x 1 was selected.

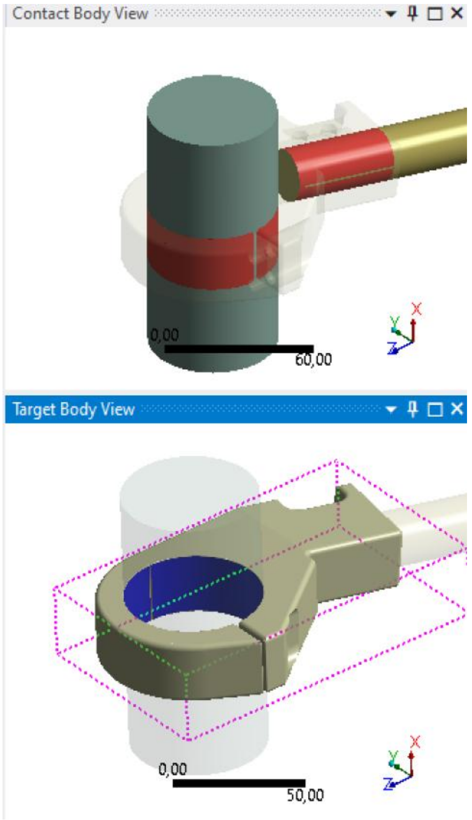
PRELIMINARY DESIGN ANALYSIS

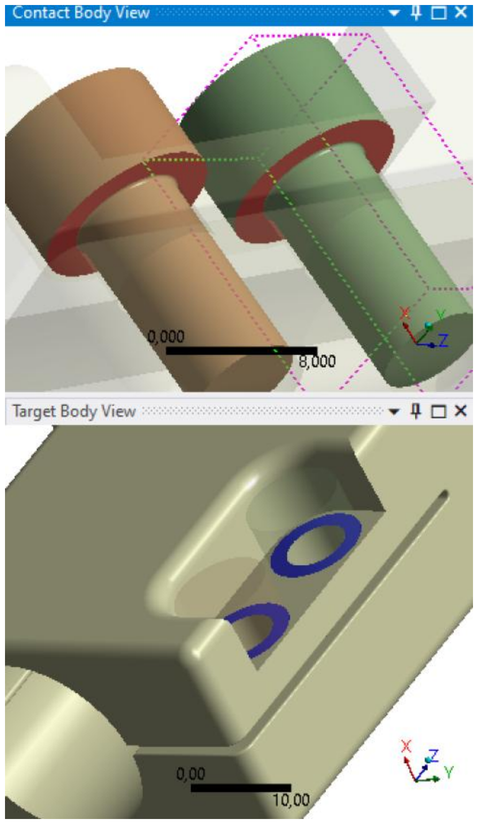


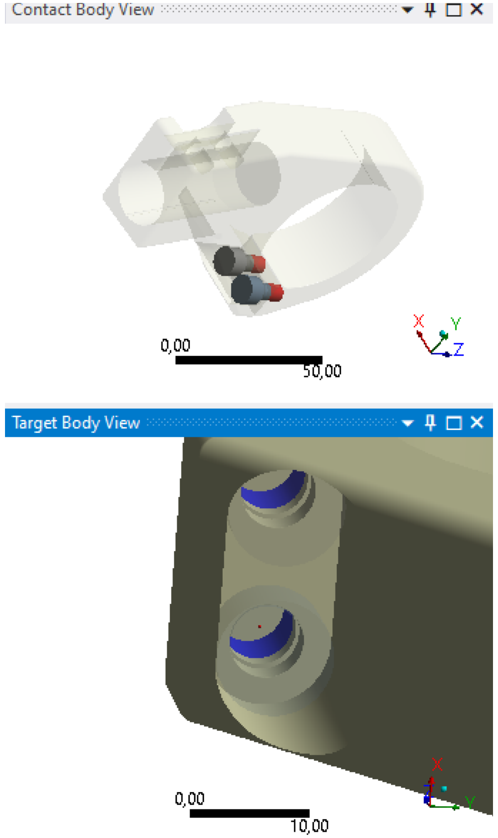


0,278 Kg

FEM - CONTACT

