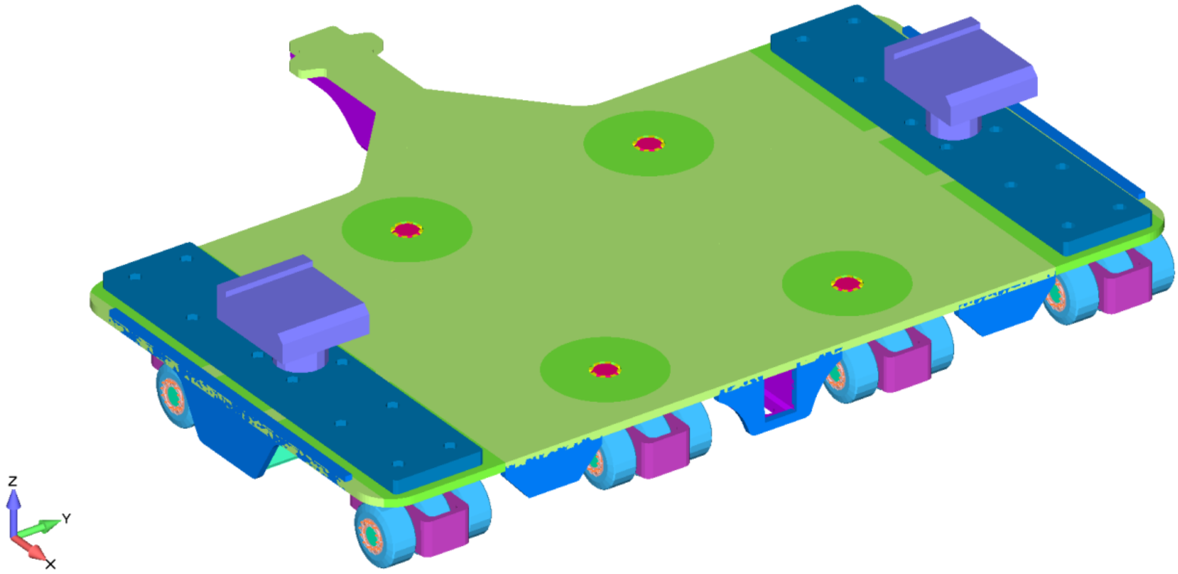

Strength Evaluation CHASSIS 10t P1

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Contents

1 Summary

- A static strength assessment using local stresses according to FKM-Guideline (6th edition) was created.
- Stainless steel (1.4301) was assumed for all structural parts.
- The chassis assembly has to carry a load of 10 tons.
- The degree of utilization according to FKM-Guideline is lower than 0,21.
- Hence, the strength for static load, i.e. up to several hundreds of repeated charges, is verified.
- The strength of the structure is limited by two radii at the mounting pins and the base plate locations in contact with mounting pins.
- Weld seams between mounting pin and base plate as well as reinforcement ribs and U profiles are overloaded but structurally irrelevant.
- Whereas the structure is not fully utilized, there is a slight plastic deformation at the connection of mounting pin and plate. We do not expect a change of pin normality outside the range of tolerances.
- M10 bolts are not overloaded within the range of static strength assessment.

2 Introduction

The current chassis is utilized for horizontal transportation of loads up to 10 tons, which are positioned centered between two lifting cylinders.

The chassis represents machinery in the sense of EU machinery directive (95/16/EC of 17 May 2006). CE marking requires an EU declaration of conformity. Among other things it has to be insured appropriate strength of all load carrying parts of the carriage. This will be verified by means of a strength evaluation according to FKM-Guideline (6th edition).

The load is applied and removed several hundred times only over the lifetime of the chassis. Therefore an assessment of fatigue strength is not necessary. Only a static strength assessment using local stresses will be performed, where the stresses are calculated with means of the finite element method.

4 Finite Element Model

4.1 Modeling Concept

All FE analysis is performed using NX Nastran 11.2 as solver, mainly SOL 106 linear static solution. For evaluation of plastic limit load SOL 601 is used, assuming ideal-plastic stress-strain curves for the material.

The welded assembly, comprising base plate, U-shaped profiles and reinforcement ribs, is meshed using 4-node shell elements of type CQUAD4. The base plate is meshed in part around all plugged-in bolts using 8-node hexahedral elements. Those solid elements are connected to the shell elements by Edge-to-Surface Gluing.

All parts of the wheel cassettes are meshed using linear hex elements, whereas all bearing rollers are modeled using combinations of gap and spring elements.

Connection between different components are usually modeled as small sliding contact, friction coefficient 0,12. The only exceptions are the inner bearing rings, which are connected to the corresponding bolts by mesh gluing.

M10 bolts between base and mounting plate are modeled as a combination of RBE2- and beam-elements (VDI-Guideline 2230, model class II), where effective cross section area is used for the beam diameter.

The applied load consists of discrete mass element of 10 tons distributed via RBE2-element between the two lifting devices. The total load consists of the applied load plus chassis weight. Two wheel positions will be evaluated: 0° and 90°.

Preload of 16.300 N for M10 stainless steel bolts is applied.

The whole structure is constraint in the wheel support points.

Figure 1 to 3 give an overview of the chassis mesh. Table 1 shows the element statistics.

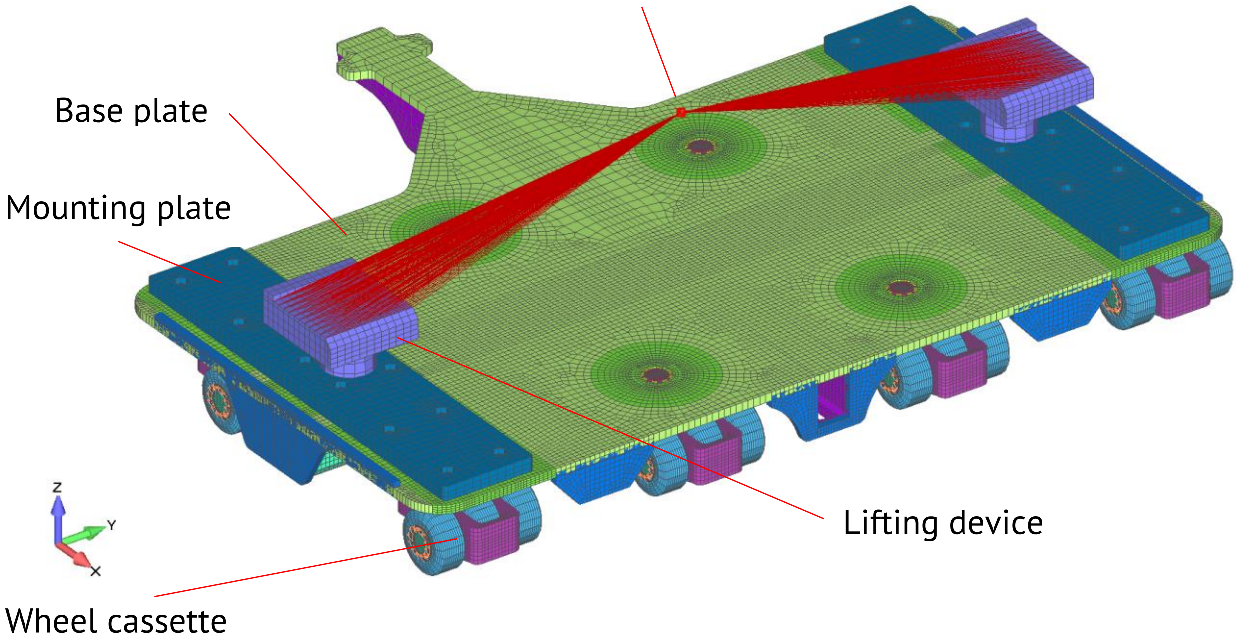

Figure 1: Mesh overview and naming of parts. Shell thickness is displayed.

