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INTERNSHIP REPORT
Analysis of a generic lifting table

Ruben Teunis
s1112392
Mechanical Engineering
Applied Mechanics, CTW

25-08-2014 – 5-12-2014

UNIVERSITY OF TWENTE.

Enschede, The Netherlands
Supervisor: T. Tinga

Hengelo, The Netherlands
Engineering Analysis
Supervisor: G.J. de Jonge

Preface

From August until December 2014 I did an internship at Viro Engineering located in Hengelo. This internship was part of my two year Master program applied mechanics at the University of Twente.
I worked at the department of Engineering Analysis. This department performs all kinds of numerical and analytical calculations in order to assess the performance of a machine or construction from clients.
The work that is done in this department is very diverse. I worked for at least four different clients and the projects ranged from designing lifting tables to clue clamps and even ballistic separators. During my internship I worked on many projects and met a lot of inspiring people. The project described in this report represents actually only a small amount of the work I did during my internship.

Hereby, I would like to thank everybody who made this internship a valuable experience. I really learned a lot and especially liked that I was treated as a full employee in every way. Above all I would like to thank Gert-Jan de Jonge and Tom Ormel. Gert-Jan for his guidance and sharing his extensive knowledge and experience. His view on a lot of things really got me thinking sometimes. Tom, as my neighbor, for always making time for all the questions I had and, during the lifting tables project, for working together in a great way on the same project.

Summary

This report covers the assessment of the performance of an existing lifting table. The aim is to redesign this table for generic use and according to the European standard for lifting tables complemented with some additional requirements from the client and Viro.

Design criteria
The design criteria are subdivided into three parts. Requirements from the client, European Standard and other requirements.
Client:
The client requires that the geometry from an existing table is used as starting point. Furthermore, The working limit load of 500 kg for general purposes, used materials and the table height ranging from 300 to 1300 mm are also specified by the client.
European standard:
The European standard states a dynamic factor of 1.4, horizontal load coefficient of 10% and number of cycles of 128000. The platform deflection is limited to 2% over the long side and 2.48% over the short side for specific cases. The stability should have a safety factor of minimal 1.3 also for specific cases. The allowable primary stress should be the minimum of 0.66xyield limit or 0.5xultimate strength. Furthermore secondary stresses and fatigue stresses are taking into consideration.
A horizontal acceleration of 0.5 m/s^2 and horizontal operator force of 300 N should be taken into account. At last, minimal gaps of the moving parts are specified.
Additional requirements:
Stresses are divided into a primary and a secondary category. This is done in order to be able to assess the results from the FEM analysis in the right way. The fatigue strength and weld connection strength are calculated. It turns out that these strengths are higher than the lowest allowable stress.

Gaps
The geometry is assessed with respect to the safety gaps. It turns out that the gaps between the scissors, the gaps between the bottom deck and the scissor leg, and the gap between the upper deck and the scissor leg do not meet the requirements according to the European standard.

Platform deflection
The platform deflection is assessed using a finite element model. It turns out that the deflection is too big in 2 out of 16 cases.

Stability
The stability is also assessed using the same finite element model. It turns out that 6 out of 8 cases are not stable.

Analytical verification
Finally, the forces at some point in the lifting table are also calculated with an analytical calculation. The mean deviation between the analytical and numerical model is about 15%.

1 Table of content

2 Introduction

A stainless steel lifting table, which is currently in use by a client of Viro, should be redesigned for generic use. This report contains the analysis of the performance of the existing lifting table. The redesign is assessed according to the European standards. First, the requirements of generic lifting tables are analyzed. Thereafter, a model is made to verify whether the current lifting tables satisfy the stated norms or not.

Since the geometry of the lifting tables is complicated it is chosen to use finite element modeling techniques to verify the structural integrity and stability of the table. The calculated stresses in volumetric components of the lifting tables will show peak stresses which will not occur during hand calculation. Therefore, stresses will be categorized into a primary and secondary type.

The goal of this analysis is to give a first indication into the structural integrity and stability of the lifting table. Load cases will be described using the European standards for lifting tables. Finite element calculations will be performed and the model will be verified with a numerical calculation. The aim is to propose useful recommendations for this table that can be implemented in order to fulfill the requirements according to the European standard.