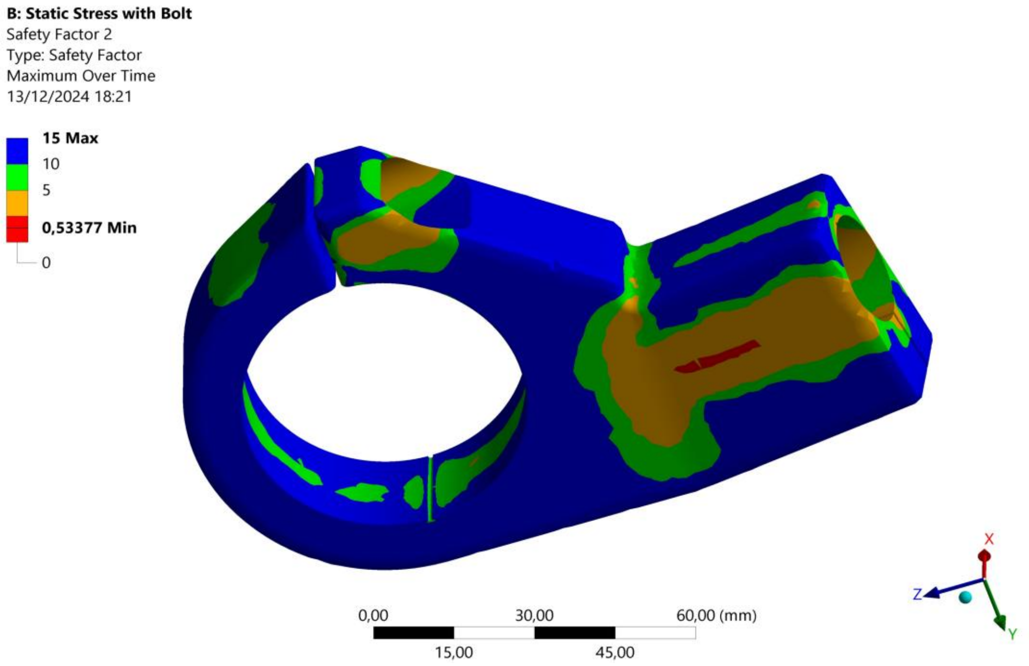


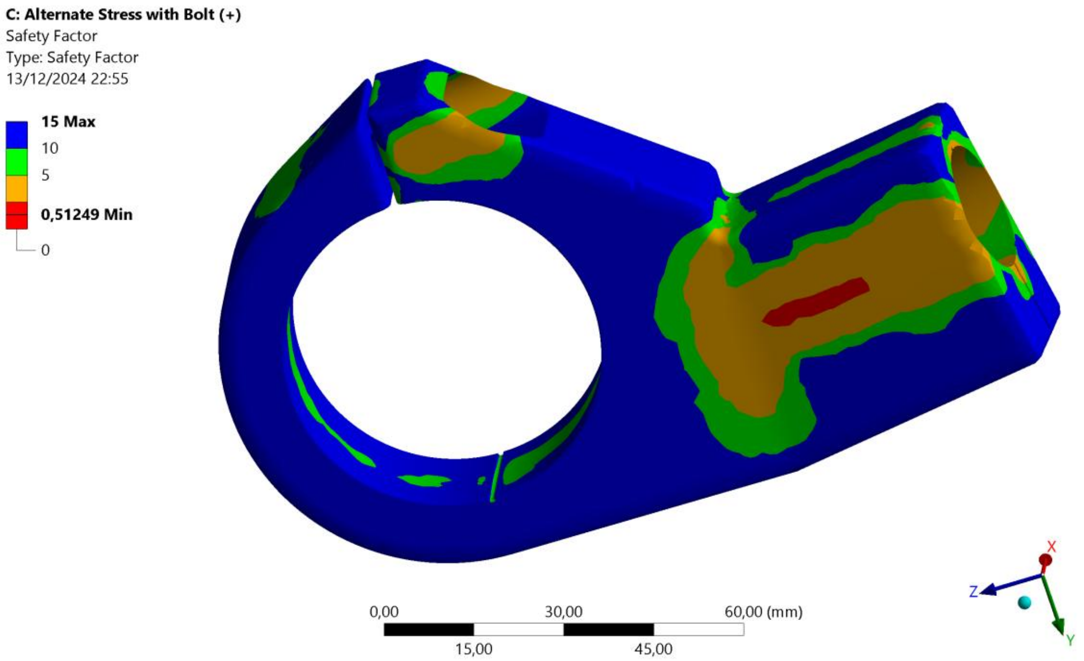




Friction coefficients	
Thread	0.39
Bolt underhead and handlebar bracket	0.39
Fork and handlebar	0.2