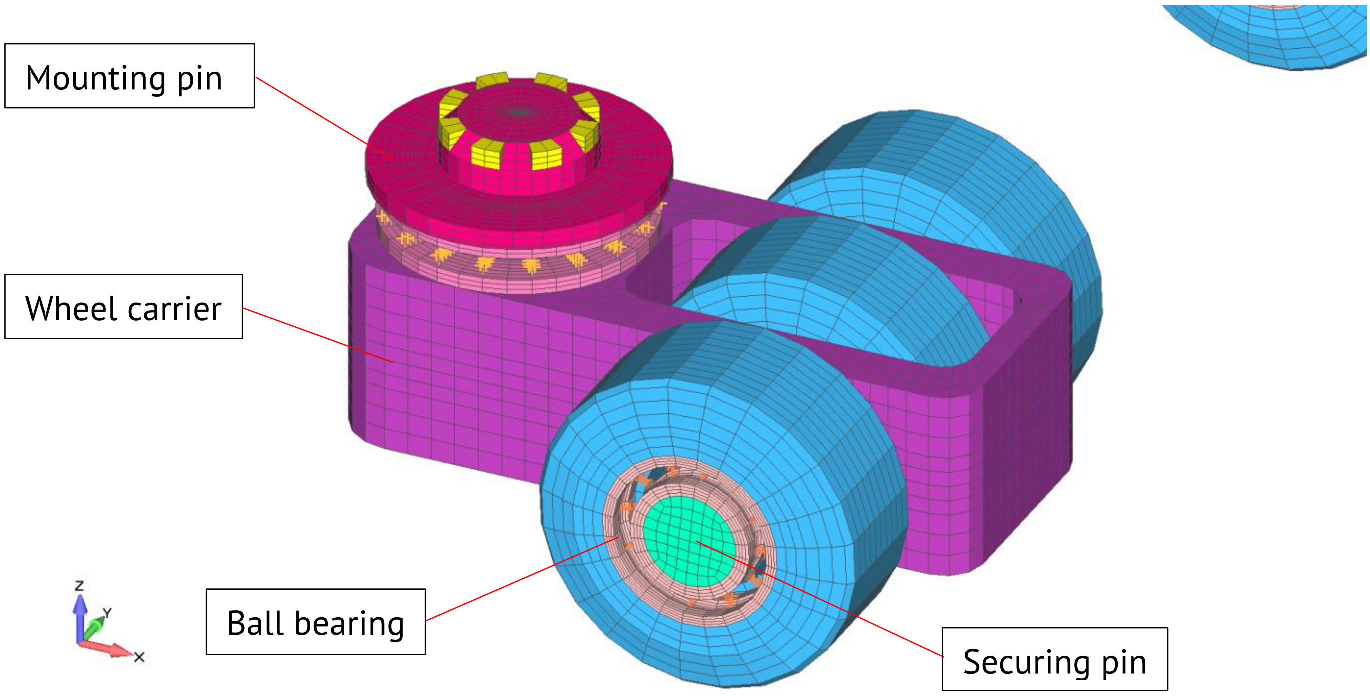

Figure 2: Meshing detail of a wheel cassette

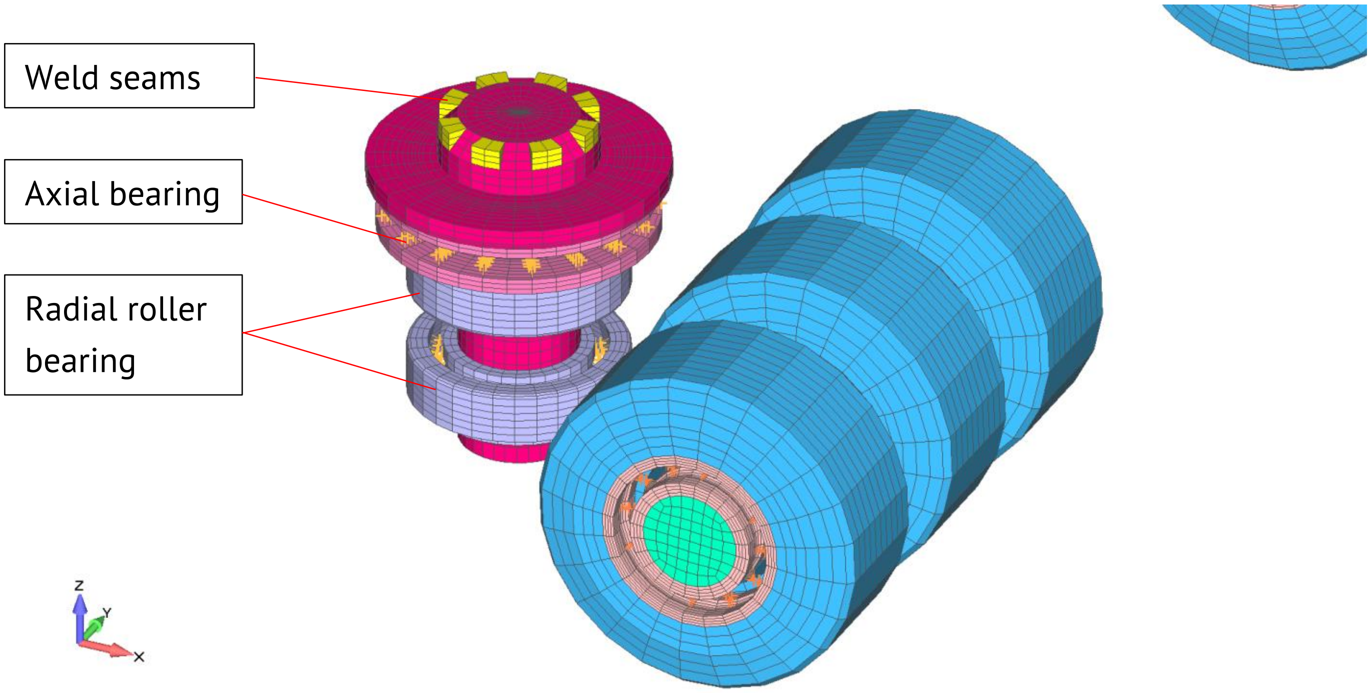

Figure 3: Meshing detail of a wheel cassette

Nodes	546.076
CQUAD4	23.788
CHEXA	401.756
CGAP	7.326
CBEAM	24
CBUSH	7.424
RBE2, RBE3	49
CONM2	1

Table 1: Node and element statistics of finite element mesh

4.2 Material Properties

For all evaluated components stainless steel 1.4301 is assumed. For ball and roller bearings we use non-specified steel properties.