3 Design criteria

The design criteria are derived from different input sources. The first source of requirements are the requirements from the client. These are for example the geometrical and the load carrying capacity requirements of the table. The second source is the European standard NEN-EN 1570-1. This standard is also referred to as the lifting table norm. The lifting table norm specifies the performance with regard to stability, platform deflection and geometry. The third source are additional requirements determined by Viro. For the assessment of the stresses EN 13445 is used. The fatigue strength is assessed using EN 1993-1-9 and the welded connections are assessed using EN 1993-1-8.

3.1 Client requirements

The maximum working limit load (WLL) of the lifting table is 500 [kg]. This table is used for generic purposes. The weight of the table is not specified. The height of the table is specified to range from 300 to 1300 [mm]. The way in which the load is applied is specified later in this section.

3.1.1 Material data and stresses

The material data for the metal components are presented in table 1. The materials are also specified by the client. Materials can be changed, but this should be done with approval of the client.

Material	***[-]***	***AISI 304 1.4301***	***42CrMo4***
Ultimate strength	[MPa]	520	900
Yield strength	[MPa]	210	650
E-modulus	[MPa]	195000	200000
Density	[kg/m^3]	7950	7950
Elongation at break	[%]	35	12
σ_m	[MPa]	140	433.3
σ_l	[MPa]	210	650
σ_b	[MPa]	210	650
σ_q	[MPa]	420	1300

Table 1: Material data

3.2 Requirements according to NEN-EN 1570-1:2011

The project description states that the lifting table should satisfy the European standard NEN-EN 1570-1. The most important requirements this norm states will be considered in the following subsections.

3.2.1 Lifting category

The SST-table is classified in Category 1 according to EN 1570-1: 2011 §5.1.2.4 Table 2. (Category 1: Lifting tables for general purpose). For this category the following holds:

- Dynamic factor 1.4
- Horizontal load 10% (moving the COG of the load)
- Number of cycles 128000

3.2.2 Platform deflection

The lifting table shall be designed to meet the following requirements regarding platform deflection:

- In one case lift half the rated load distributed over half the length of the platform.
- In another case lift one third of the rated load distributed over half the width of the platform.
- In Neither case shall hazardous tilting or deflection of the upper deck take place. The maximum tilting or deflection shall not exceed:
- Y=1+(2-l)%, For longest side 2%, for short side 2.48%.
- (See for reference EN 1570-1:2011 §5.1.2.7.1).

3.2.3 Stability

The lifting table shall be designed to meet the following requirements regarding stability:

- Safety minimum 1.3 (EN 1570-1:2011 §5.1.3.1).
- For the calculation the rated load shall be evenly distributed over an area half the length times half the width of the platform, in any overturning position.

3.2.4 Stresses

The lifting table shall be designed to meet the following requirements regarding stresses:

- Static: The maximum allowable primary stress is the minimum of:
  {0.66 x yield limit; 0.5 x de ultimate strength} (EN 1570-1:2011 §5.1.2.2):
- Fatigue: the fatigue strength at 128000 cycles. (1 cycle = lifting and lowering).

3.2.5 Accelerations

The lifting table must withstand a horizontal acceleration of 0.5 [m/s^2] during maximum loading in the lowest position according to EN 1570-1: 2011 §5.3.2. This section implies a maximum travel speed when the table is not in its lowest position, however for this case no requirements regarding allowable accelerations are being made. As starting point for the load calculations the conservative assumption is being made to apply 0.5 [m/s^2] in all positions of the table. Seismic loads are excluded in calculations described in this report.

3.2.6 Design conditions

The lifting table is used in clean room conditions:

Temperature	: 0 to +40 [º C]
Floor slope	: ≤ 2.5 [%] (EN 1570-1:2011 §5.1.3.4).
Horizontal acceleration	: 0.5 [m/s^2] (EN 1570-1:2011 §5.3.2).
Table configuration	: On castor wheels
Operator force	: 300 [N] (horizontal force according to EN 1570-1:2011 §5.6.12).