PRELIMINARY DESIGN RESULT





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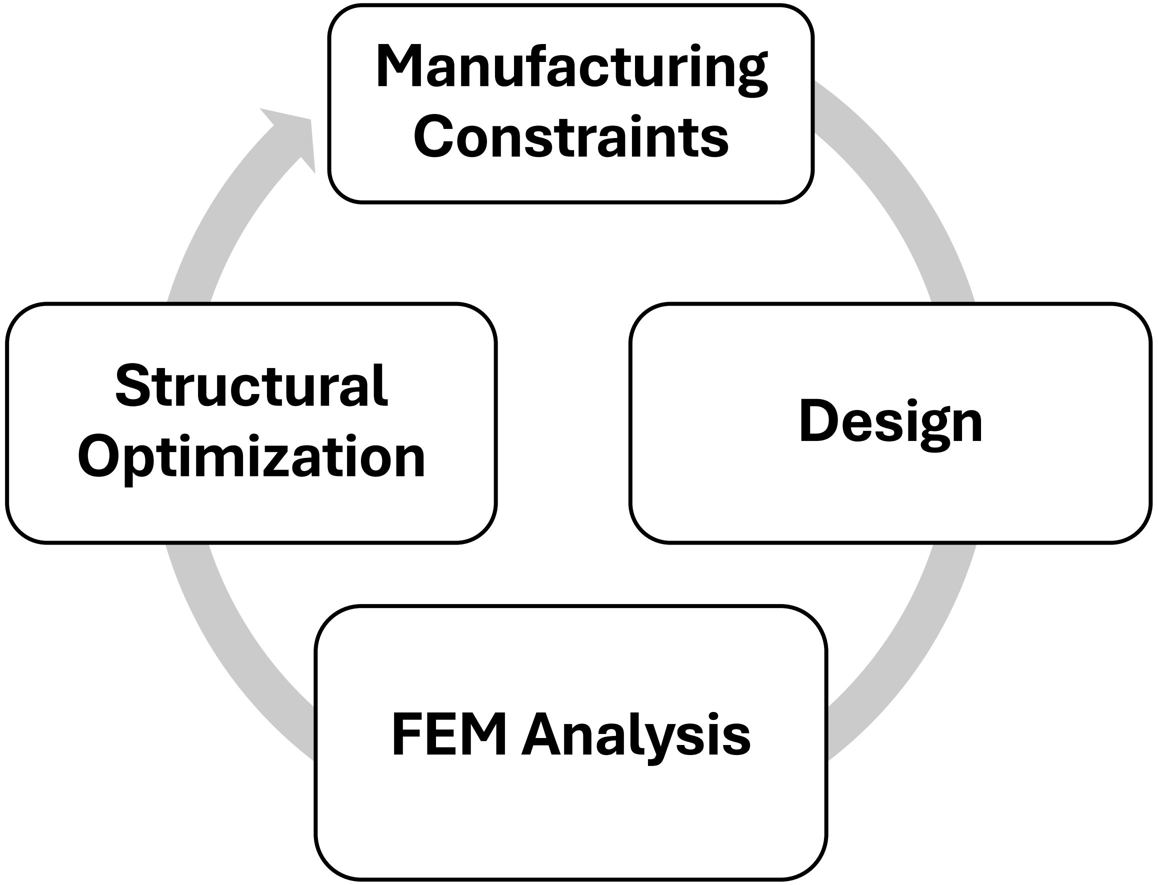
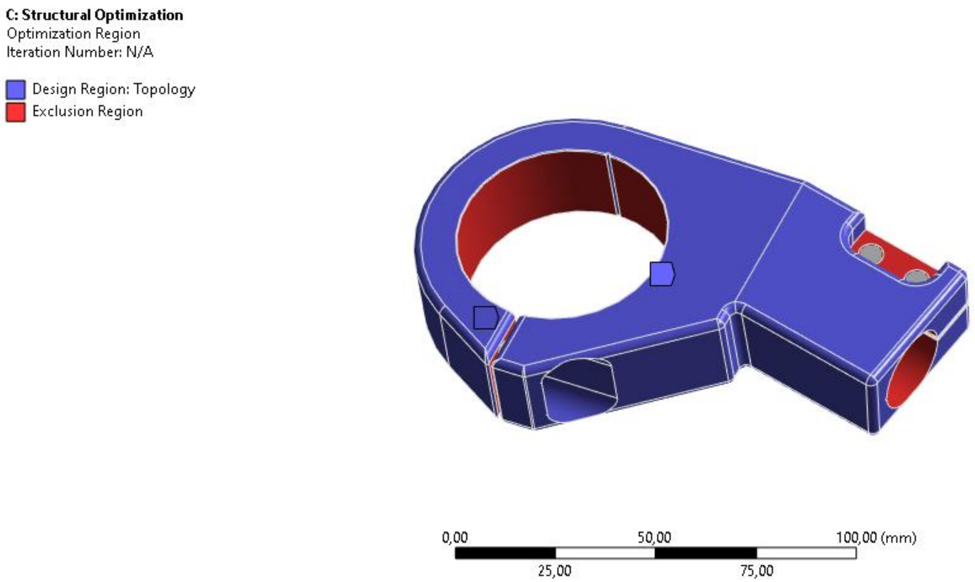
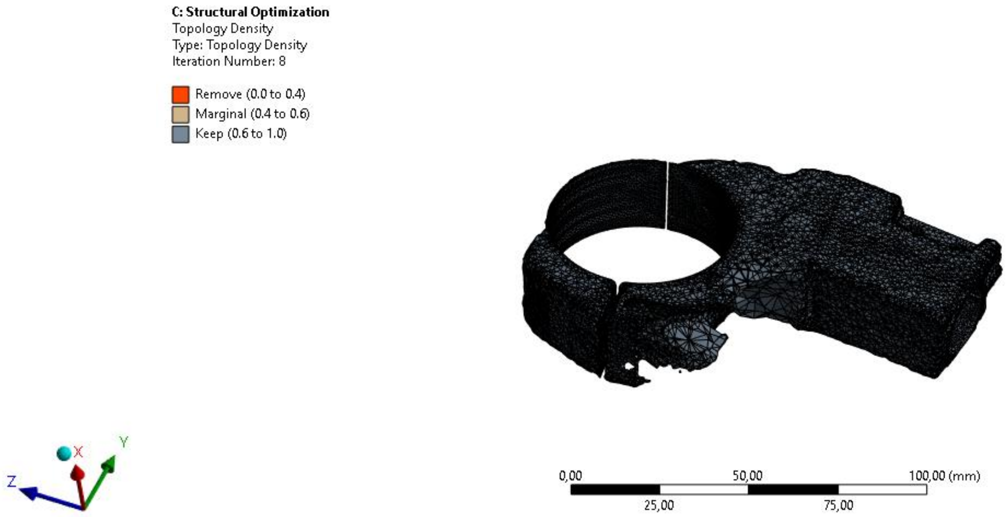
WORKFLOW



