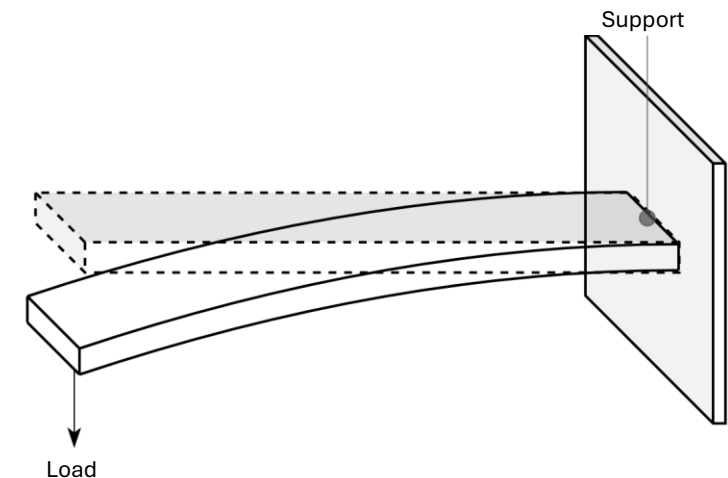


NUMERICAL MODELLING AND SIMULATION – PART A

REPORT 1 CANTILEVER BEAM MODEL



TARGETS



Validate the **convergence** of **numerical results** against the **analytical** solutions derived from De Saint-Venant's theory.



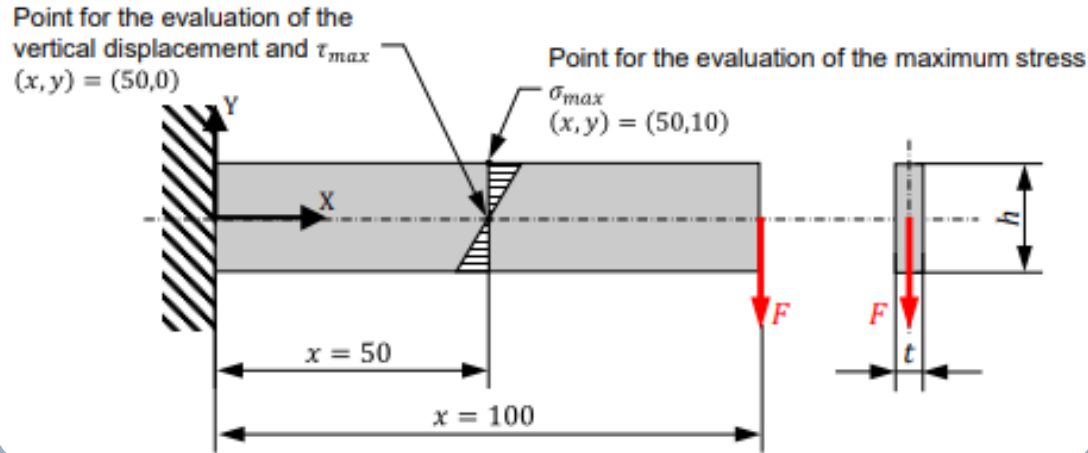
Compare results obtained using **three-node** and **four-node** shell elements.



Assess the **impact** of various **boundary conditions** on the representation of **stress** and **strain** fields.

PROBLEM ILLUSTRATION

CANTILEVER BEAM



Point of evaluation



beam midpoint
($x = 50 \text{ mm}$ and $y = 0 \text{ mm}$)

Why? → edges results could show a bigger error due to FEM nature

$$J = \frac{t h^3}{12} = 3333,33 \text{ mm}^4$$

CANTILEVER BEAM

MATERIAL

- Category: Steel
- $E = 206000 \text{ MPa}$
- $\nu = 0,3$
- $\rho = 7,8 \cdot 10^{-9} \text{ Gkg/l}$

DIMENSION

- $t = 5 \text{ mm}$
- $l = 100 \text{ mm}$
- $h = 20 \text{ mm}$

$$\sigma_{max} = \frac{6 F x_p}{t h^2} = 150 \text{ MPa}$$

$$\tau_{max} = \frac{3 F}{2 t h} = 15 \text{ MPa}$$

$$f_p = \frac{F x_p^2}{6 E J} (3 l - x_p) = -0,1517 \text{ mm}$$

METHODS - DISCRETIZATION

SOFTWARES

- **FEA:** Hypermesh 2024
- **Solver:** Optistruct
- **Visualizer:** Hyperview 2024

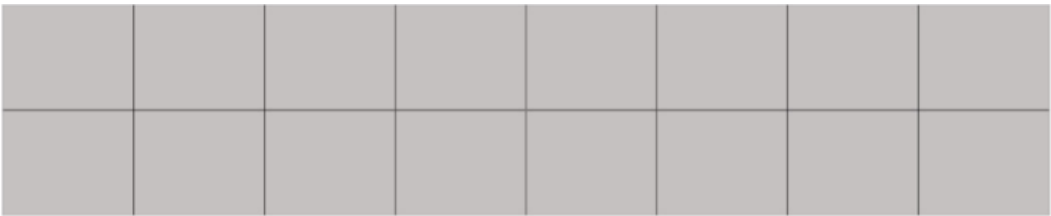
Number of elements

height	Width	Total nodes
2	10	33
4	20	105
10	50	561
12	60	793
14	70	1065
16	80	1377
20	100	2121
24	120	3025

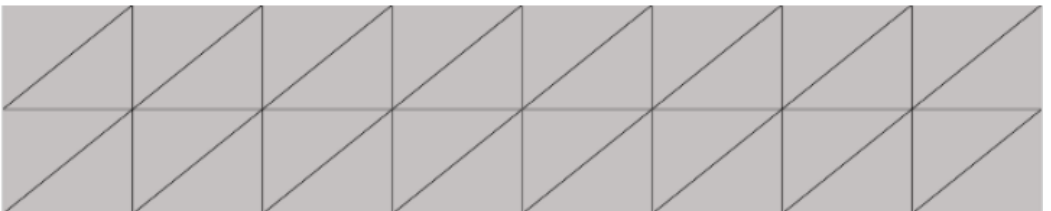
Discretization → 2 D model (Shell elements) **Why?** → $t \ll l$ and h

TYPE OF DISCRETIZATION

4
NODES
SQUARE



3
NODES
R-TRIAS



Density of elements

Density on width = density on length
To take into account equal shape of the elements

PROPERTY

PSHELL



METHODS - BOUNDARY CONDITIONS

LOADS

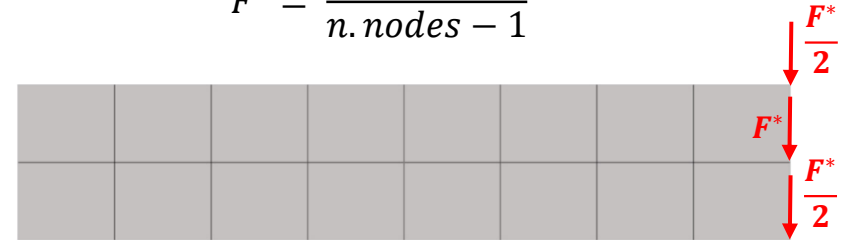
- a) **Single load** applied on **midpoint** of right side

$$F = 1000 \text{ N}$$



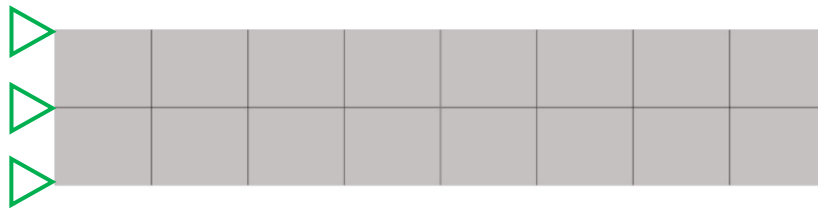
- b) **Distributed load** applied on **each nodes** of right side