For all linear analysis a Hookean material model is used with data according to Table 2.

For estimation of plastic limit load a nonlinear static analysis of a sub-model (base plate plus mounting pin) is performed assuming ideal-plastic stress-strain behavior: nearly horizontal curve above yield stress of 220 MPa.

Parts	Weld assembly, mounting and securing pin, wheel carrier	Bolts	Roller bearings	Wheels
Material	**1.4301**	**A2-70**	**Steel**	**PU**
Density [kg/m^3]	7.850	7.850	7.850	1.400
Elastic modulus [MPa]	200.000	200.000	200.000	5.000
Poisson number ν	0,3	0,3	0,3	0,4
Yield strength R_p [MPa]	220	450		
Tensile strength R_m [MPa]	520	700		
Elongation at break A [%]	45			

Table 2: Assumed material properties

5 Results of FE Analysis

5.1 0° position of wheel cassettes, linear-elastic analysis

Figure 4 und Figure 5 show the stress distribution of the chassis. The whole structure is moderately charged. Higher loads arise at the outer front wheel cassettes.

Details of stresses at critical locations are shown in Figure 7.

Four critical points can be identified:

1. Weld seam between reinforcement rib and U-profile (Figure 4) with structural stress of 340 MPa
2. Lower radius (0,5 mm) with stress 560 MPa
3. Upper radius (0,5 mm) with stress 700 MPa
4. Upper edge of base plate with stress 335 MPa

The weld seam stresses are too high. But failure of the weld does not lead to a breakdown of the structure. The reinforcement ribs can carry longitudinal loads anyway.

Axial equivalent stress in M10 bolts, neglecting bending moments, are below yield strength of 450 MPa for A2-70 stainless steel bolts (Figure 8: Equivalent stress [MPa] M10 bolts (ignoring bending moments). So, static strength of the bolts is verified.

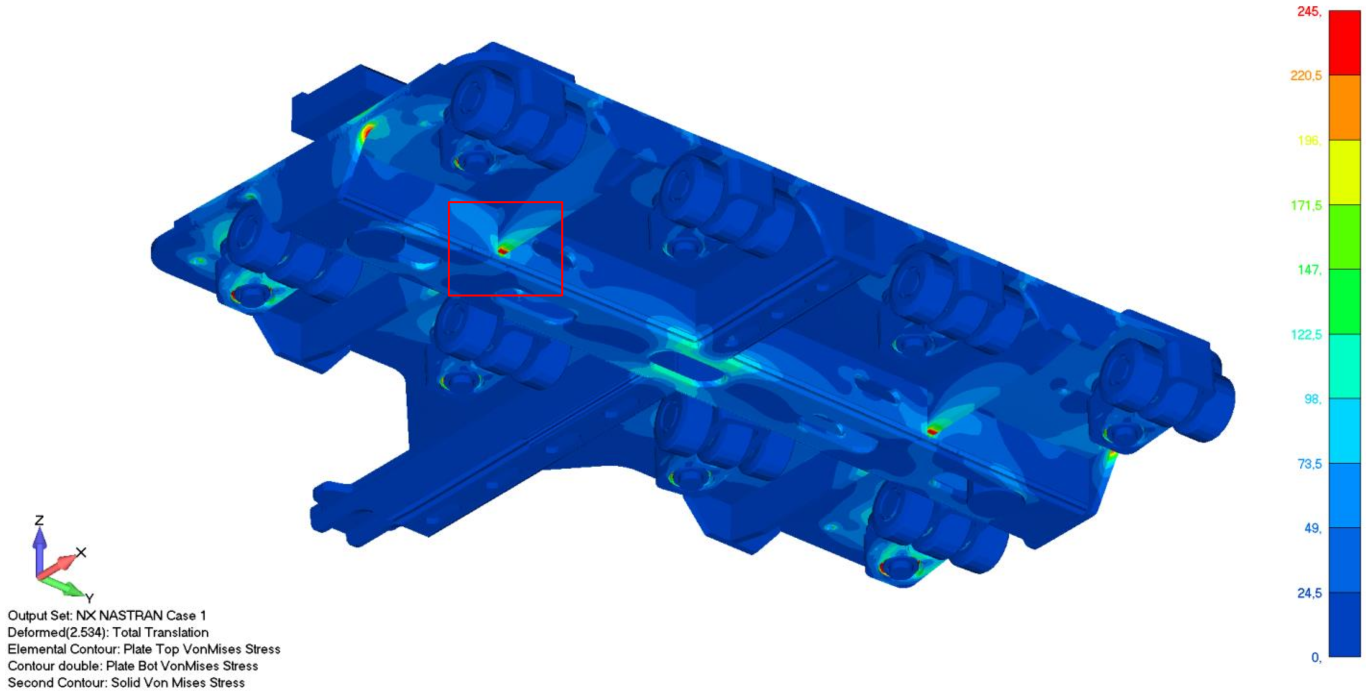

Figure 4: Equivalent stress [MPa]. Shell thickness displayed, displacements 10 times enlarged