OPTIMIZATION

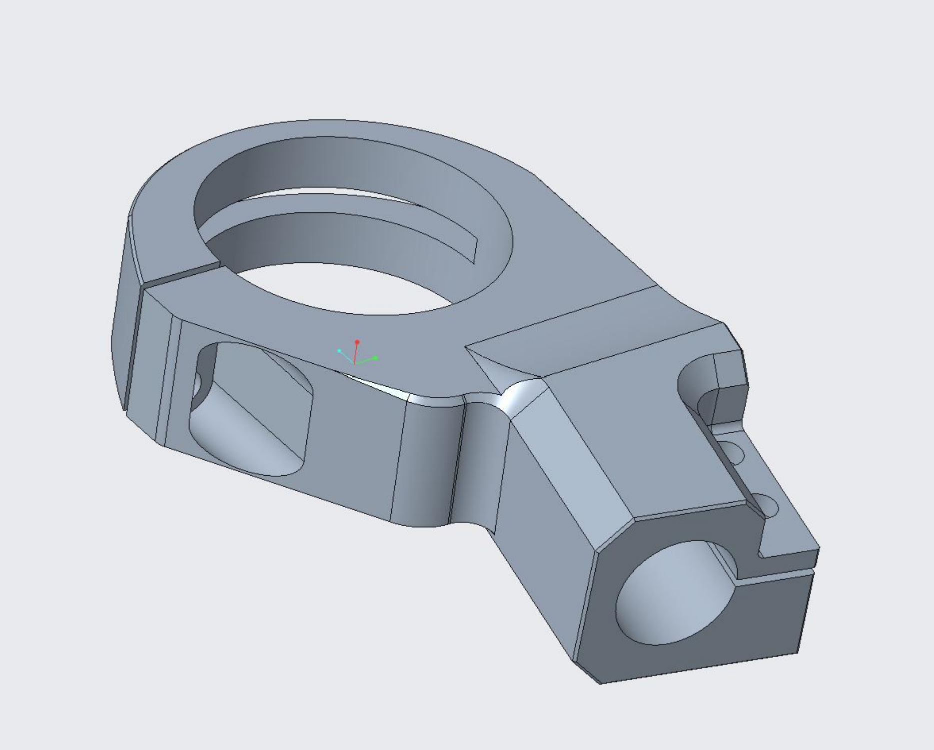




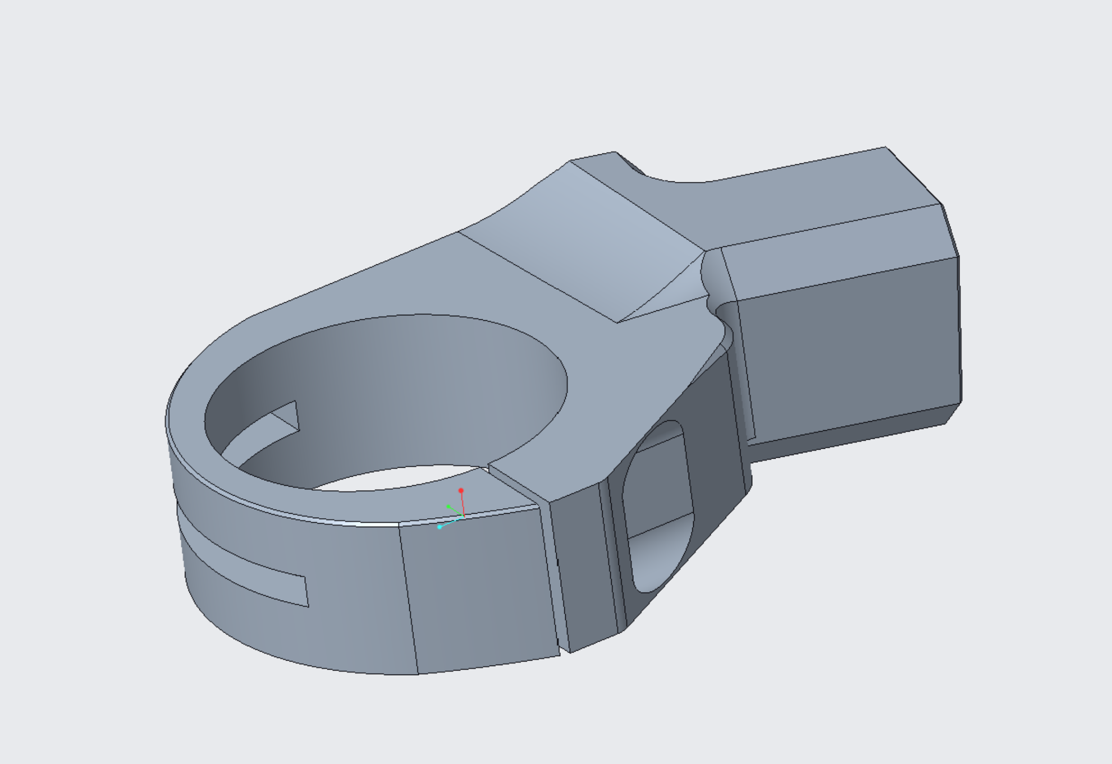
Structural Optimization Tool removed the less stressed material, leaving mass in the most stressed areas.