$$F^* = \frac{F}{n.\text{nodes} - 1}$$

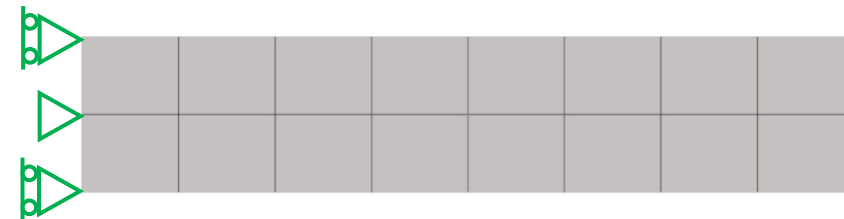


CONSTRAINTS

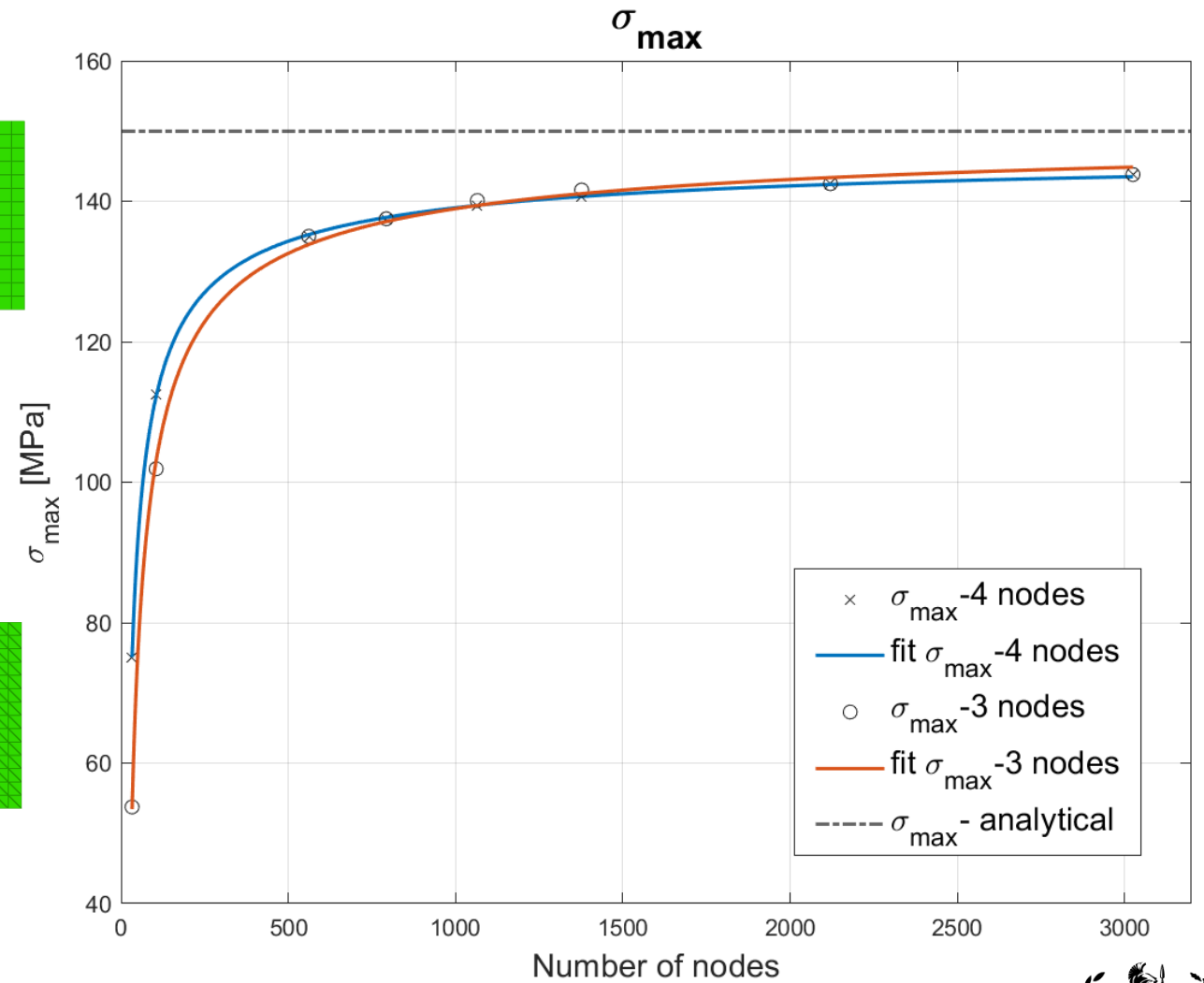
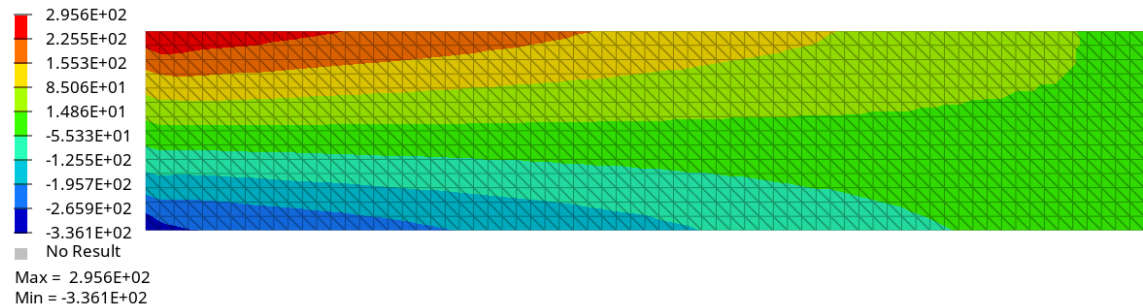
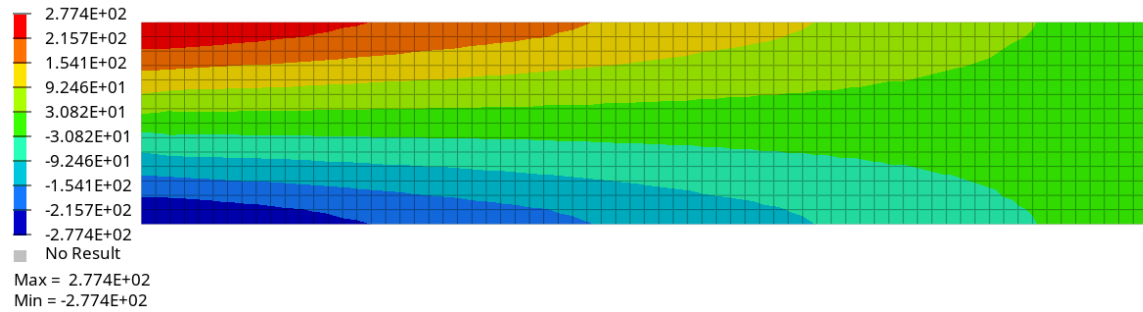
- a) Constraining to 0 the displacements **along x and y** of **all nodes**



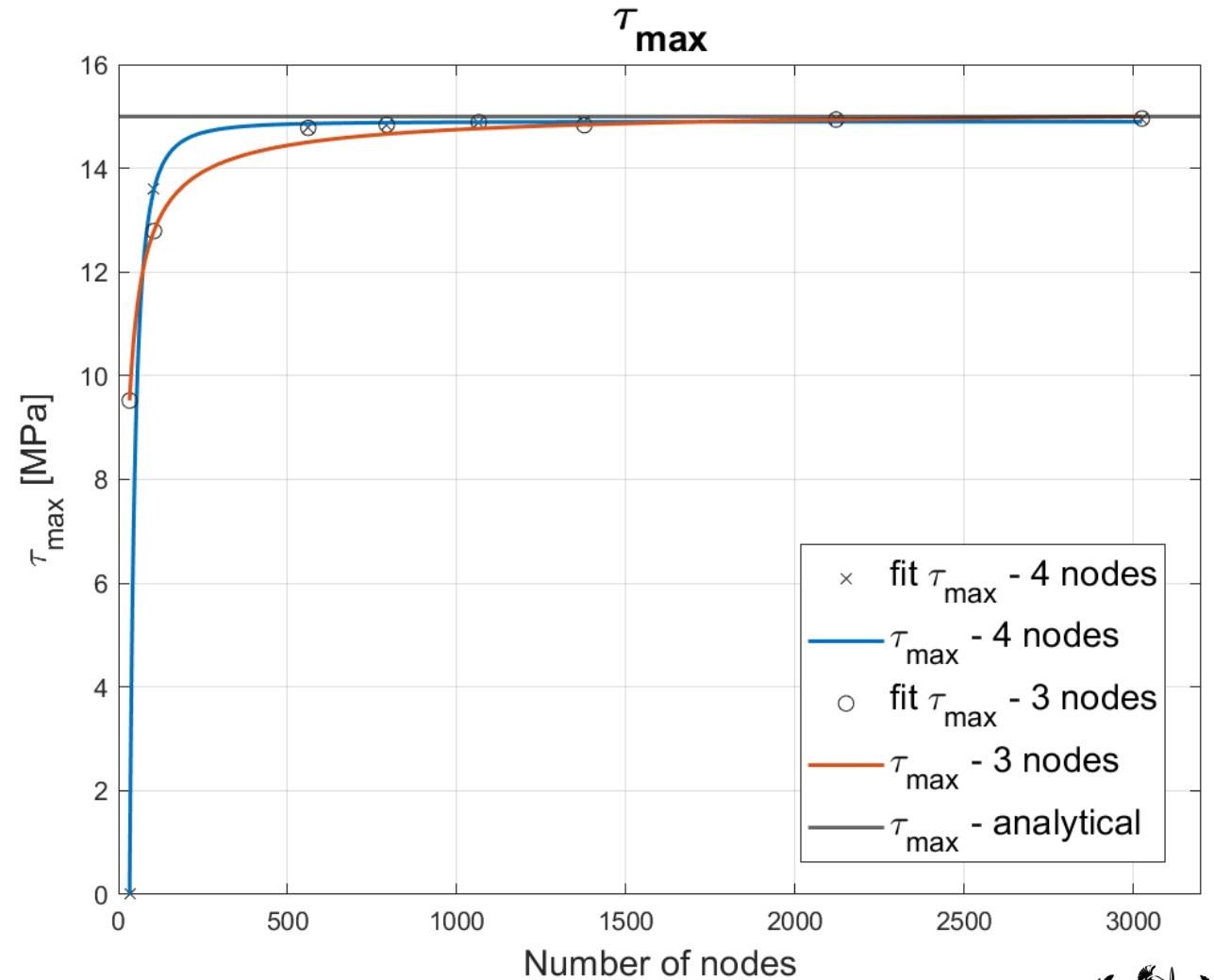
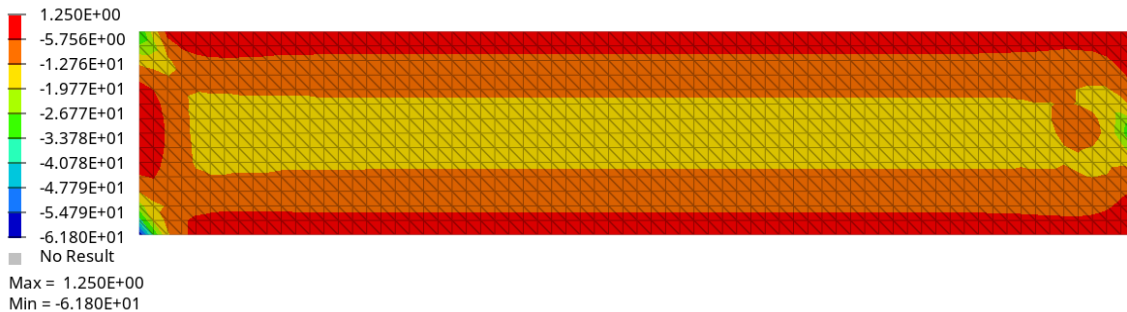
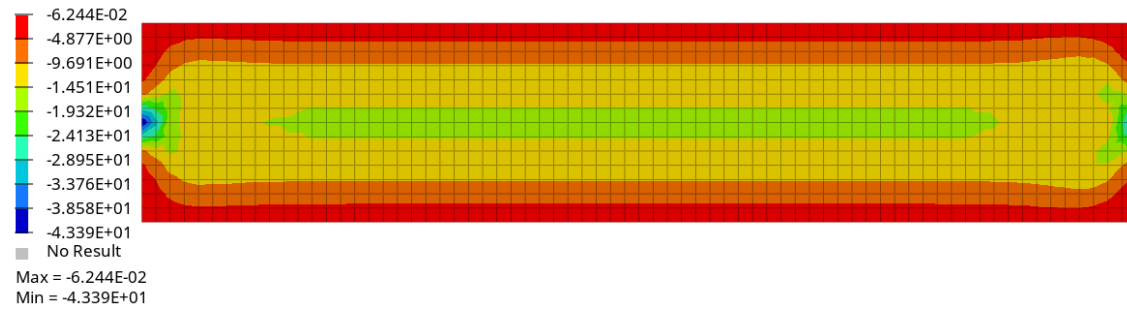
- b) Constraining to 0 the displacements **along x** of **all node** and **along y** of **middle node** – to simulate deformation in y axis