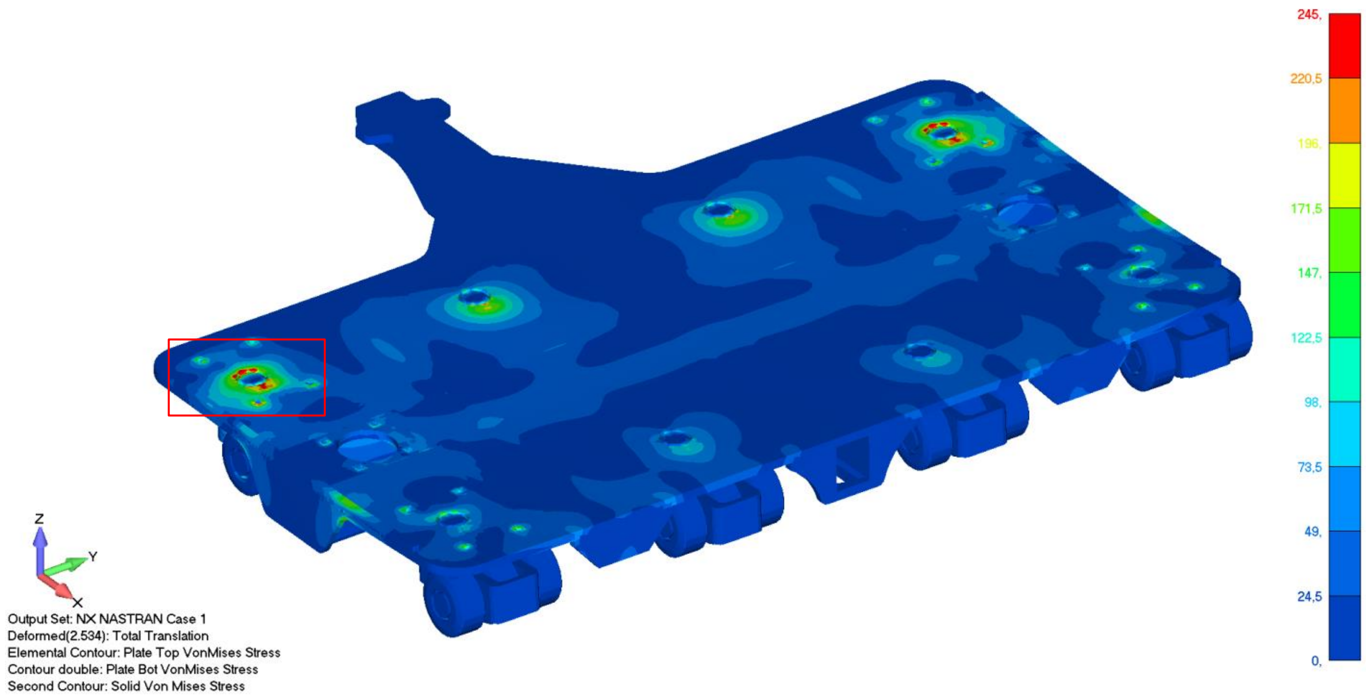

Figure 5: Equivalent stress [MPa]. Shell thickness displayed, displacements 10 times enlarged