CHASSIS AND BODY DESIGN MANUFACTURING

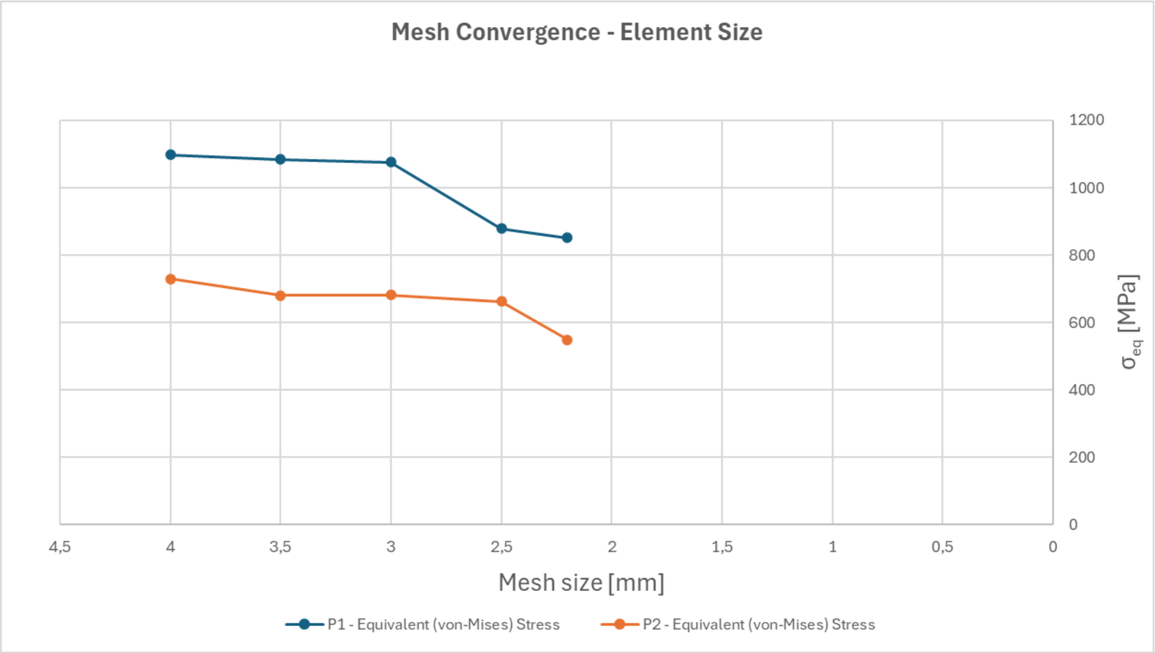
FINAL DESIGN



0,271 Kg

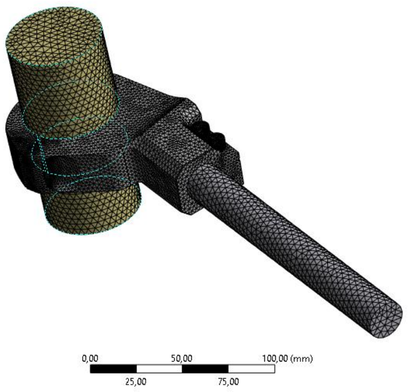


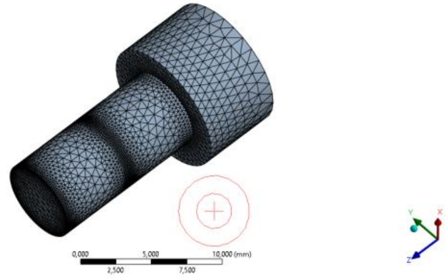
MESH CONVERGENCE



After some mesh refinements iterations, general 2,5mm mesh size was the best choice for validation FEM.

The most relevant, including the bolts', were meshed with 1 mm mesh size, plus others refinements.





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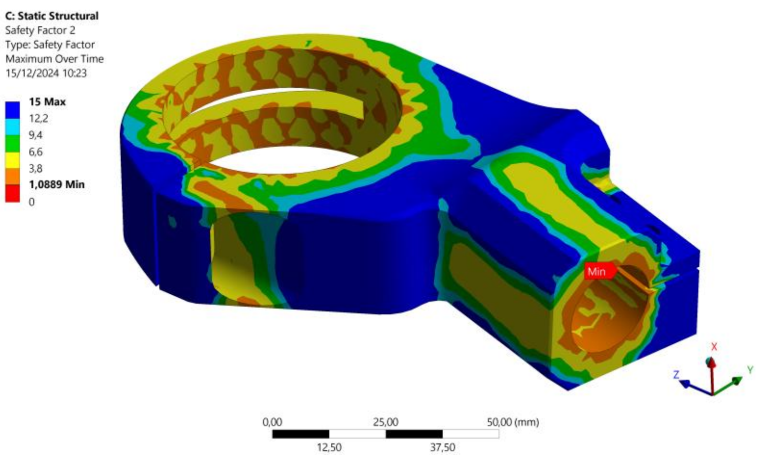
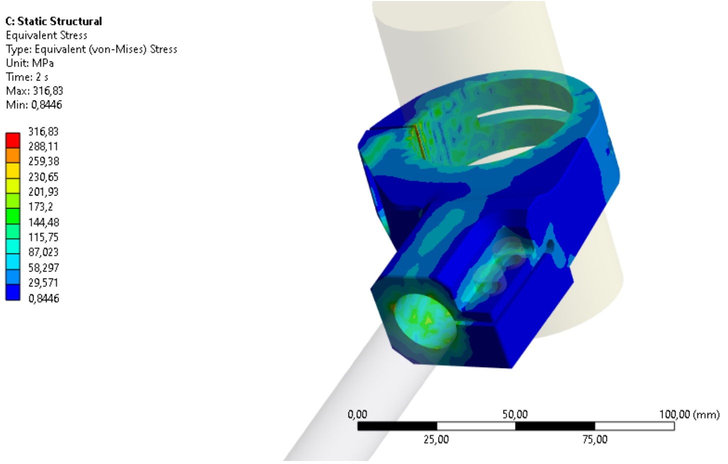
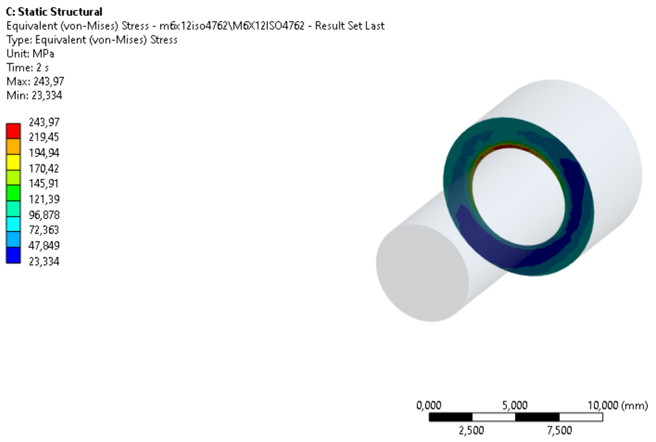
FEM RESULTS