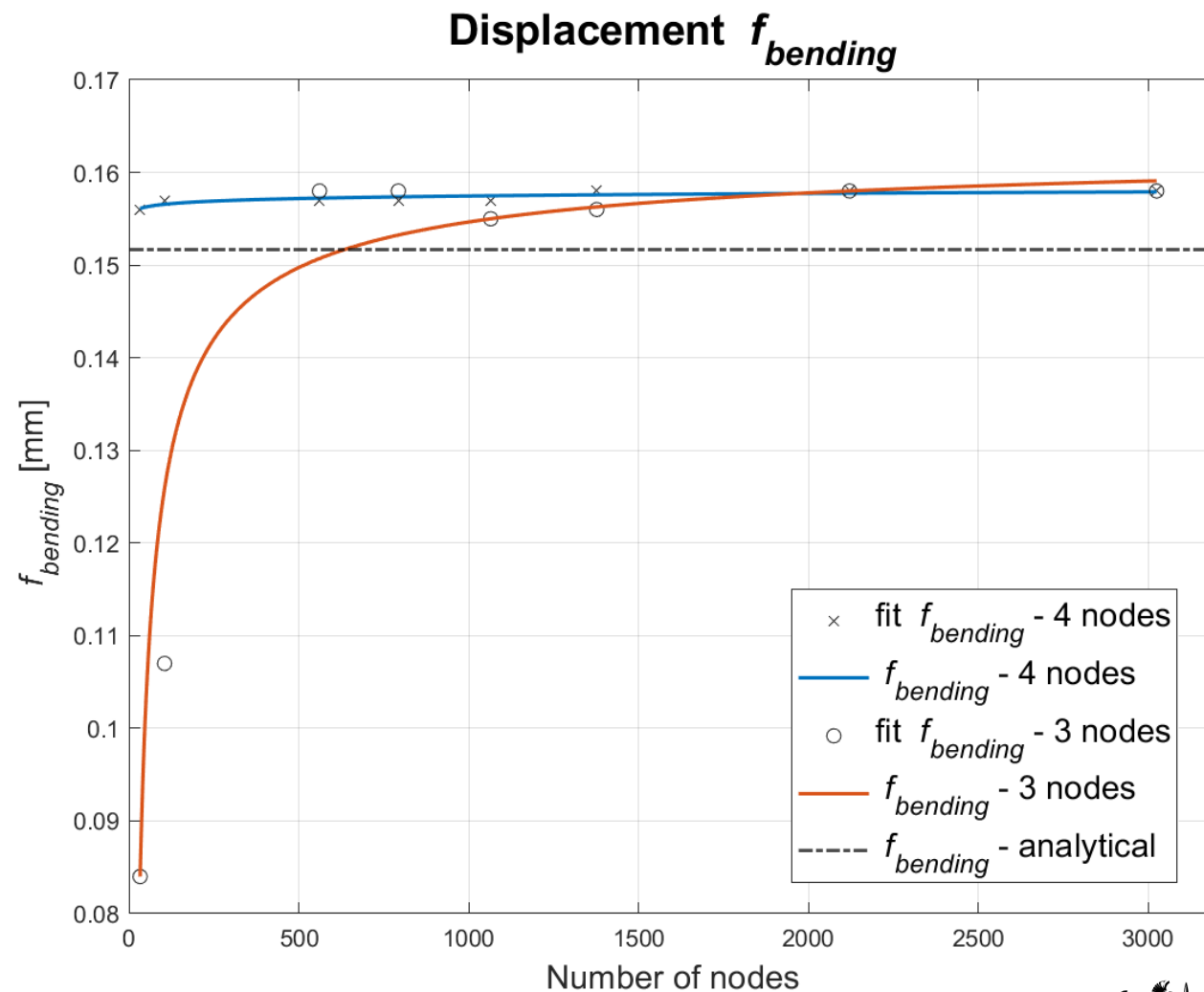
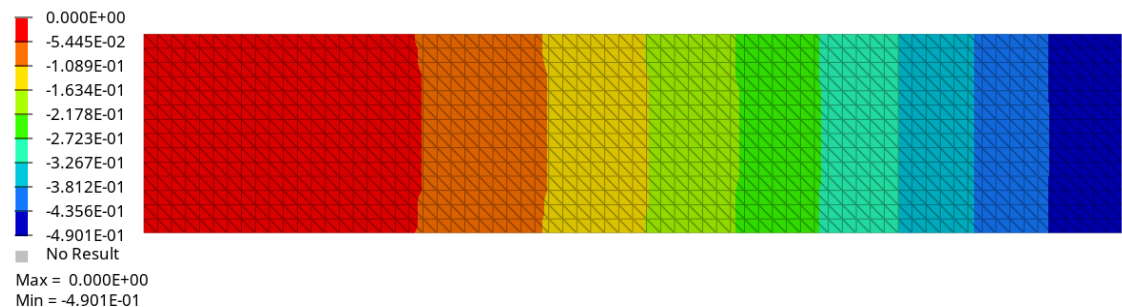
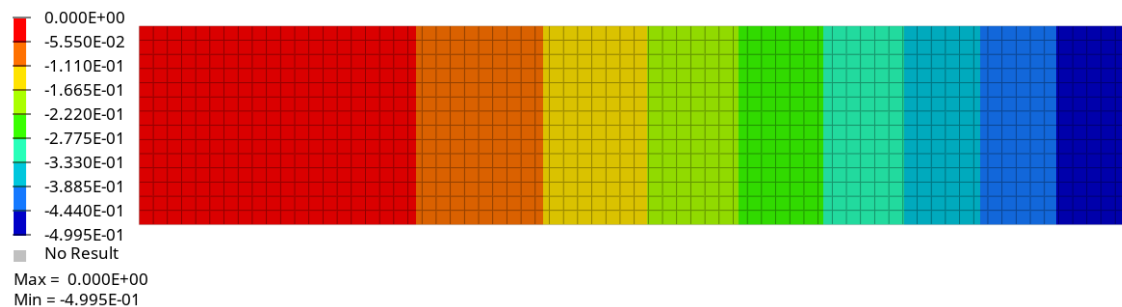
RESULTS – σ_{MAX} GRAPH



RESULTS – τ_{\max} GRAPH



RESULTS – DISPLACEMENT GRAPH



RESULTS – RIEPILOGATION

Elements type	Elements number	σ_{MAX} [Mpa]	Err% σ_{MAX}	τ_{MAX} [Mpa]	Err% τ_{MAX}	$f_{bending}$ [mm]	Err% $f_{bending}$
4 nodes	2 x 10	75	50%	0	100%	0,156	2,84%
	4 x 20	112,5	25%	13,6	9,33%	0,157	3,49%
	10 x 50	135	10%	14,78	1,47%	0,157	3,49%
	12 x 60	137,5	8,33%	14,84	1,07%	0,157	3,49%
	14 x 70	139,3	7,13%	14,89	0,73%	0,157	3,49%
	16 x 80	140,6	6,27%	14,91	0,6%	0,158	4,15%
	20 x 100	142,5	5%	14,94	0,4%	0,158	4,15%
	24 x 120	143,8	4,13%	14,96	0,27%	0,158	4,15%
3 nodes	2 x 10	53,76	64,16%	9,52	36,53%	0,084	44,63%
	4 x 20	101,9	32,07%	12,79	14,73%	0,107	29,47%
	10 x 50	135	10,00%	14,78	1,47%	0,158	4,15%
	12 x 60	137,5	8,33%	14,84	1,07%	0,158	4,15%
	14 x 70	140,1	6,60%	14,78	1,47%	0,155	2,18%
	16 x 80	141,6	5,60%	14,83	1,13%	0,156	2,84%
	20 x 100	142,5	5,00%	14,94	0,40%	0,158	4,15%
	24 x 120	14,8	4,13%	14,96	0,27%	0,158	4,15%

Reference values		
σ_{MAX} [Mpa]	τ_{MAX} [Mpa]	$f_{bending}$ [mm]
150	15	0,152

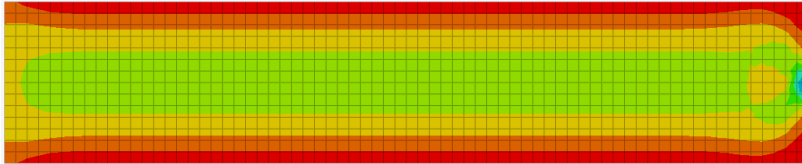
The better overall approximation is achieved with the four-node element with a 14x70 discretization, as shown in the table.

RESULTS - BOUNDARY CONDITIONS

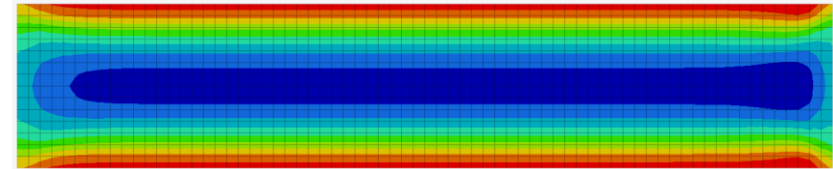
LOADS

In **case b)** result shows a **better distribution** of τ_{xy} on the **right side** where the load is applied.

a) Single force



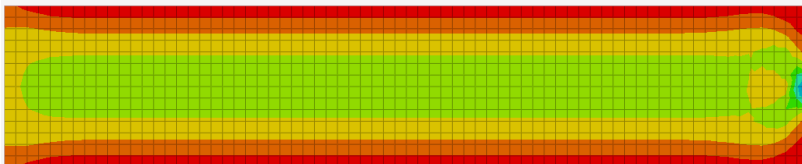
b) Distributed forces



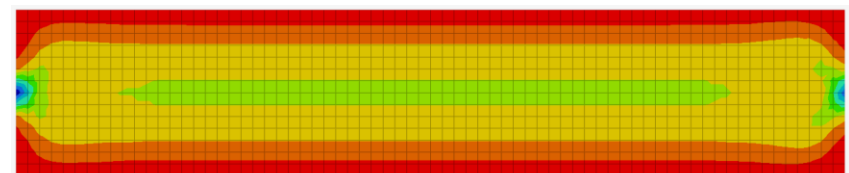
CONSTRAINTS

In **case a)** result shows a **better distribution** of τ_{xy} on the **left side** where the constraints are applied.

a) All nodes fully constrained



b) Only middle node fully constrained



CONCLUSIONS



Stresses: 4-node elements yield more accurate results than three-node elements with a lower discretization, while with a higher one both approaches converge to the analytical values with no significant differences.



Displacement: 4 nodes elements give good approximation already with minimum elements (2x10) whereas 3 nodes need higher discretization.



Boundary conditions: Overall, a distributed load and constraints tend to produce a better approximation of τ_{xy} near the application point.

NUMERICAL MODELLING AND SIMULATION – PART A

REPORT 2 ROD ENGINE



TARGETS



To perform a **linear static analysis** of the most **critical load conditions** for the rod (slightly before and after Top Dead Center).

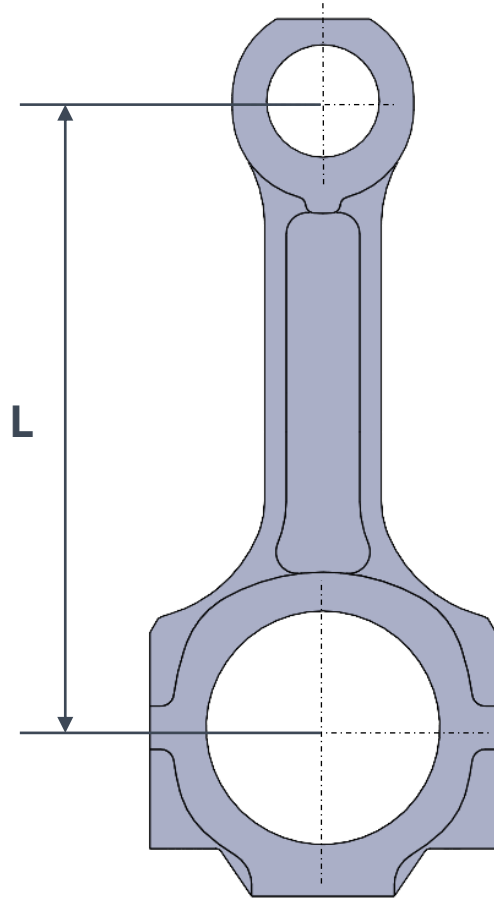


To go into **detail** of the **most stressed areas**.



To evaluate the **influence** of **different meshes** and how **boundary conditions** are applied.