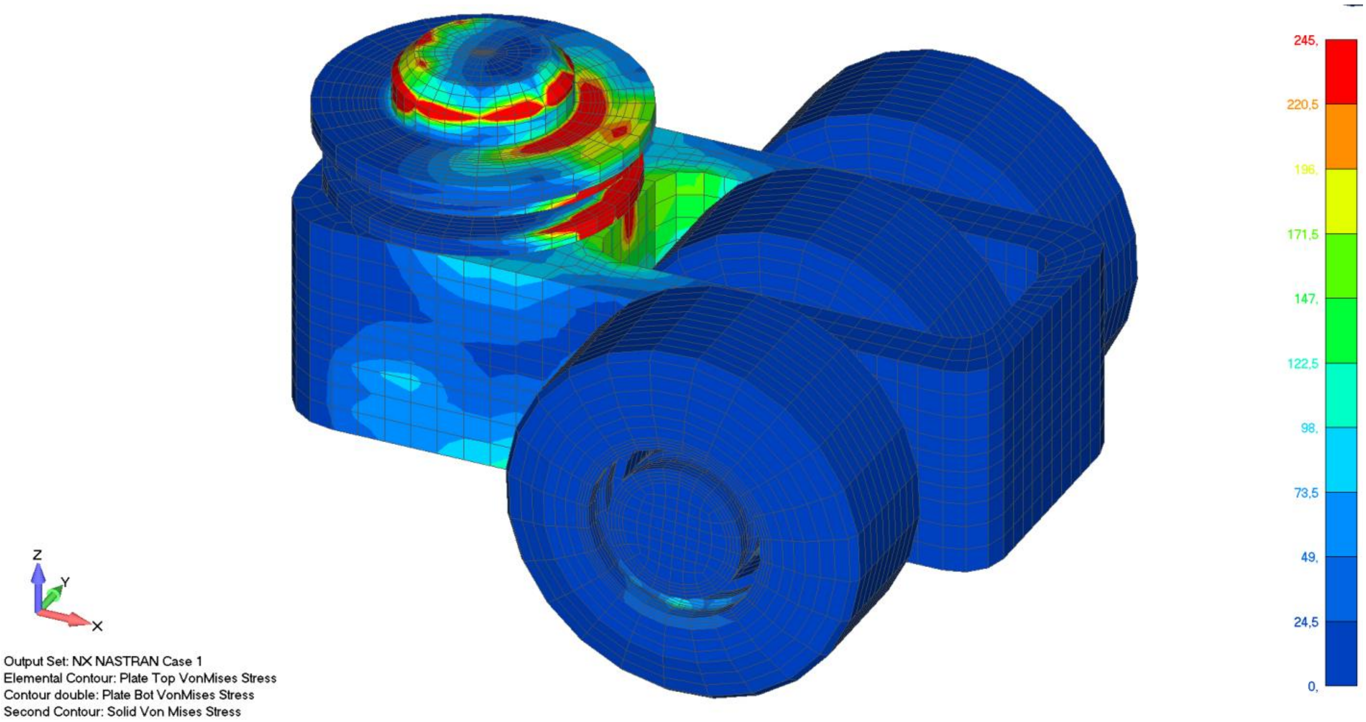

Figure 6: Equivalent stress [MPa], detail highest loaded wheel cassette

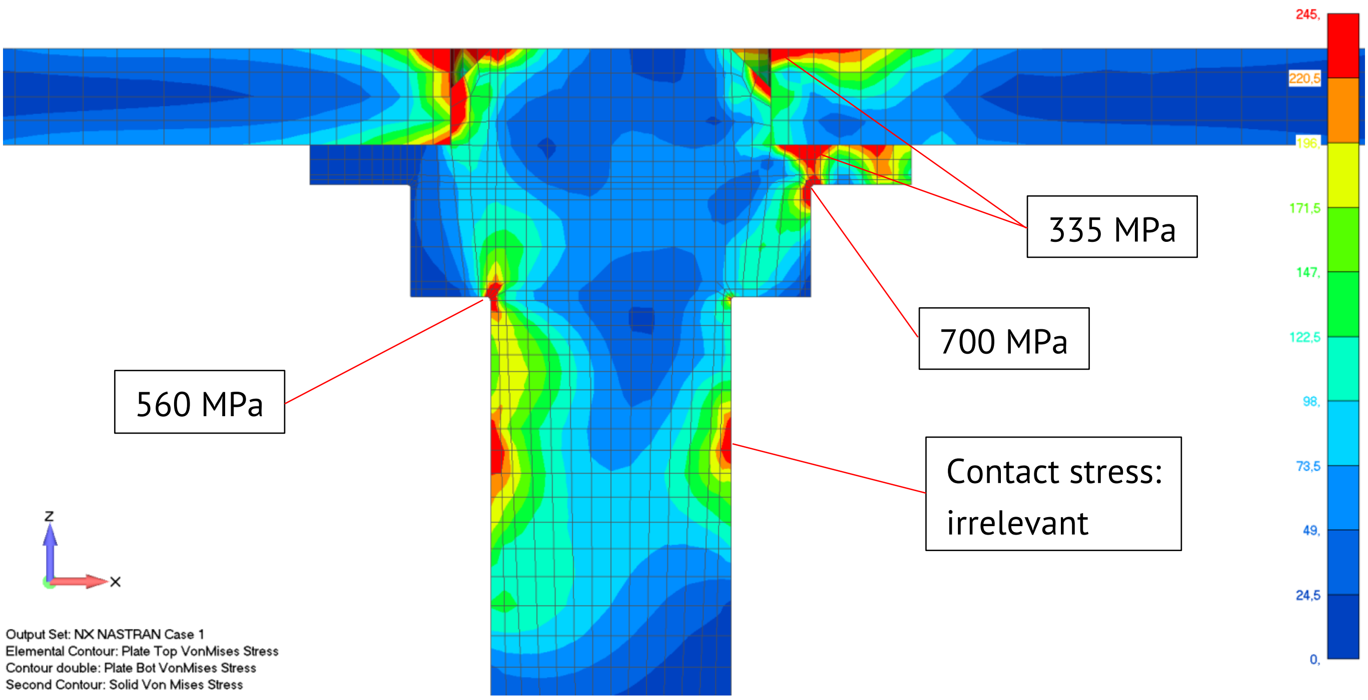

Figure 7: Equivalent stress [MPa], cross section mounting bolt and base plate

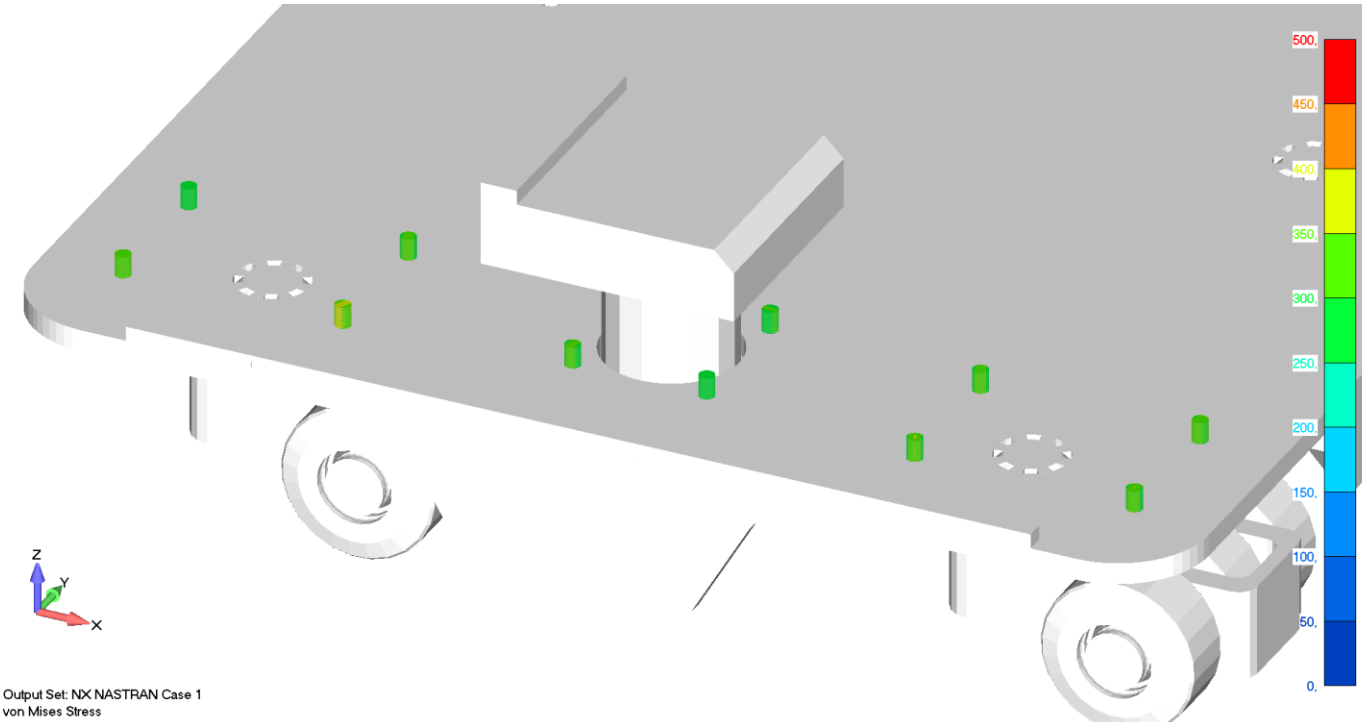

Figure 8: Equivalent stress [MPa] M10 bolts (ignoring bending moments)

5.2 90° position of wheel cassettes, linear-elastic analysis

The critical points in this position are the same as in 0° position. The overall stresses are a bit lower because of more even load distribution on outer wheels.

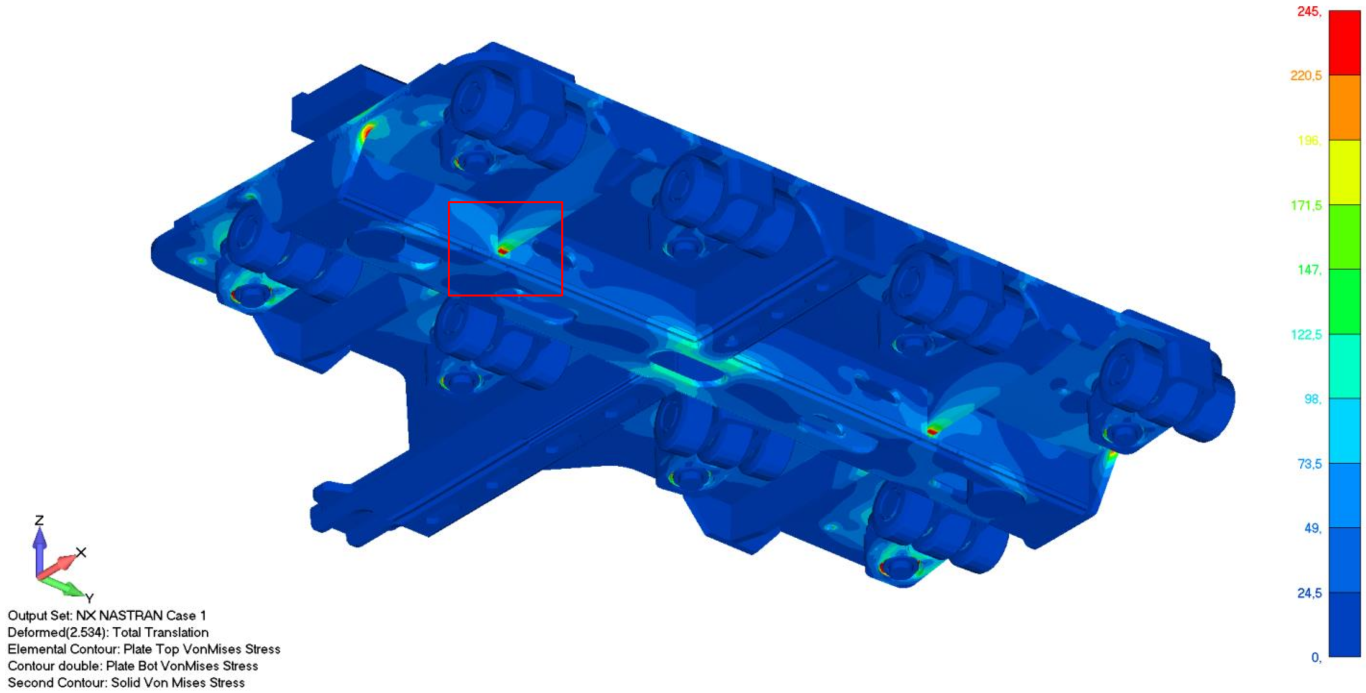

Figure 9: Equivalent stress [MPa]. Shell thickness displayed, displacements 10 times enlarged