3.2.7 Safeguarding

The lifting table norm states with respect to safeguarding: “Generally crushing and shearing shall be avoided by the following minimum gaps between moving parts and between moving and fixed parts which are in reach of persons on the platform or standing adjacent to the travel zone.” (EN 1570-1:2011 § 5.2.1):

- For fingers, 25 mm
- For toes, 50 mm
- For hands, 100 mm
- For arms and closed hands, 120 mm
- For feet, 120 mm
- For the body, 500 mm

In scissor type lifting tables, the minimum safety gaps between the scissor legs shall not be less than 30 [mm] and the distance between scissor leg and the inside of the base-frame member shall not be less than 50 [mm], unless rigid or flexible protection is fitted (EN-1570-1:2011 §5.2.3). See figure 1. The requirements mentioned in this paragraph are discussed in chapter 4.

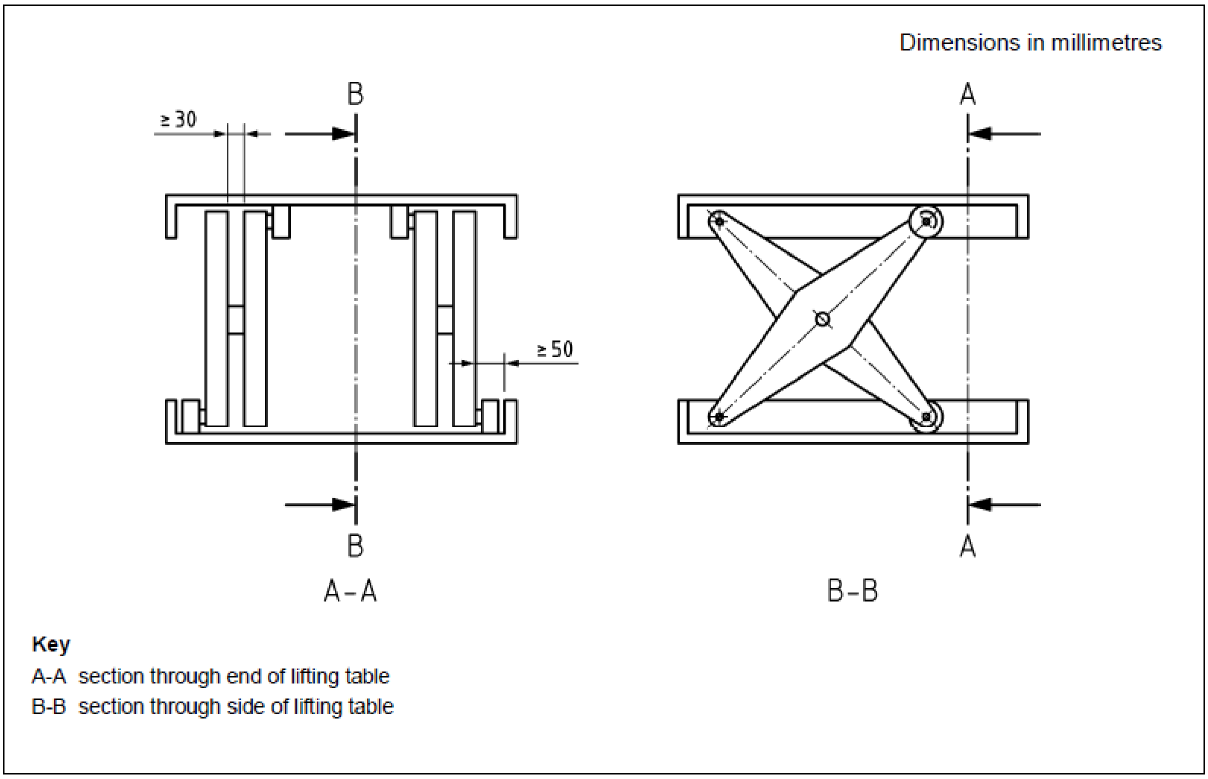


Figure 1: Gaps between lifting table arms and between arms and base

3.3 Requirements according to other norms

Some properties are not defined by either the client or the lifting table norm. In this subsection requirements to some additional properties are described. The source of these requirements is chosen by Viro.

3.3.1 Allowable stress values for lifting table components

The stresses that appear during the FEM calculation will be assessed with European standard NEN-EN 13445 (unfired pressure vessels) as indicated earlier in this report. For the stresses a safety factor of 1.5 will be taken into account.
The occurring stresses can be subdivided into the following components:

- σ_m: Primary membrane stress = yield strength / safety factor
- σ_l: Local primary membrane stress = 1.5 * yield strength / safety factor
- σ_b: Primary bending stress = 1.5 * yield strength / safety factor
- σ_q: Secondary stress = 3 * yield strength / safety factor

Values for the used materials are given in table 1.