PROBLEM ILLUSTRATION



PROPRIETIES

MATERIAL

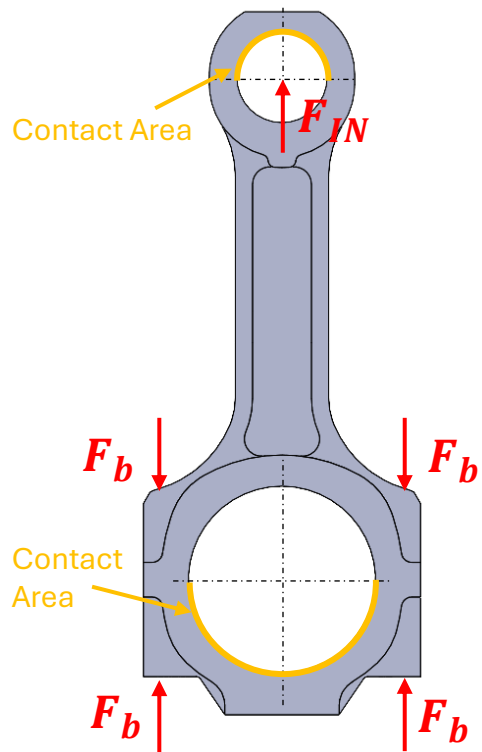
- **Category:** cast iron
- $E = 165000 \text{ MPa}$
- $\nu = 0,28$
- $\rho = 7,2 \cdot 10^{-9} \text{ Gkg/l}$

DIMENSIONS

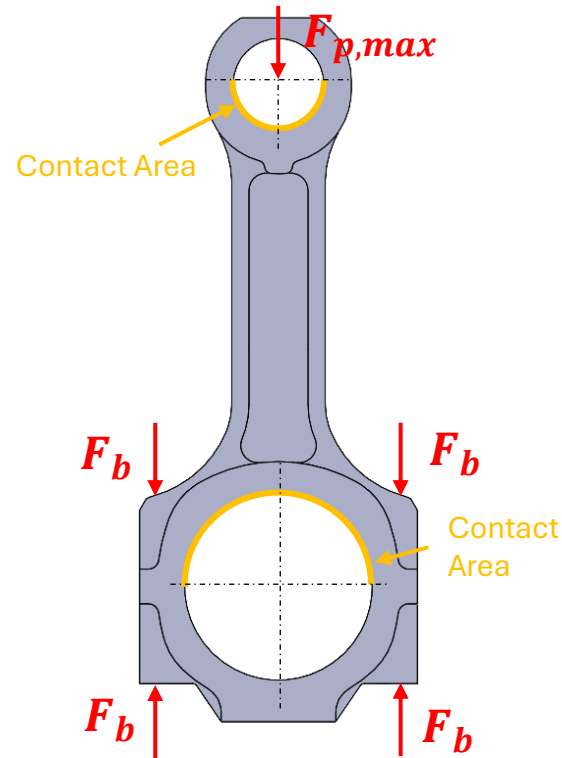
- $Stroke = 90,4 \text{ mm}$
- $bore = 80 \text{ mm}$
- $L = 145 \text{ mm}$
- $bolts = M8 \times 1,25$
- $Mass = 0,81 \text{ kg}$

METHODS - FORCES

BEFORE TDC



AFTER TDC



$$F_{p,max} = \frac{\pi D^2}{4} * p_{max} = 73,93 * 10^6 N$$

$$J = \frac{t h^3}{12} = 3333,33 mm^4$$

$$F_b = 32000 N$$

$$R = \frac{Stroke}{2} = 45,2 mm$$

$$n_{max} = 5000 rpm$$

$$\lambda = \frac{R}{L} = 0,3117$$

$$a_{TDC} = (1 + \lambda) R \omega^2 = 16255 m/s^2$$

$$F_{IN} = m_{ROD} * a_{TDC} = 13166,28 N$$

METHODS - DISCRETIZATION

SOFTWARES

- **FEA:** Hypermesh 2024
- **Solver:** Optistruct
- **Visualizer:** Hyperview 2024

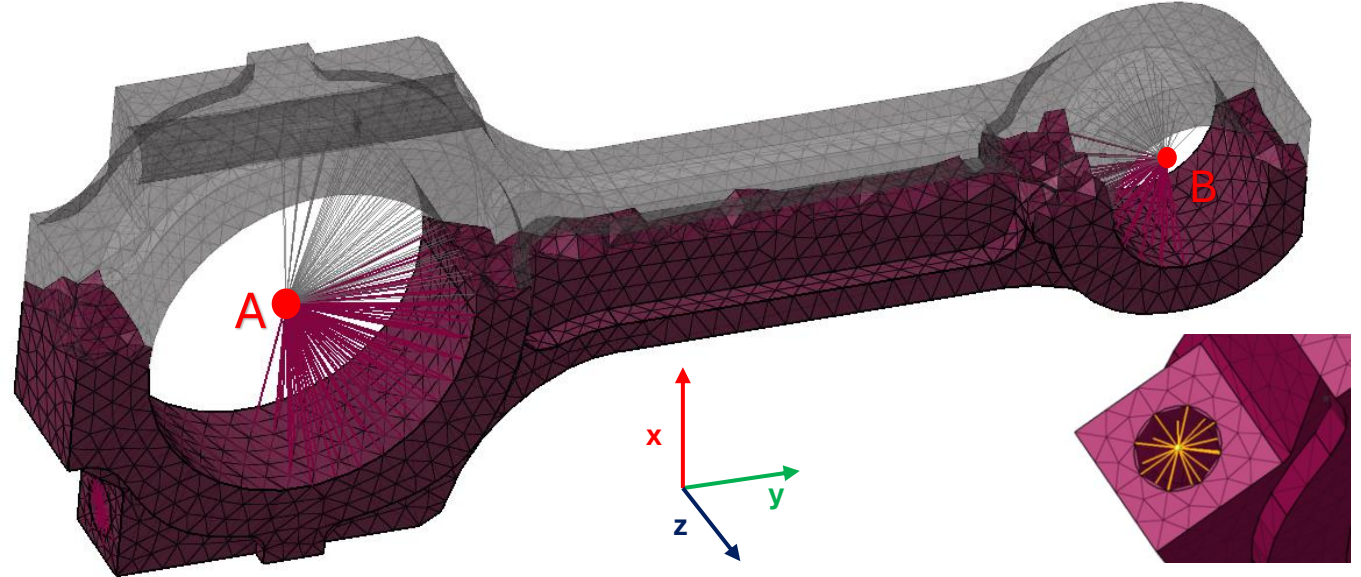
MESH

- **Type:** 3D - Tetrahedral elements
- **Size:** 4 mm

CONNECTORS: RBE2

Loads and constraints are applied to the model through rigid connectors (big and small eyes and bolts)

TYPE OF DISCRETIZATION



TYPE OF CONSTRAINTS

Big eye (A): All constrained

Small eye (B): x and z displacement constrained

PROPERTY

PSOLID

RESULTS – BEFORE TDC (INERTIA)

Big end is the most stressed part due to bolts tightening



Compressive load in the big end

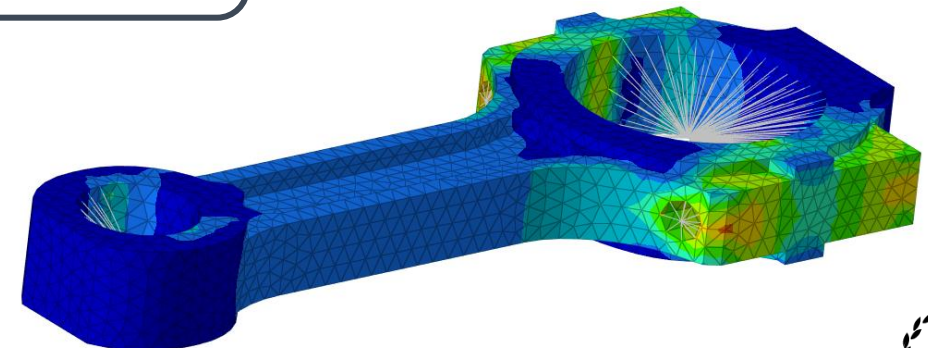
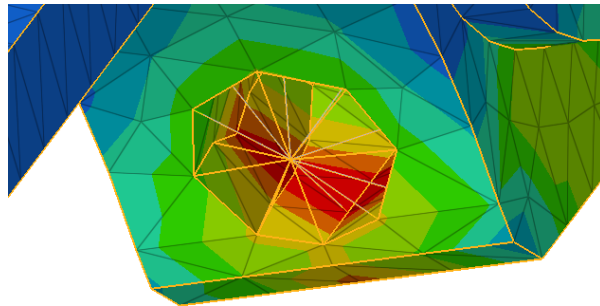


Maximum stress in the fillets where bolts are screwed → 248 MPa

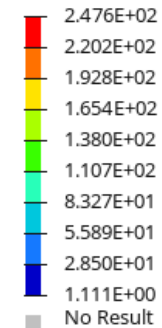


Stress concentration in notch

Connecting rod loaded with tensile stress



Contour Plot
Element Stresses (2D & 3D)(vonMises)
Global System
Advanced Average



Max = 2.476E+02
Grids 999
Min = 1.111E+00
Grids 2559

RESULTS – AFTER TDC (GAS PRESSURE)

Connecting rod is the most stressed part due to gas pressure load



Compressive load along connecting rod

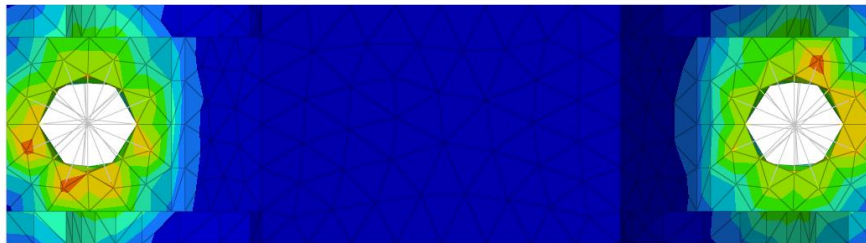


Maximum stress in the external fillet between connecting rod and big end
→ (302MPa)



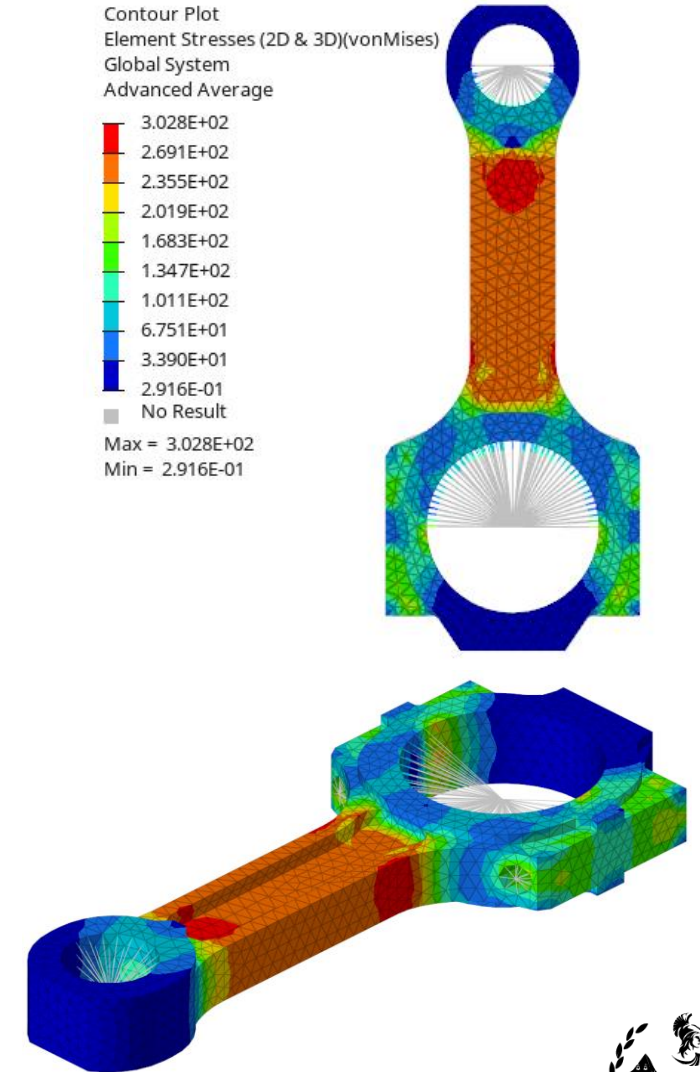
Stress concentration below small end due to material deformation → 270 MPa