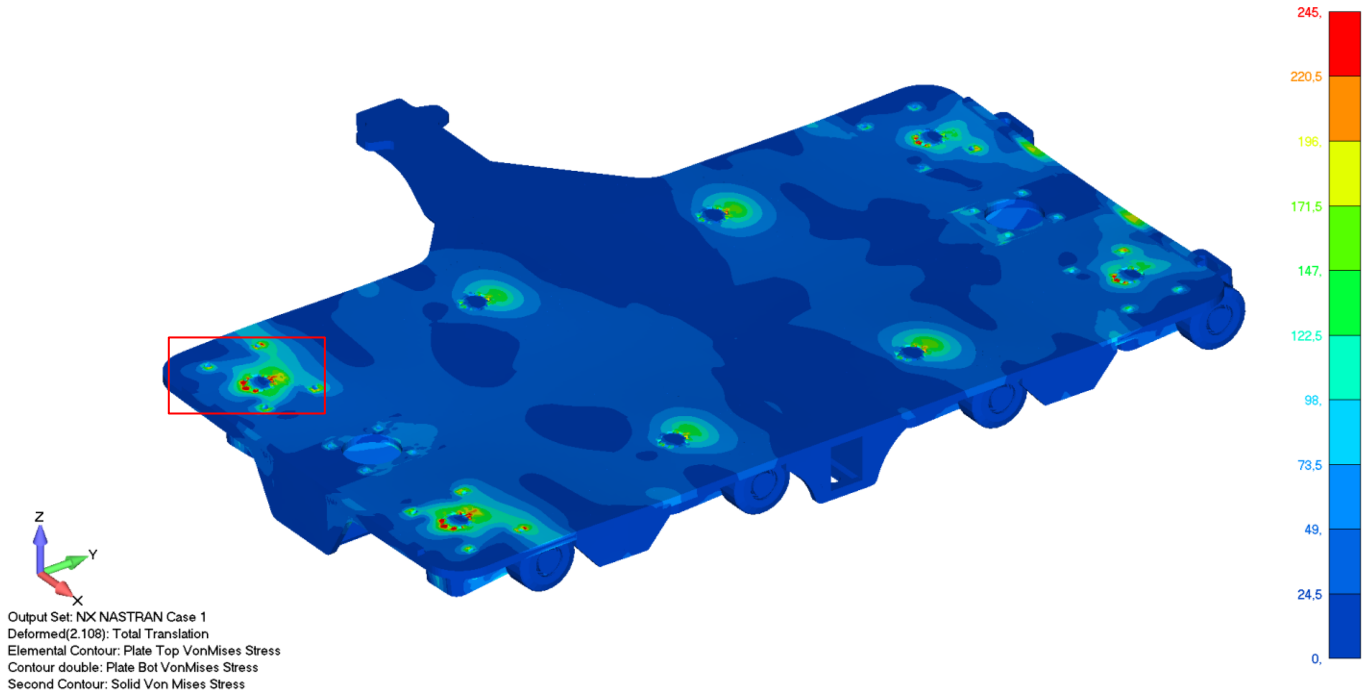

Figure 10: Equivalent stress [MPa]. Shell thickness displayed, displacements 10 times enlarged