The previously mentioned stresses are defined in the European standard and repeated here:

Primary stresses are defined as stress which satisfy the laws of equilibrium of the applied loads. Regarding the mechanical behavior of a structure, the basic characteristic of a primary stress is, that in case of high increment of external loads, it is not self-limiting. As plasticity develops, a stage is reached where no further beneficial redistribution of stress can take place.
Primary membrane stresses are defined as the average value of the respective stress components distributed over the section governing the load-bearing behavior defined by the supporting line segment (see C. 4.4 of NEN-EN 13445). If a membrane stress is averaged over a localized portion of a cross-section, it is a local membrane stress.
Primary bending stresses are defined as primary stresses distributed linearly across the considered section and proportionally to the distance from the neutral axis.

Secondary stresses are stresses developed by constraints due to geometric discontinuities, by the use of materials of different elastic moduli under external loads, or by constraints due to differential thermal expansions. With respect to the mechanical behavior of the structure, the basic characteristic of a secondary stress is that it is self-limiting, i.e. local flow deformation loads to a limitation of the stress. Secondary stresses lead to plastic deformation when equalizing different local distortions in the case of excess of the yield strength.

3.3.2 Fatigue strength

The number of cycles to be taken into consideration for a generic lifting table are specified in the European norm for lifting table. Previously the number of cycles is determined to be 128000. Calculation of the fatigue strength of 128000 cycles is done according to European standard EN 1993-1-9 §7.1 and Annex B (Design of steel structures).
For constant amplitude nominal stress ranges the fatigue strength can be obtained as follows:

$$\Delta\sigma_R = \sqrt[m]{\frac{2x10^6}{N_R}}\Delta\sigma_c \qquad with\ m = 3\ for\ N \leq 5x10^6, \Delta\sigma_c = Fatigue\ strength\ at\ 2\ million\ cycles$$

The fatigue strength at 2 million cycles is determent using annex B, the geometric hot spot stress method. This method indicates the fatigue resistance of welds. The worst category is chosen, $\Delta\sigma_c$ = 90. With these values a fatigue stress of $\Delta\sigma_R$ =150 [MPa] is obtained. The safety factor of 1.5 is already included. Since the lowest strength of the used material is lower than this fatigue strength no further measures are required.

3.3.3 Welded connections

For welded connections no strength reduction is taken into consideration. A conservative and simplified method for determining the allowable static stresses in the welds following the European standard NEN-EN 1993-1-8 §4.5.3.3 is used. This assessment can be used under the condition that the welded connection has matching cross section with the welded parts.

The allowable stress in a weld is:

$$f_{vw,d} = \frac{f_u/\sqrt{3}}{\beta_w \gamma_{M2} sf} = 0.31\, f_u = 160\ MPa$$

Since the allowable stress for the normal material is lower than this weld stress, taking the weld not into consideration is a conservative method.

4 Gaps

In this chapter the safety gap sizes for feet and hands are checked. The focus is mostly on the two special cases shown in figure 1.
Figure 2 shows the lifting table from two different orientations. The gaps are indicated with numbers and every gap will be discussed individually.