Bolt holes are the second most stressed zone → 250MPa



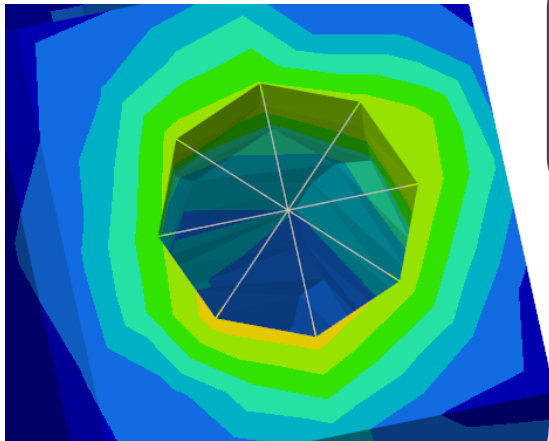
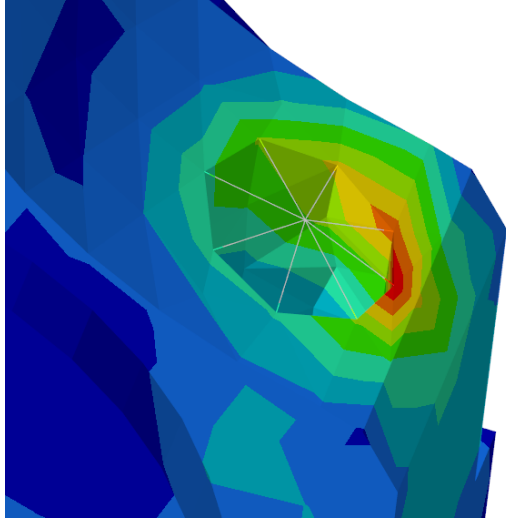
Contour Plot
Element Stresses (2D & 3D)(vonMises)
Global System
Advanced Average

3.028E+02
2.691E+02
2.355E+02
2.019E+02
1.683E+02
1.347E+02
1.011E+02
6.751E+01
3.390E+01
2.916E-01
No Result
Max = 3.028E+02
Min = 2.916E-01



IMPROVEMENT - BOUNDARY CONDITIONS

CASE A

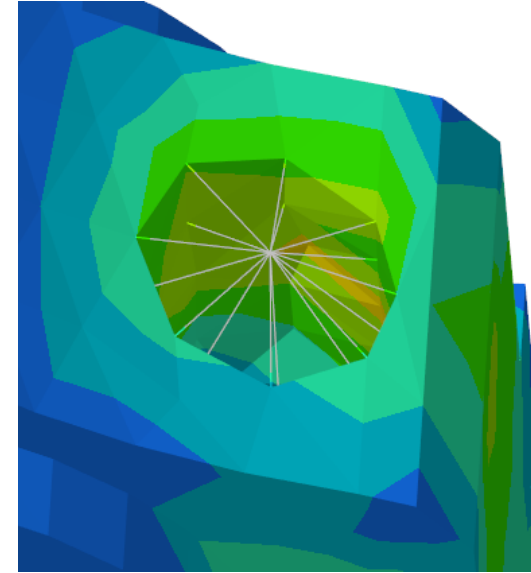


Case A: stress distributed along the edge

Case B: stress applied in a surface

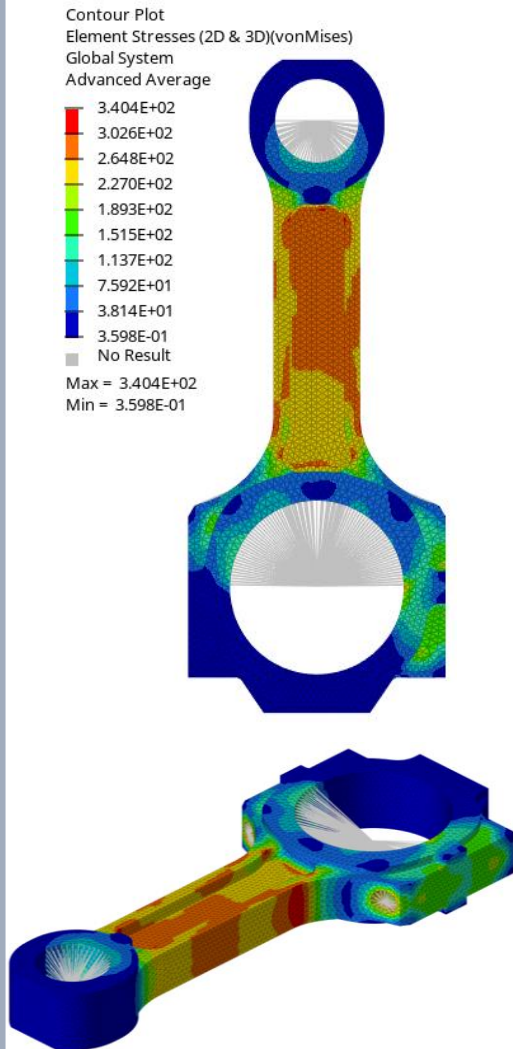
Resulting effect: Case B better distribution of the load, near to reality

CASE B

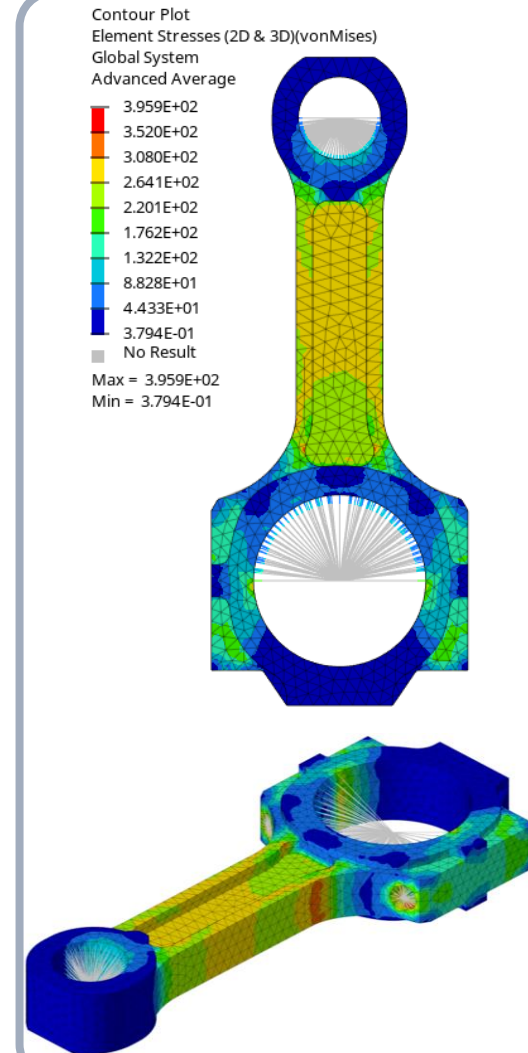
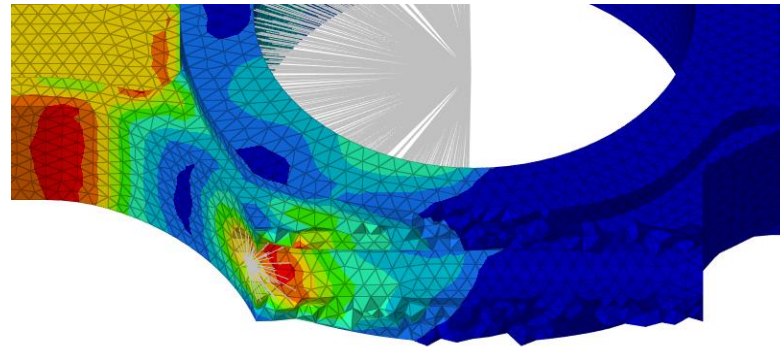


IMPROVEMENT - MESH

MESH 2mm



Overall, **improving the mesh quality** results in an **increase in stress** at the same critical points, especially at the **fillet between the big end and the rod**, as well as the **bolt connection**.



MESH 4mm – SECOND ORDER

CONCLUSION



Stresses: the **rod after TDC** (gas pressure case) appears to be **more stressed**, particularly the connecting rod and the screw holes have high compression load.



Boundary conditions: application of **small eye constraint** and **redistribution of rigid connectors** for the bolts have **improved the reliability** of the simulation.

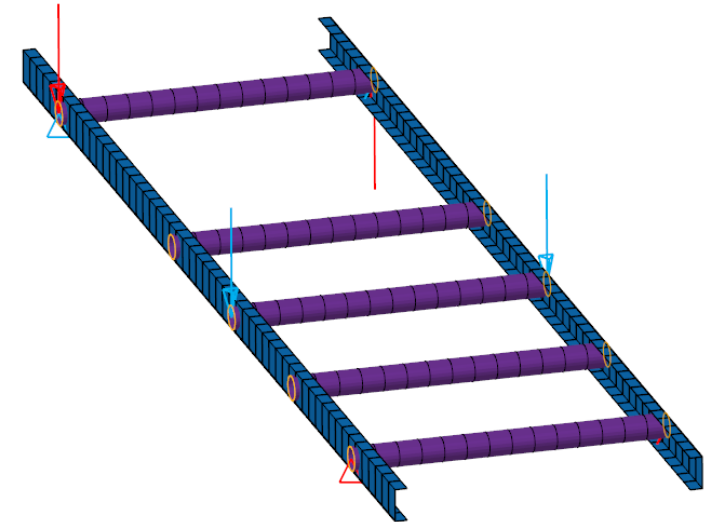


Mesh: **Improving the mesh quality** generally leads to an **increase in stress** throughout the rod, with a more pronounced effect at critical points.

NUMERICAL MODELLING AND SIMULATION – PART A

REPORT 3

STUDY OF A COMMERCIAL VEHICLE CHASSIS



TARGETS



To understand the **1D behavior**.