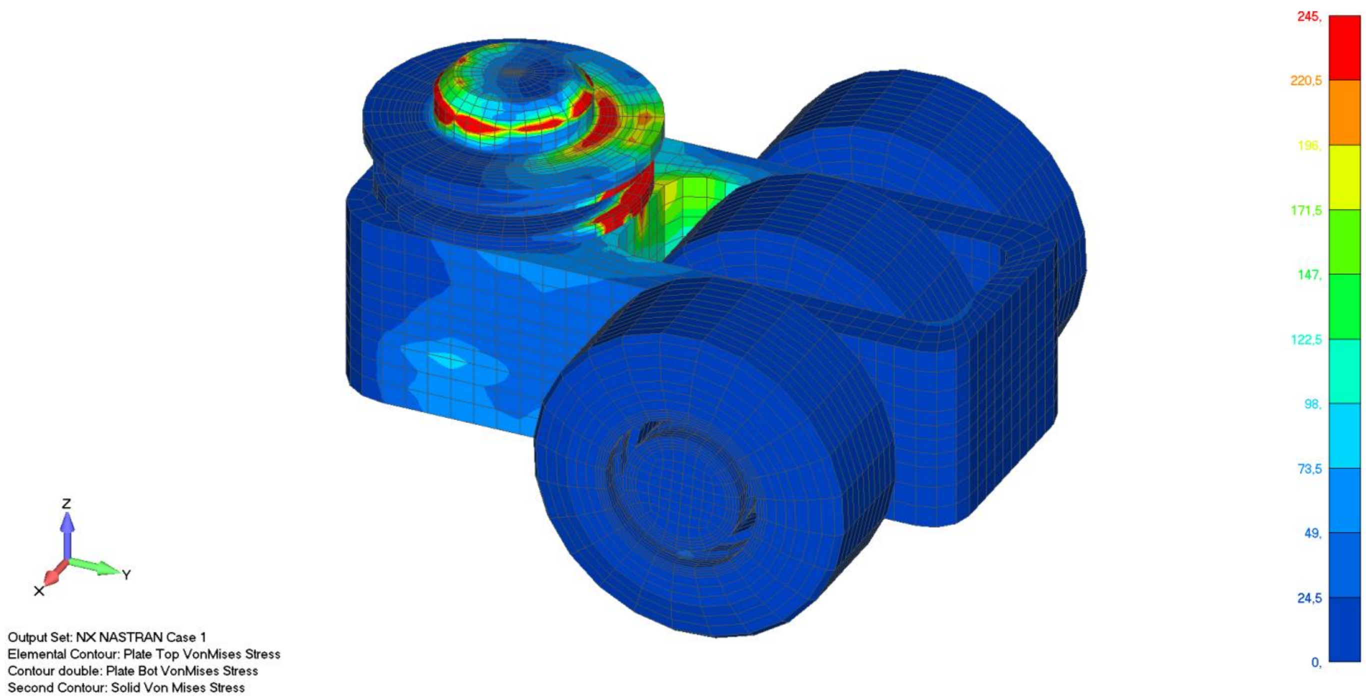

Figure 11: Equivalent stress [MPa], detail highest loaded wheel cassette

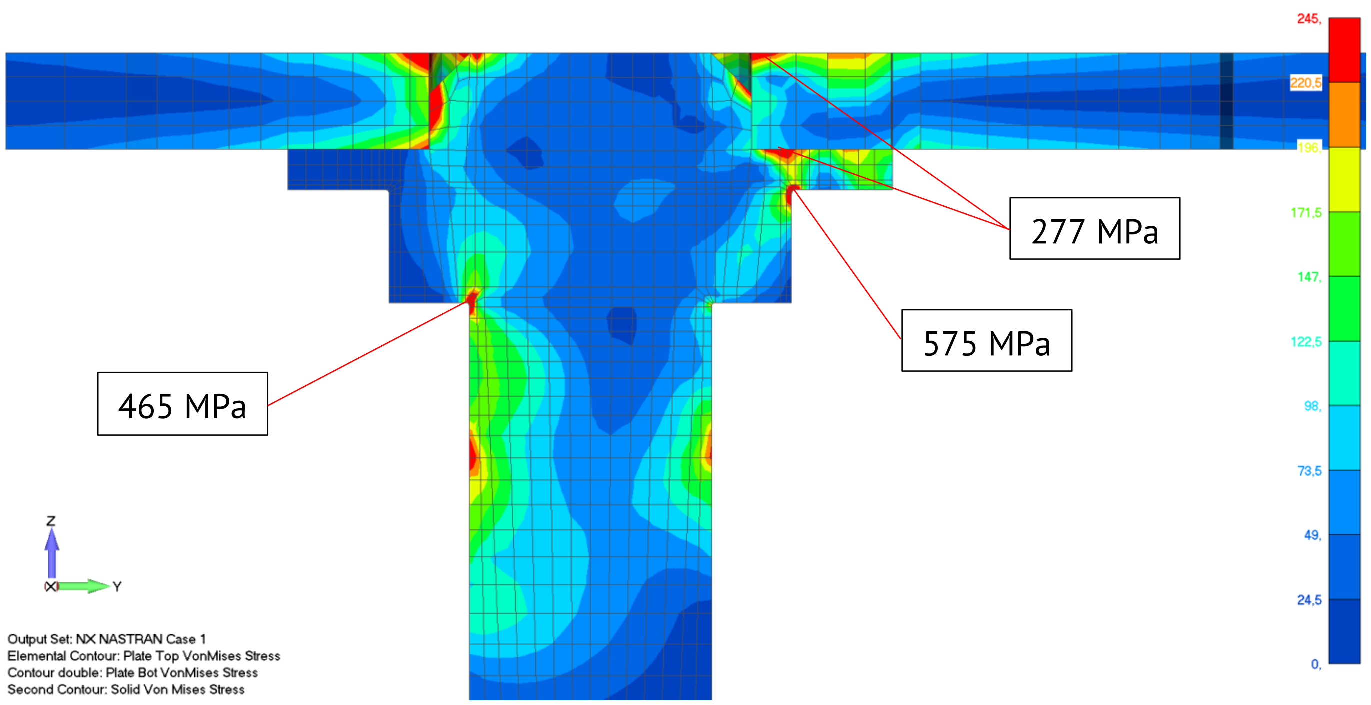

Figure 12: Equivalent stress [MPa], cross section mounting bolt and base plate

5.3 Results of nonlinear analysis

For a subassembly consisting of base plate cutout and the most heavily loaded mounting pin a FE submodel was created. Loads out of linear-elastic analysis (1 g gravity) for 0° wheel position were mapped onto the submodel.

Ideal-plastic material model was assumed for 1.4301, which means linear stress-strain curve up to 220 MPa and horizontal progression beyond.

It is not quite clear whether the heavily loaded locations on the mounting pin and plate (Figure 7) can lead to local or global failure. Therefore the goal of this analysis is to evaluate plastic limit load of the structure.

So a nonlinear static analysis is performed by increasing gravity from 0 to 15 g. Elastic limit of the structure is reached with gravity 0,31 g (220 MPa / 700 MPa). The results show no plastic collapse up to 15 g.

Hence, plastic notch factor (plastic limit load divided by elastic limit load) is:

$$K_p > 48$$

Figure 13 and Figure 14 show total strain for a load level of 15 g. Strains are below 10% except for weld seams.