Static Max Loads **SF**



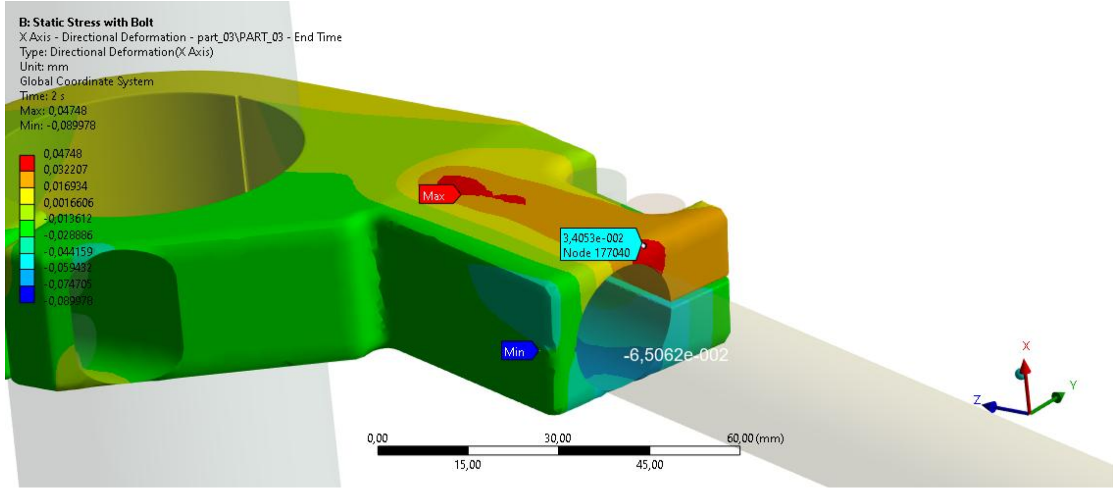
Equivalent Von Mises Stress σ_{eq}

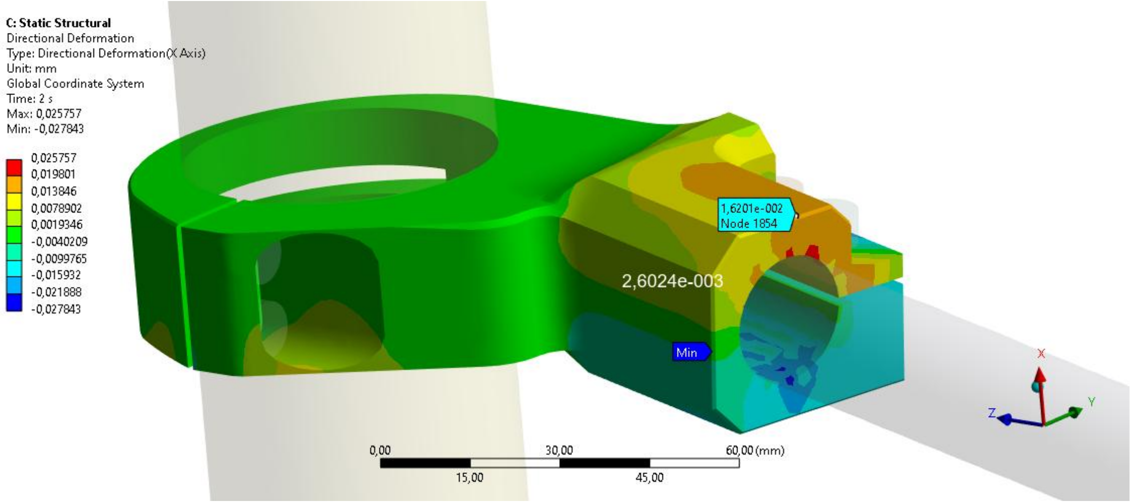


Underhead Pressure

CHASSIS AND BODY DESIGN MANUFACTURING

OLD VS NEW DESIGN COMPARISON

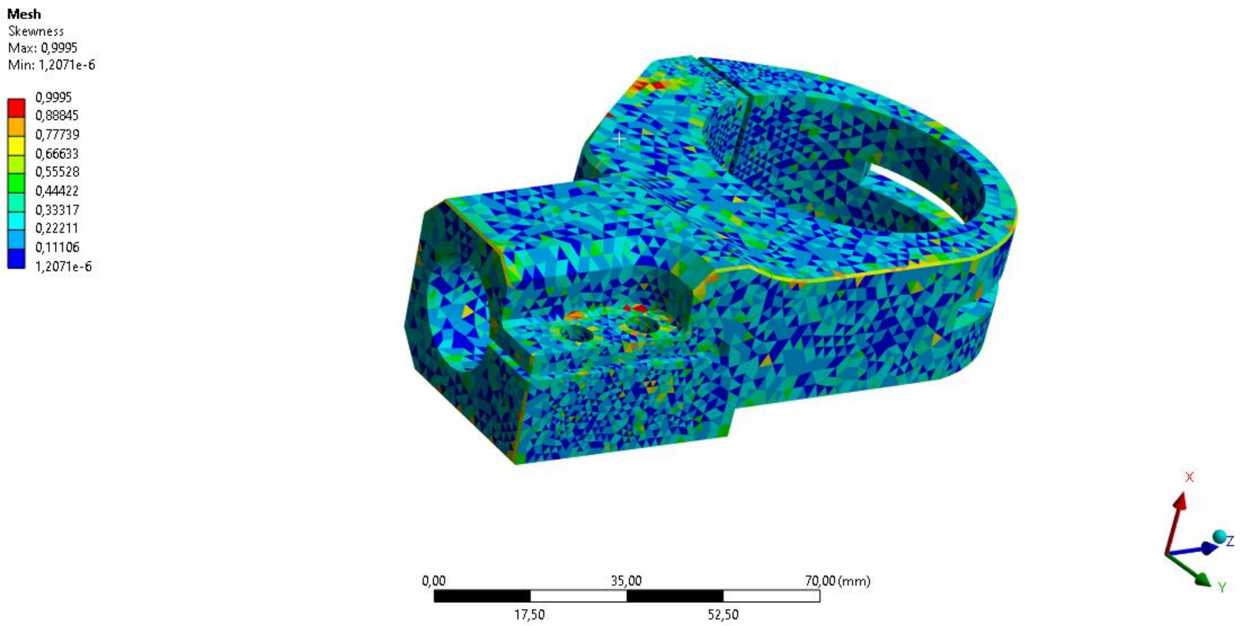


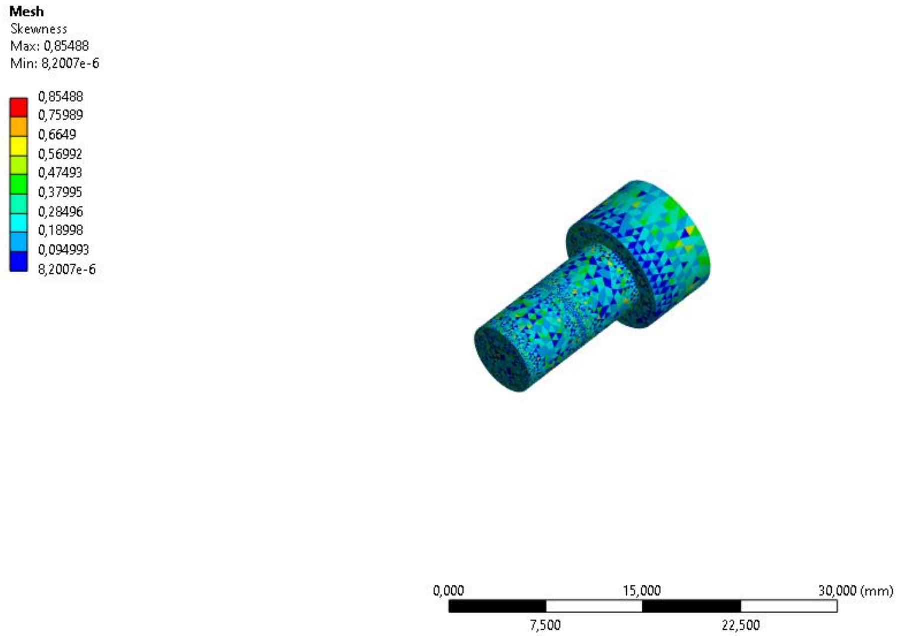


Weight	0,278 Kg	0,271 Kg	-3%
Compliance	0,034 mm	0,016 mm	- 47%

FEM RESULTS ACCURACY

Mesh skewness leads to stress singularities in some areas.
This problem must have to be taken into account in the post-processing phase.