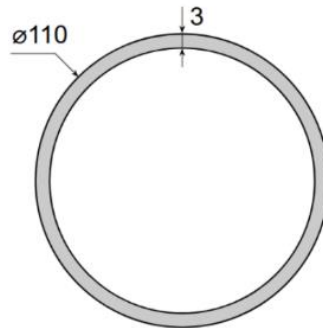
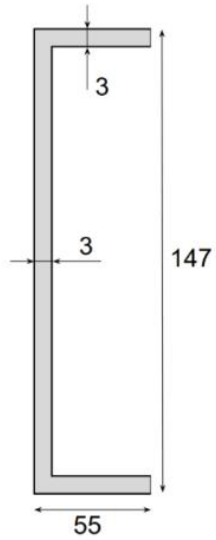
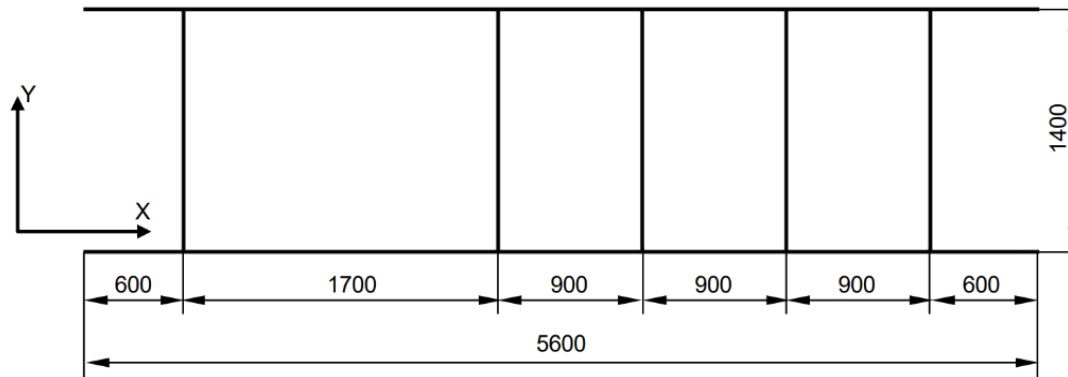


To evaluate **bending and torsional stiffness** of the frame.



To evaluate of **influence of different parameters** (material, geometry, dimensions, orientation).

PROBLEM ILLUSTRATION



MATERIAL

- **Category: Steel**
- $E = 206000 \text{ MPa}$
- $\nu = 0,3$
- $\rho = 7,85 \cdot 10^{-9} \text{ Gkg/m}^3$

GEOMETRY

- **Longitudinal beams:** C section, open sides oriented towards center of structure.
- **Cross beams:** hollow circular cross section.

METHODS - DISCRETIZATION

SOFTWARES

- **FEA:** Hypermesh 2024
- **Solver:** Optistruct
- **Visualizer:** Hyperview 2024

MESH

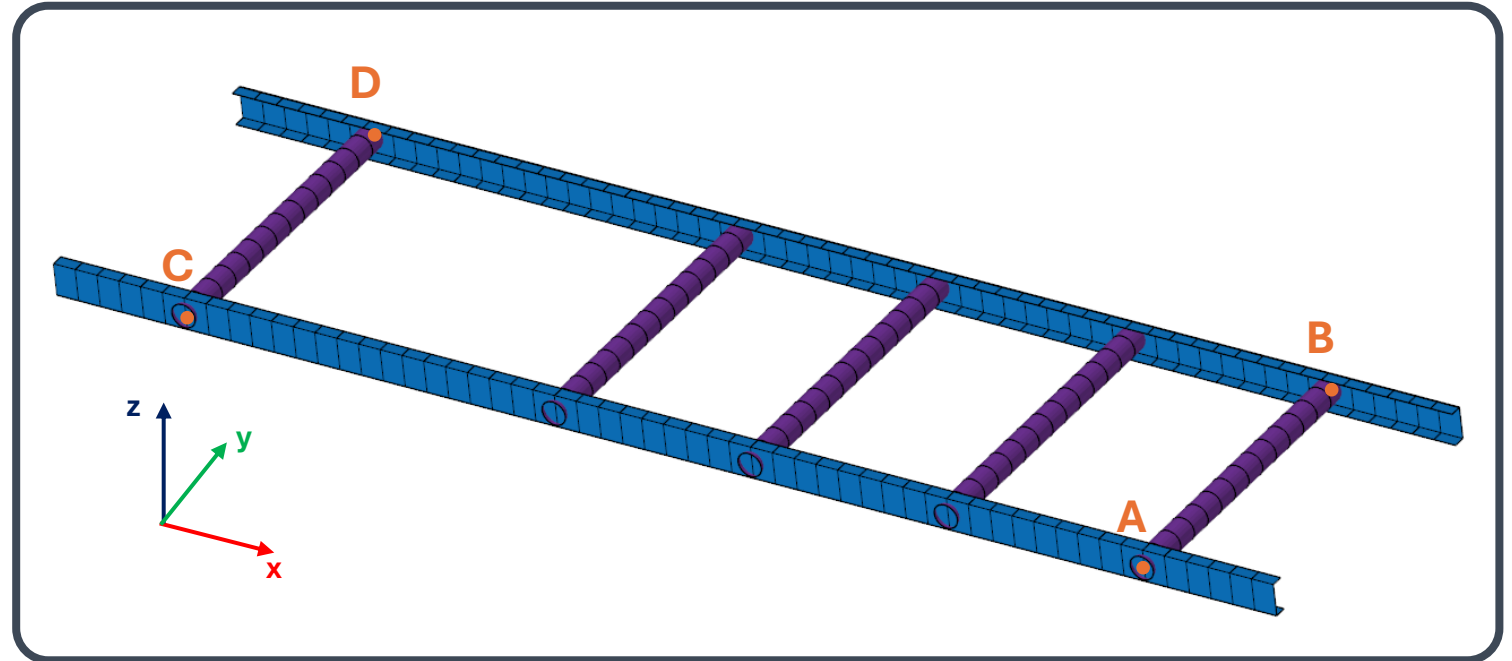
- **Type:** 1D
- **Size:** 100 mm

TYPE OF CONSTRAINTS

displacement along **x,y,z** and
rotation around **z** and **x**
constrained:

Torsion case: nodes A and B

Bending case: nodes A,B,C,D



SECTION TYPE

Longitudinal beam: CHAN

Cross beam: TUBE

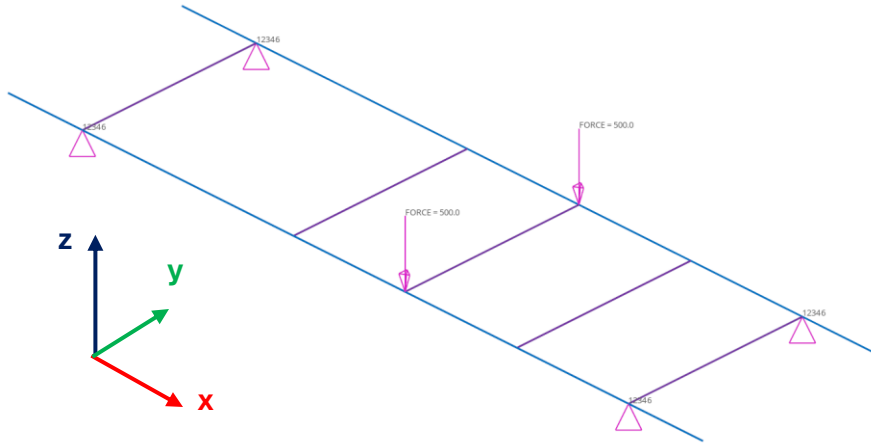
PROPERTY

PBEAML

LOAD CASES AND FORMULAS

Linear field: to calculate the stiffnesses a generic test force of 500 N is used

BENDING



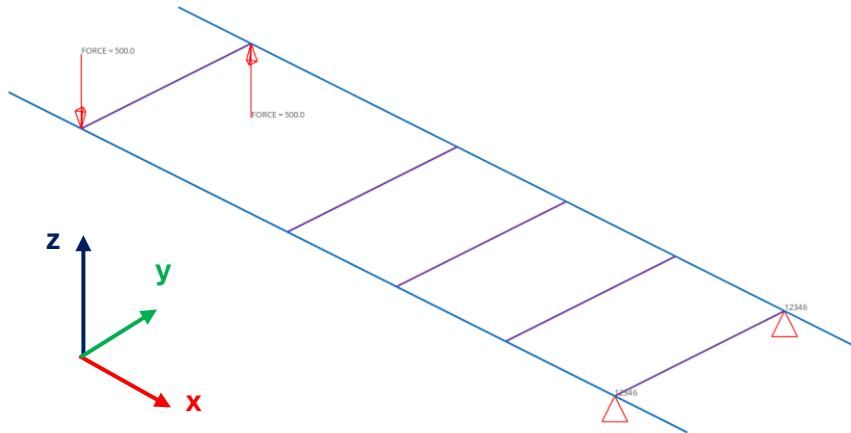
Forces applied in the **middle of longitudinal** beam to generate a **bending moment**

Δz = maximum z displacement in the node where the structure is loaded

$$F = 500 \text{ N}$$

$$K_b = \frac{F}{\Delta z}$$

TORSION



Couple of forces applied in junction point between **suspension** of an axle and chassis to generate a **torsion moment**

$$F = 500 \text{ N}$$

Hp: small angles $\rightarrow \tan(\vartheta) \cong \vartheta \rightarrow \vartheta = \frac{\Delta z}{b}$

$$b = 1400 \text{ mm}$$

$$M_t = F \cdot b = 700000 \text{ Nmm}$$

$$K_t = \frac{M_t}{\vartheta}$$

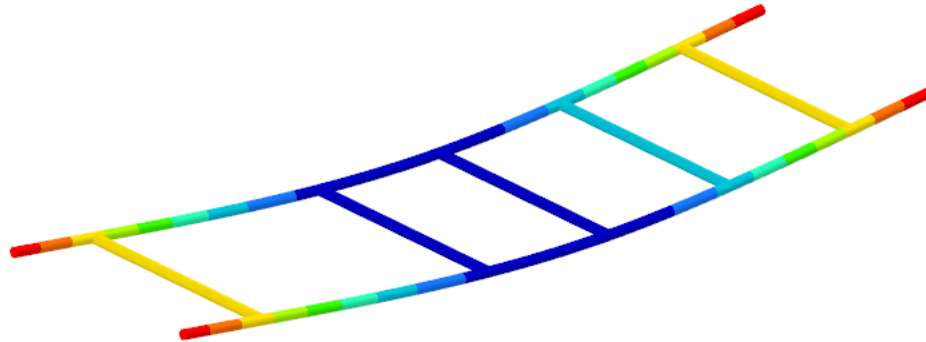
RESULTS

BENDING

Contour Plot
Displacement(Z)
Analysis system

7.493E-01
4.753E-01
2.012E-01
-7.282E-02
-3.469E-01
-6.209E-01
-8.950E-01
-1.169E+00
-1.443E+00
-1.717E+00
No Result

Max = 7.493E-01
Min = -1.717E+00



$$\Delta z = 1,717 \text{ mm}$$

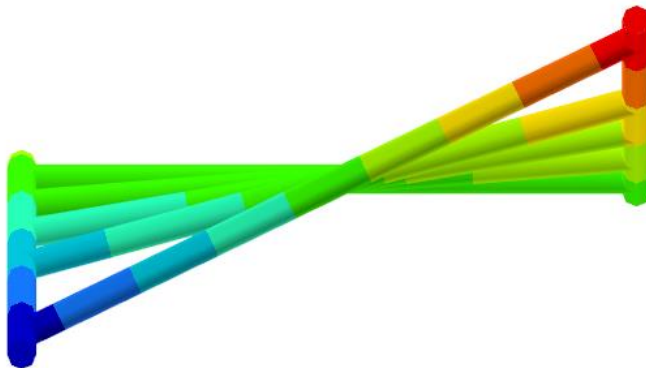
$$K_b = \frac{F}{\Delta z} = 291,21 \text{ N/mm}$$

TORSION

Contour Plot
Displacement(Z)
Analysis system

7.249E+00
5.555E+00
3.860E+00
2.166E+00
4.717E-01
-1.223E+00
-2.917E+00
-4.611E+00
-6.305E+00
-8.000E+00
No Result

Max = 7.249E+00
Min = -8.000E+00



$$\Delta z_{left} = 8 \text{ mm}$$

$$\Delta z_{right} = 7,249 \text{ mm}$$

$$\Delta z_{average} = 7,623 \text{ mm} \quad \vartheta = \frac{\Delta z_{average}}{b} = 1,09 \cdot 10^{-2} \text{ rad}$$

$$K_t = \frac{M_t}{\vartheta} = 6,43 \cdot 10^7 \text{ Nmm}$$

PARAMETERS INFLUENCE

Case	Description
a.	Reference- Mesh 100 mm
b.	Mesh 25 mm
c.	Al 7075 T6
d.	Rotated long. beam
e.	Double thickness cross beam
f.	Double thickness long. beam
g.	Half diameter cross (same thickness)
h.	Double section dimensions long. beam (same thickness)
i.	box section (cross beam)

Case d.