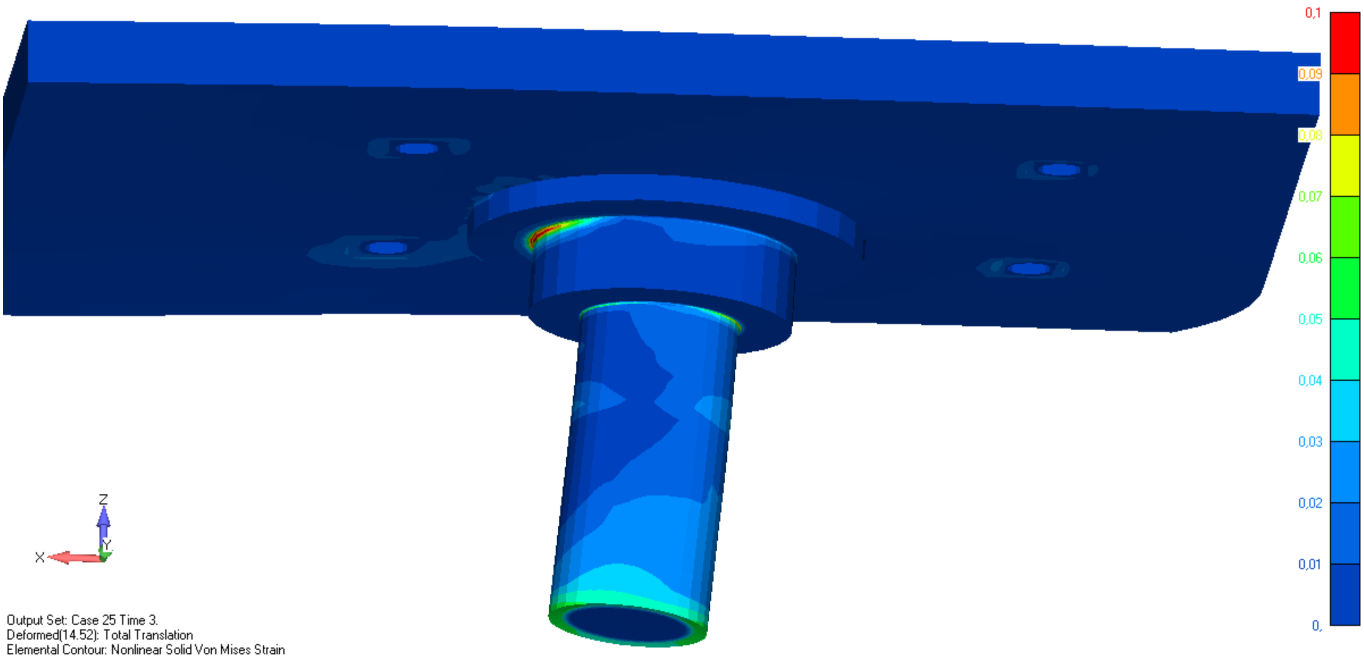

Figure 13: Total strain for gravity 15 g

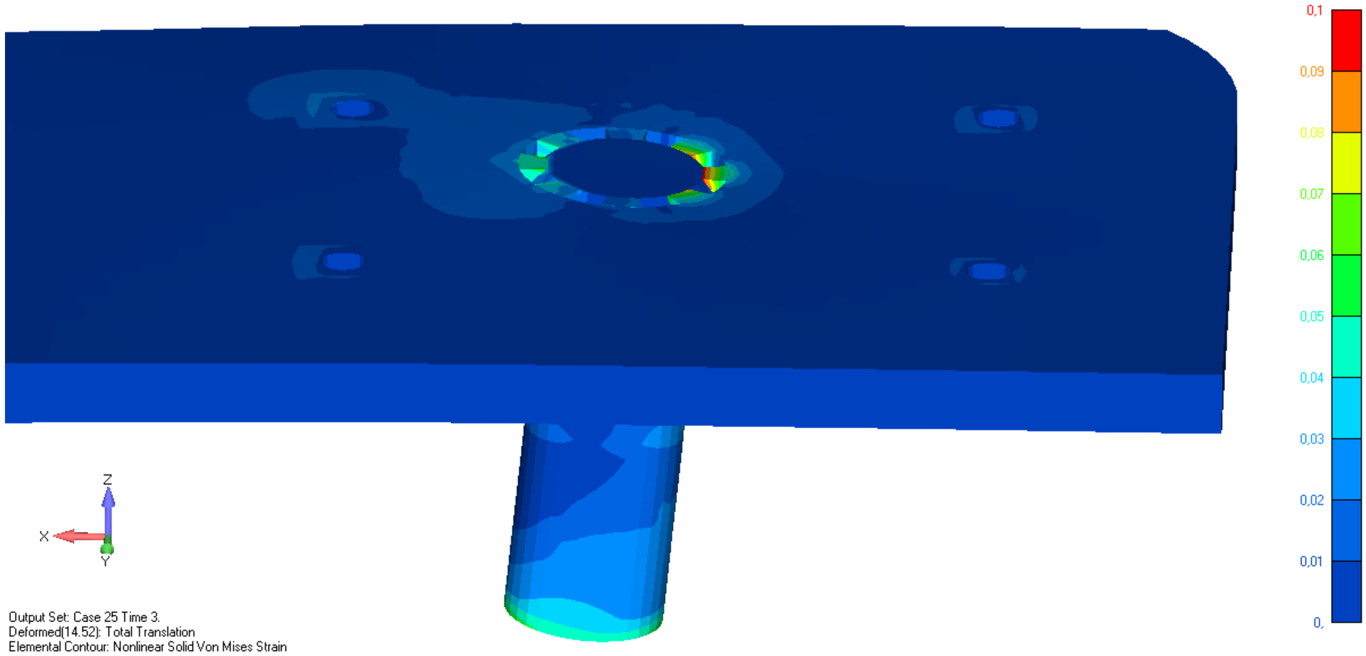

Figure 14: Total strain for gravity 15 g

6 Static Strength Evaluation according to FKM-Guideline

6.1 Introduction

We use the three most charged critical points of the structure (0° position) for this verification. The respective maximum equivalent stresses at these points are the relevant measures of the stress level.

The weld seam between reinforcement rib and U-profile will not be evaluated. It is rather likely that this weld will fail. But this is of no relevance to the strength of the overall structure. The ribs are continuously fed through the profiles and carry longitudinal loads in any case.

The same is true for the welds between mounting pins and base plate. They are considerably overloaded. But the pins will carry their load anyway.

Table 3 summarizes stress parameters at those critical points.

Critical point		Relevant Stress Value [MPa]	
1	Lower radius mounting pin	560	Bending stress, multiaxiality factor 0,87
2	Upper radius mounting pin	700	Bending stress, multiaxiality factor 0,49
3	Upper edge base plate	337	Bending stress, multiaxiality factor 0,54

Table 3: Critical points to be evaluated

6.2 Material properties

Yield strength for 1.4301 is

$$R_p \; = 220 \; MPa$$

Technology size factors do not have to be considered in the case of stainless steel. So this is the characteristic strength value for all parts to be evaluated besides M10 bolts.

Temperature application range is between -40 °C and 100 °C. So temperature factors are equal to unity.

6.3 Design factors and component strength

The strength of a non-welded part is:

$$\sigma_{SK} \;=\; R_p \cdot \; n_{pl}$$

where n_{pl}, the section factor, is a measure for strength increase in taking advantage of plasticity reserves. This value is defined on the one side by material resistance against fracture at the ground of charged notches (**local failure**). On the other side it is limited if a stress bearing cross-section is in the state of plastic flow in its whole (**global failure**).

$$n_{pl} \;=\; MIN\;(\sqrt{E \; \cdot \varepsilon_{ertr}/R_p} \;\; ; \; K_p)$$

Here ε_{ertr} is the (material and load depending) limit value of total strain. K_p is the plastic notch factor, the quotient between plastic limit load and elastic limit load.

At the three critical locations, we calculated K_p to be higher than 48. So it is assumed that the dominating failure mechanism would be local failure by fracture growth.

At the two critical points on the mounting pin with multiaxiality of 0,87 and 0,49 limit strain values are 69% and 51%. So section factors are $n_{pl} \; = \;$ 25,0 and 21,5 respectively and the corresponding linear-elastic limit stresses are 5.500 MPa and 4.730 MPa.

Critical point		Relevant Stress Value [MPa]	
1	Lower radius mounting pin	5.500	Limited by local fracture
2	Upper radius mounting pin	4.730	Limited by local fracture
3	Upper edge base plate	4.840	Limited by local fracture

Table 4: Minimum strength values at critical locations

At the upper edge of base plate with multiaxiality of 0,54 one gets $\varepsilon_{ertr} = 53\%$, n_{pl} =22,0 and elastic limit stress 4.840 MPa.

In summary local and global failure of the stainless steel parts is very unlikely.

Critical point		Relevant Stress Value [MPa]	
1	Lower radius mounting pin	5.500	Limited by local fracture
2	Upper radius mounting pin	4.730	Limited by local fracture
3	Upper edge base plate	4.840	Limited by local fracture

Table 4 shows minimum strength values related to linear-elastic theory.

6.4 Safety factors

It is assumed that probability of the occurrence of service stress level is high and consequences of failure are moderate. Than a basic safety factor of

$$j_p \quad = 1{,}4$$

can be assumed.

6.5 Assessment

For the three critical locations maximum degree of utilization can be calculated and are represented in Table 5. So the structure of the chassis is far to be overloaded by applying a charge of 10 tons.

Critical point		**Degree of utilization**	
1	Lower radius mounting pin	**0,14**	Limited by local fracture
2	Upper radius mounting pin	**0,21**	Limited by local fracture
3	Upper edge base plate	**0,10**	Limited by local fracture

Table 5: Maximum degree of utilization at critical locations