CHASSIS AND BODY DESIGN MANUFACTURING

FATIGUE ANALYSIS - BOLT

Fork_Handlebar Bolt					
Load introduction factor			Alternate stresses		
Φk	0,42		Φn	0,03	
Joint type	SV 6		d	6	mm
h	20	mm			
a_k	5	mm			
l_A	5	mm			
l_A/h	0,25				
a_k/h	0,25		σ_{asv}	59,5	MPa
n	0,06				

Bar_Handlebar Bolt	σ_a	
Front Bolt	0,12	Mpa
Rear Bolt	0,07	Mpa
	σamax < σasv	

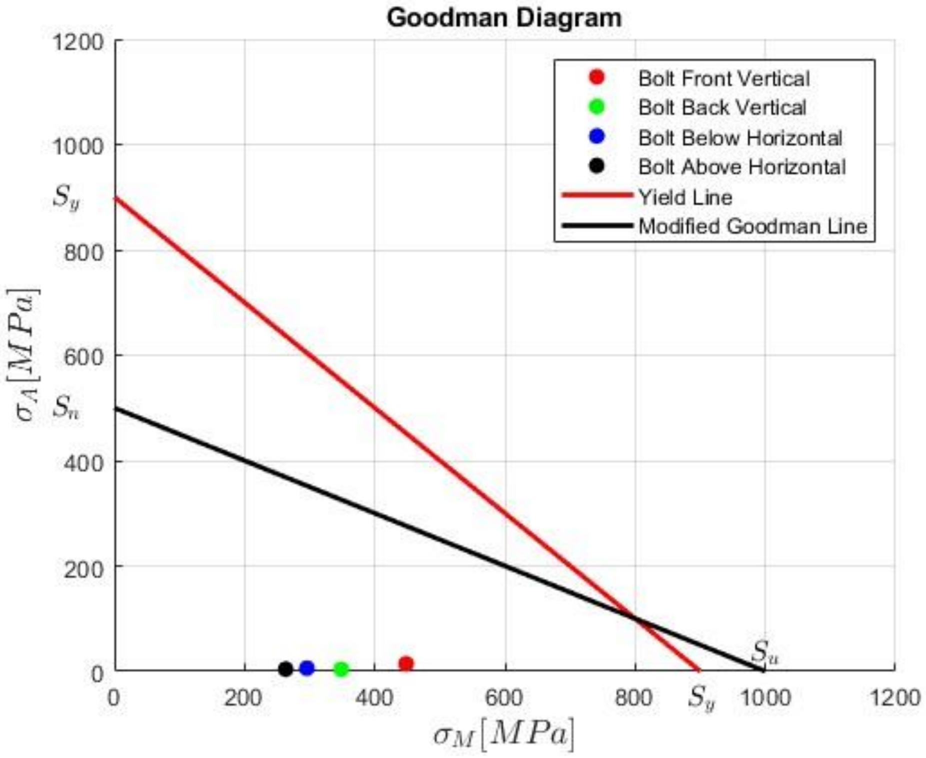
Fatigue calculations for Fork_Handlebar connection.

Fatigue analysis was done on each bolt following VDI 2230.

Since σ_{asv} >> σ_a the bolt meets the requirements.