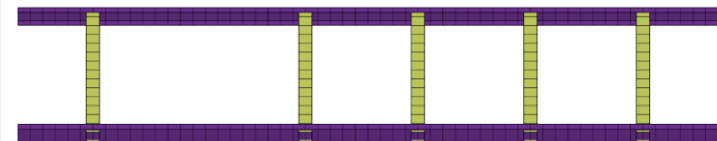
Case g.



Case h.



Case i.



PARAMETERS INFLUENCE

Case	Description	Torsion case					Bending case	
		Δz_{left}	Δz_{right}	$\Delta z_{average}$	ϑ	K_t	Δz	K_b
		[mm]	[mm]	[mm]	[rad]	[Nmm ²]	[mm]	[N/mm]
a.	Reference- Mesh 100 mm	8	7,249	6,7675	9,67E-03	7,24E+07	1,717	291,21
b.	Mesh 25 mm	7,909	6,883	6,7675	9,67E-03	7,24E+07	1,717	291,21
c.	Al 7075 T6	21,25	23,48	19,84	2,83E-02	2,47E+07	4,934	101,34
d.	Rotated long. beam	6,768	8,481	6,767	9,67E-03	7,24E+07	1,717	291,21
e.	Double thickness cross beam	3,917	5,018	4,006	5,72E-03	1,22E+08	1,717	291,21
f.	Double thickness long. beam	7,038	7,375	6,358	9,08E-03	7,71E+07	0,9175	544,96
g.	Half diameter cross (same thickness)	55,83	62,53	52,035	7,42E-02	9,43E+06	1,717	291,21
h.	Double section dimensions long. beam (same thickness)	5,957	7,793	6,0475	8,64E-03	8,10E+07	0,09061	5518,15
i.	box section (cross beam)	5,613	6,687	5,478	7,83E-03	8,95E+07	1,717	291,21

No significantly changes	Improvement	Worsening
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CONCLUSIONS



Torsional stiffness: primarily **influenced** by the **inertia moment of the crossbeam section**, with a higher value corresponding to a greater torsional stiffness.



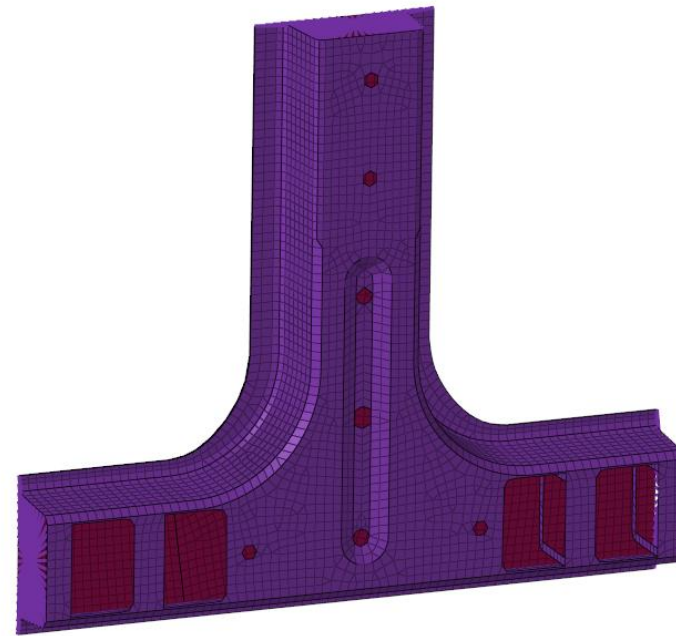
Bending stiffness: largely **depends** on the **inertia moment of the longitudinal beam section**; a greater moment of inertia results in increased bending stiffness.



Material: overall, both torsional and bending **stiffnesses** are directly **depended** on **mechanic characteristics** of the material.

NUMERICAL MODELLING AND SIMULATION – PART A

REPORT 4 Car body T-joint



PROBLEM ILLUSTRATION

HAT



PLATE

MATERIAL

- Category: Steel
- $E = 206000 \text{ MPa}$
- $\nu = 0,3$
- $\rho = 7,85 \cdot 10^{-9} \text{ Gkg/m}^3$

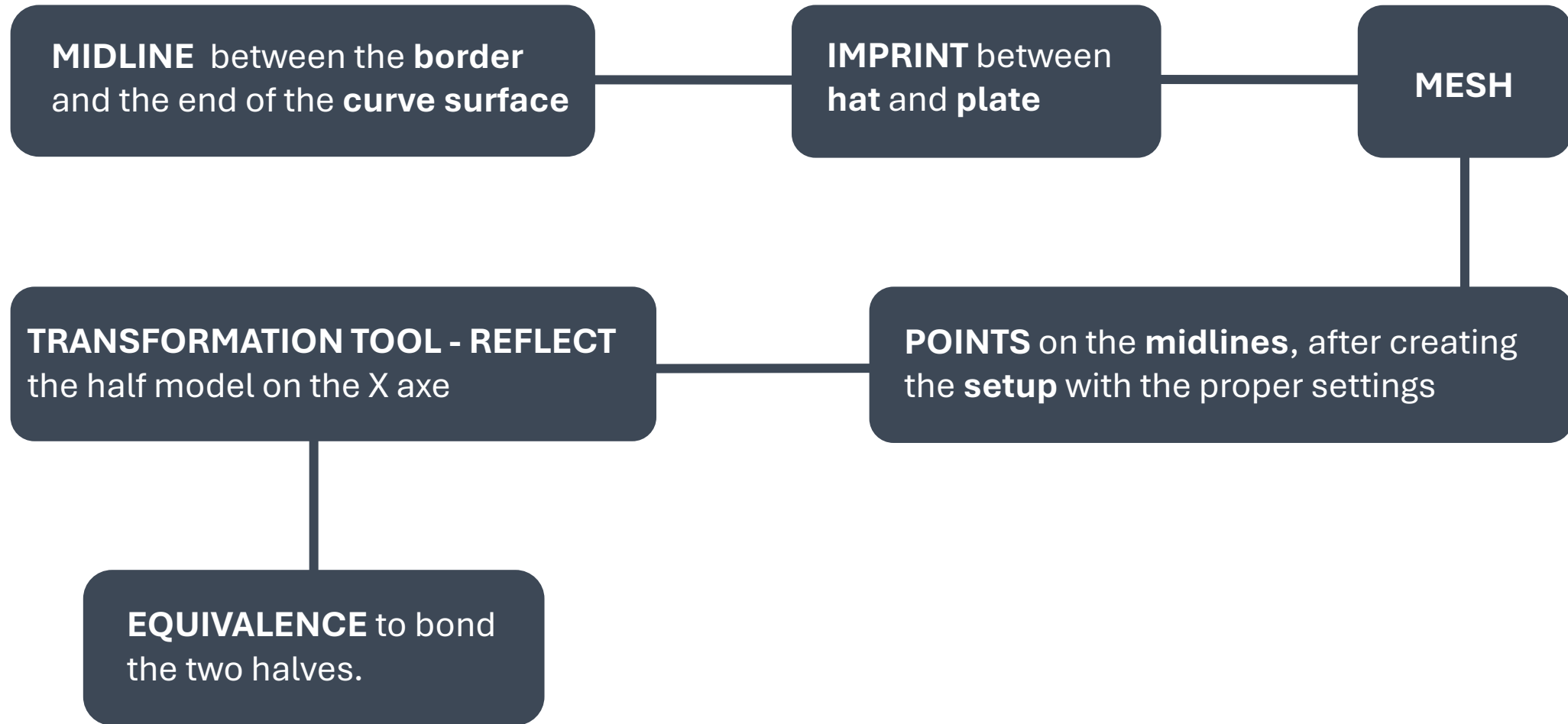
TARGETS

- To describe the main steps to create the welding.
- To calculate torsional stiffness (on Y axis) and bending stiffness (on X and Z axis).
- To evaluate the influence of number of spotwelds on the stiffnesses.

THICKNESS

$t = 1 \text{ mm}$

METHODS – STEPS OF CREATION



METHODS - DISCRETIZATION

SOFTWARES

- **FEA:** Hypermesh 2024
- **Solver:** Optistruct
- **Visualizer:** Hyperview 2024

MESH

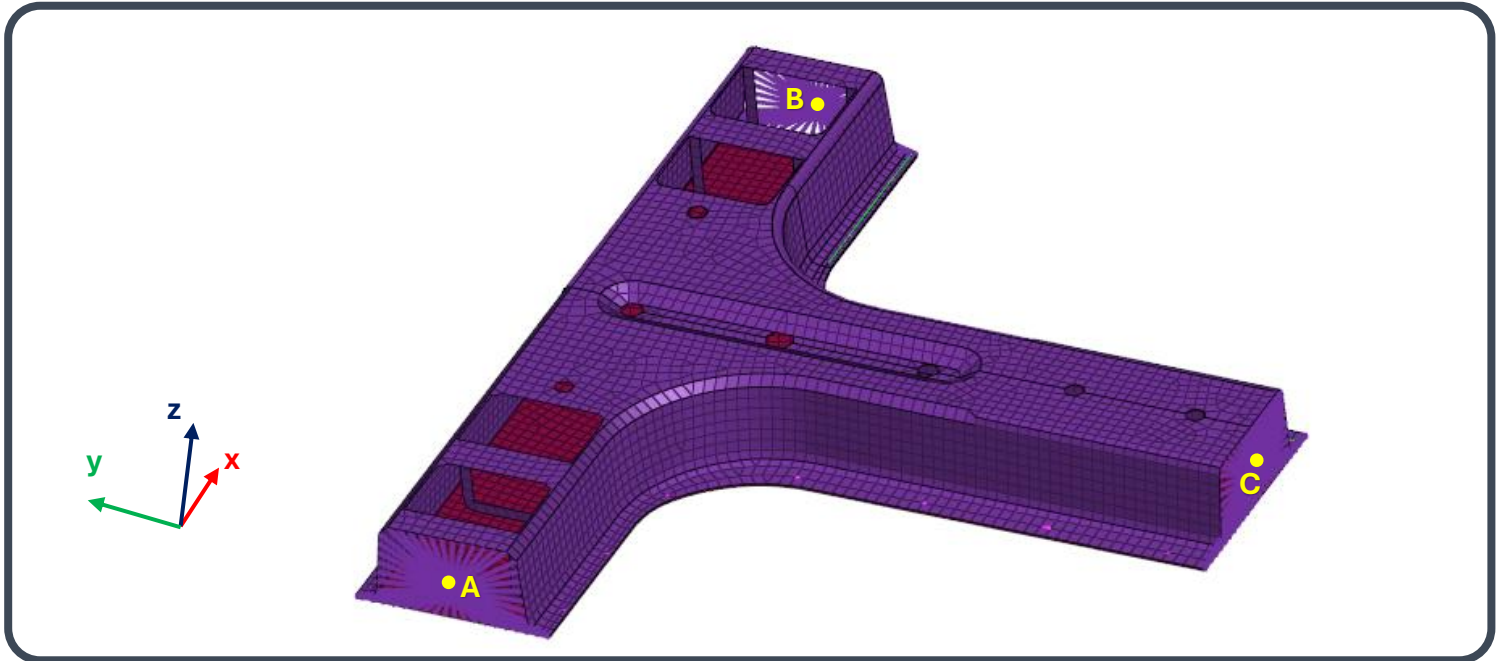
- **Type:** 2D – 4 nodes
- **Size:** 5 mm

PROPERTY

PSHELL

CONNECTORS

RBE2



TYPE OF CONSTRAINTS – APPLIED ON MASTER NODES

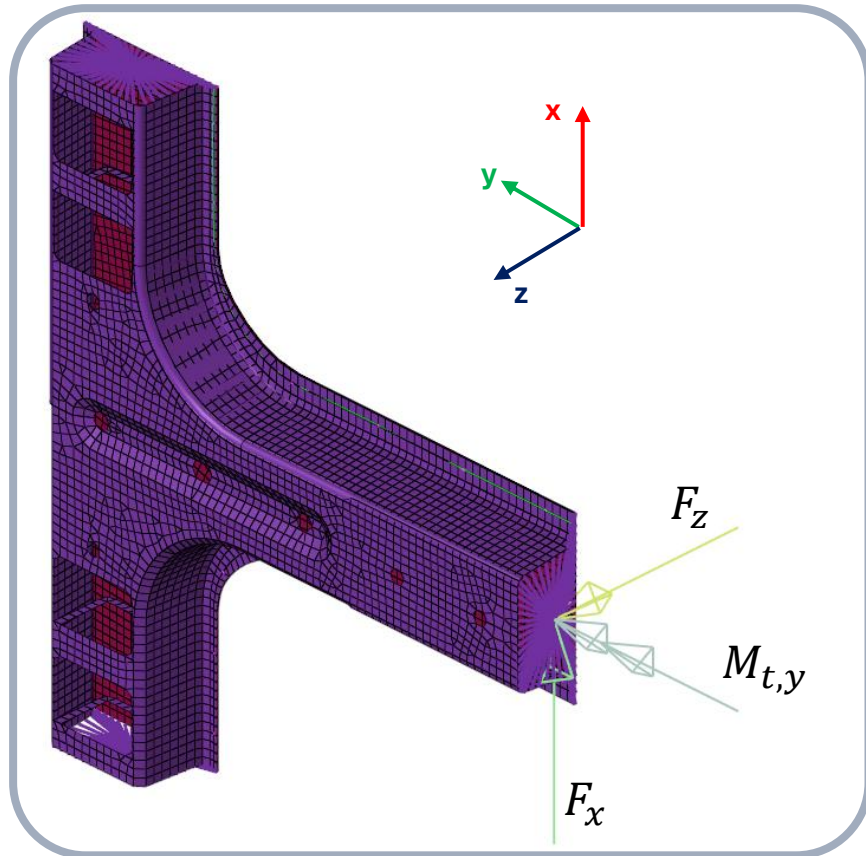
Torsion case: A,B fully constrained, C only rotation around Y unconstrained.

Bending case: A,B fully constrained.

LOAD CASES AND FORMULAS

Linear field: test loads are applied on the **master node** of the rigid element of the **upper arm section**.

Δz = max displacement in the node where the structure is loaded.



BENDING

Shear forces applied to obtain **bending stress**

$$F_x = 1000 \text{ N} \quad F_z = 100 \text{ N} \quad K_{b,i} = \frac{F_i}{\Delta z_i}$$

TORSION

Momentum applied to generate a **torsional moment**.

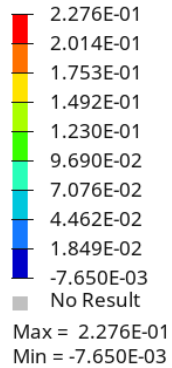
Hp: small angles $\rightarrow \tan(\vartheta) \cong \vartheta \rightarrow \vartheta = \frac{\Delta z}{b}$

$$M_{t,y} = 1000 \text{ Nm} \quad K_{t,y} = \frac{M_t}{\vartheta}$$

RESULTS

BENDING – $K_{b,x}$

Contour Plot
Displacement(X)
Analysis system

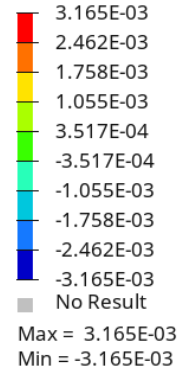


$$\Delta x = 0,2276 \text{ mm}$$

$$K_{b,x} = 439,37 \text{ N/mm}$$

TORSION – $K_{t,y}$

Contour Plot
Displacement(Z)
Analysis system



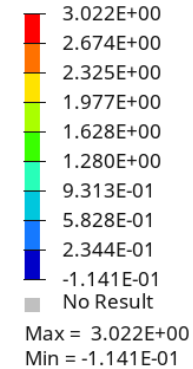
$$\Delta z_t = 3,17 \cdot 10^{-3} \text{ mm}$$

$$\vartheta_t = 4,82 \cdot 10^{-5} \text{ rad}$$

$$K_{t,y} = 1,71 \cdot 10^7 \text{ N} \cdot \text{mm} / \text{rad}$$

BENDING – $K_{b,z}$

Contour Plot
Displacement(Z)
Analysis system



$$\Delta z = 3,022 \text{ mm}$$

$$K_{b,z} = 33,09 \text{ N/mm}$$

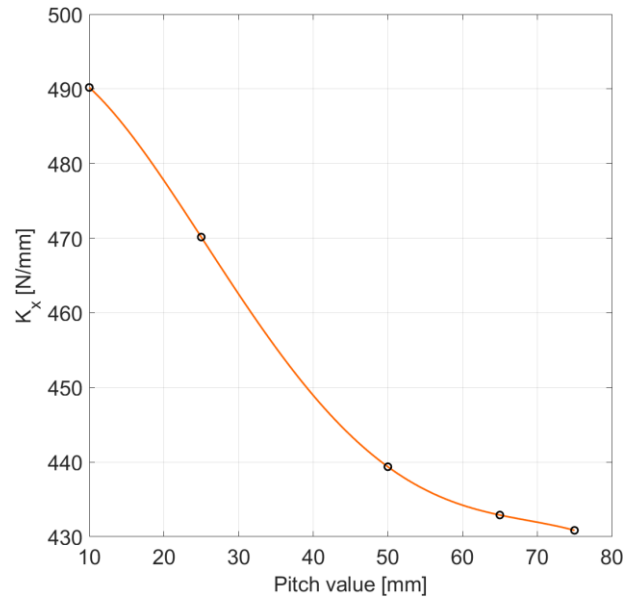
PARAMETERS INFLUENCE

Several analysis were performed **varying** the **pitch distance** to understand the **influence** of **spot welds** density on **stiffness**.

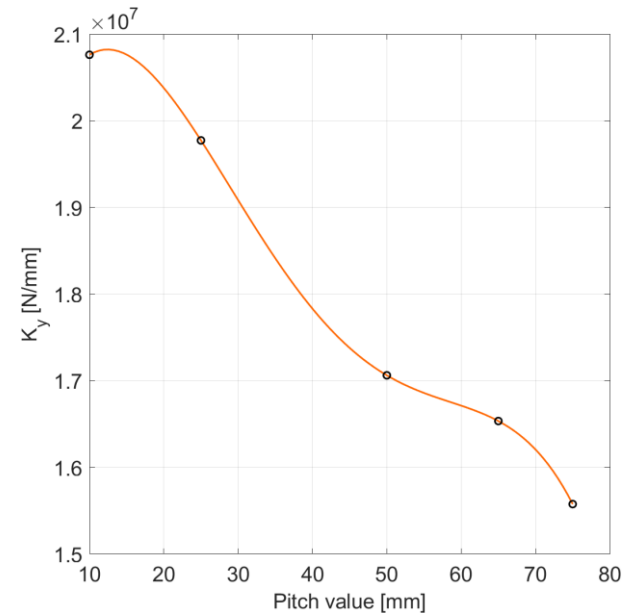
Case	Pitch value	Bending - z		Bending - x		Torsion - y		
		Δz	$K_{b,z}$	Δx	$K_{b,x}$	Δz_t	ϑ_t	$K_{t,y}$
	[mm]	[mm]	[N/mm]	[mm]	[N/mm]	[mm]	[N/mm]	[Nmm/rad]
a	10	2,875	34,78	0,204	490,20	2,60E-03	4,82E-05	2,08E+07
b	25	2,944	33,97	0,2127	470,15	2,73E-03	5,06E-05	1,98E+07
c	50	3,022	33,09	0,2276	439,37	3,17E-03	5,86E-05	1,71E+07
d	65	3,064	32,64	0,231	432,90	3,27E-03	6,05E-05	1,65E+07
e	75	3,066	32,62	0,2321	430,85	3,47E-03	6,42E-05	1,56E+07

PARAMETERS INFLUENCE

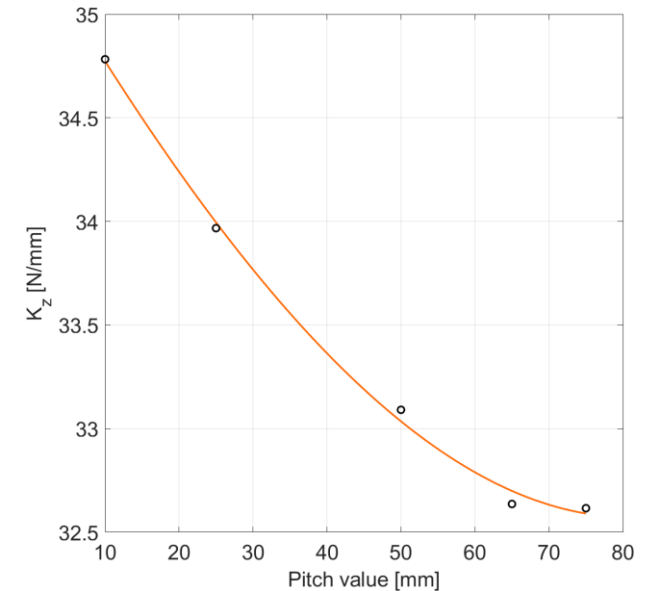
BENDING – $K_{b,x}$



TORSION – $K_{t,y}$



BENDING – $K_{b,z}$



Overall, **reducing the pitch distance** results in **increased stiffness** in all directions.

CONCLUSIONS



Torsion: The results indicate exceptionally **high torsional stiffness** along the **Y-axis**, demonstrating strong resistance to torsional loads.



Bending: Along the **X-axis**, the bending stiffness **is significantly higher** than along the **Z-axis**, indicating that the T-joint is better suited to withstand frontal stress compared to lateral stress.



Pitch value: **increasing the number of spot welds enhances the stiffness** of the structure in both torsional and bending scenarios. However, it is important to note that the correlation is not linear.