FATIGUE ANALYSIS - BOLT

GOODMAN DIAGRAM



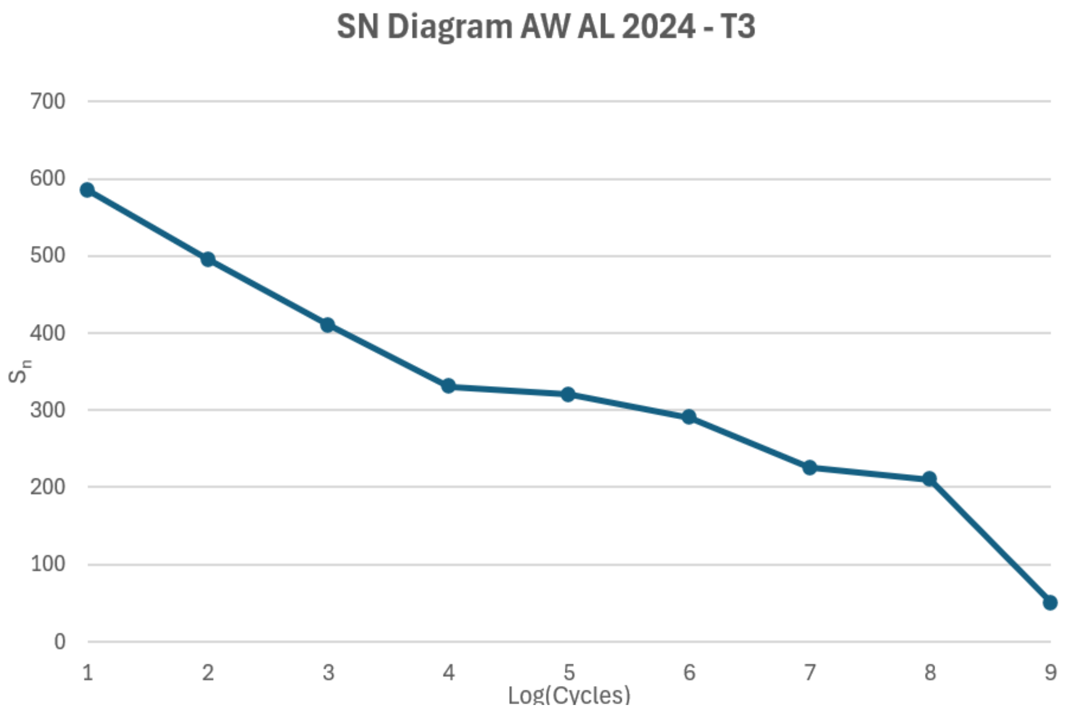
Extracting Equivalent Sigma from FEM results, on thread bolt surfaces, Goodman diagram was applied.

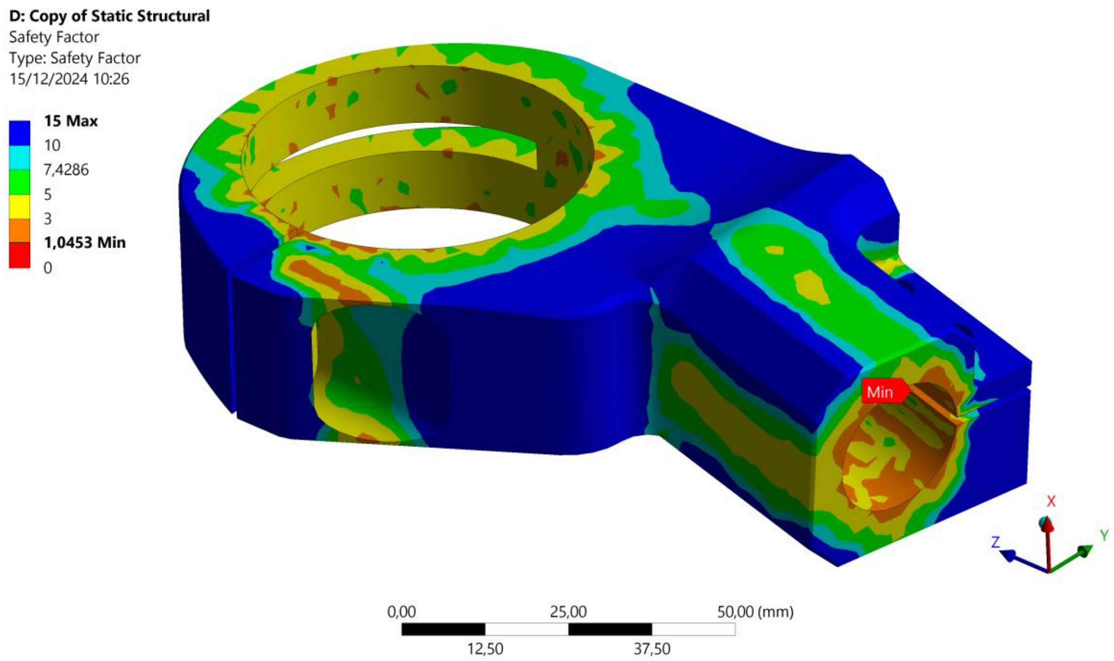
Results were plotted using Matlab script.

Validation Fatigue - Goodman Diagram					
	σ_-	σ_+	σ_A	σ_M	
Front Vertical Bolt	434,05	462,3	14,125	448,175	MPa
Back Vertical Bolt	352,15	344,73	3,71	348,44	MPa
Lower Horizontal Bolt	289,8	301,44	5,82	295,62	MPa
Upper Horizontal Bolt	267,26	259,03	4,115	263,145	MPa

FATIGUE ANALYSIS – HANDLEBAR BRACKET

SN DIAGRAM





CONSIDERATIONS AND IMPROVEMENTS

- Data from FEM analysis were carefully reviewed to understand singularities and mesh errors.
- A more resistant material, could be an excellent improvement for saving more weight.
- CNC manufacturing is a significant manufacturing constraint, that limits the Topological Optimization capabilities.
- From another perspective, CNC manufacturing is a more proven and well-tested production method.

THANKS FOR YOUR ATTENTION