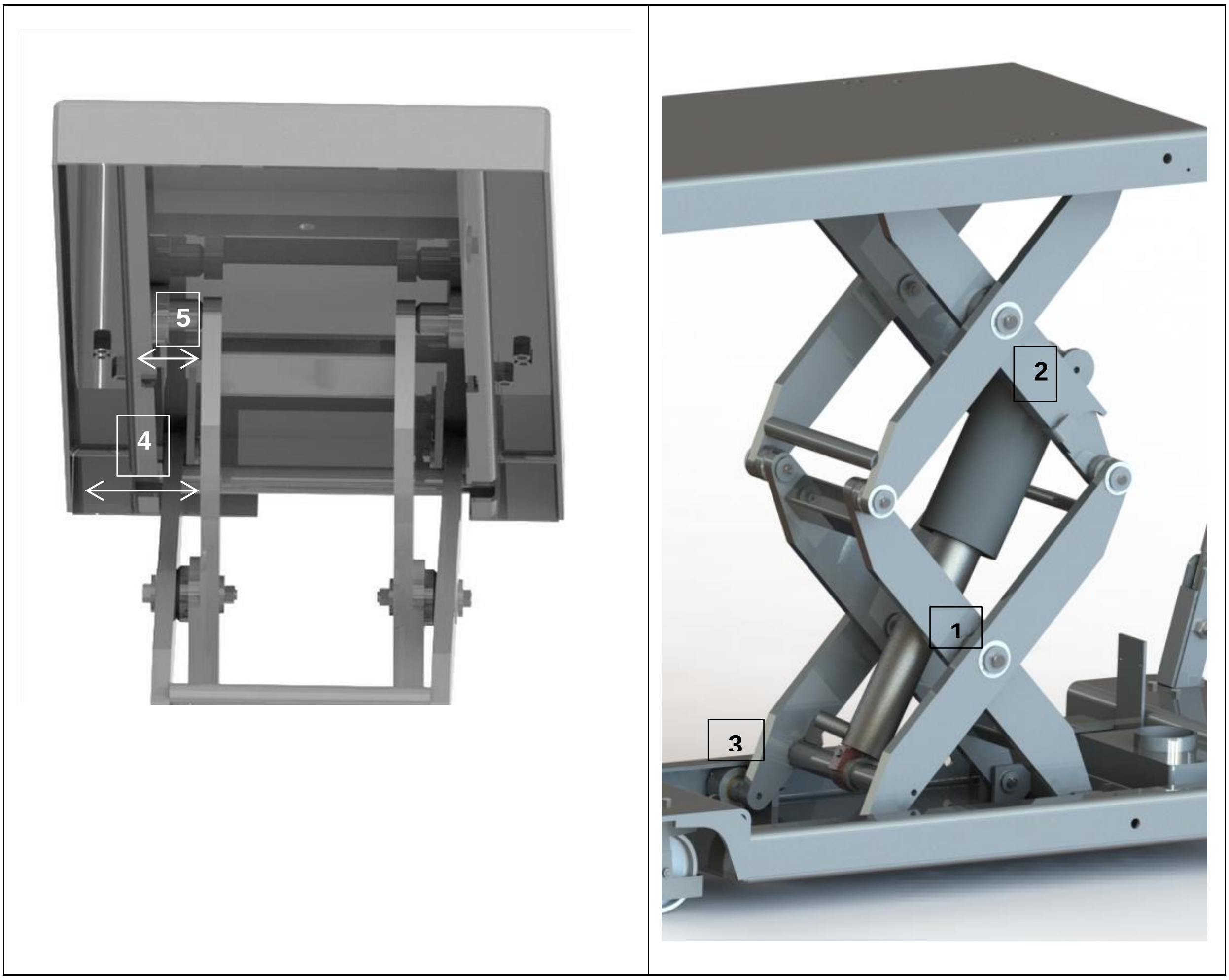


Figure 2: Solid works model with gap indicators

The first case is the gap between the scissors. At this moment this gap is 25 [mm]. Paragraph 3.2.7 says that this gap should be at least 30 [mm]. So this gap does not conform with the European standard.
The second gap is between the maintenance plate and the scissor leg. The maintenance plate is basically just reducing the space between the scissors. Taking into account the plate, the gap is only 13 [mm] between the scissors. It can be concluded that this situation needs a revision because this gap is also less than 30 [mm].
The gap indicated with number three is between the scissor leg and the bottom deck. This gap should be at least 50 [mm]. The gap is 39 [mm] at this moment. So this gap is also not conform the European standard.
The fourth gap is between the upper deck and the scissor leg. This gap should be at least [50] mm. At this moment this gap is 98 [mm]. So, this is the first gap that is conform the European standard. There is also a plate that keeps the cam wheel in its place, this is indicated with number five. Taking into account this plate reduces the gap to 7 [mm], which is far too small.
Further gaps are at this moment not considered. The current situation should first be redesigned with respect to the European standard.

5 Description of the analysis

The performance of the lifting table will be assessed in three parts. The first part is the calculation of the platform deflection, the second part is the stability analysis and finally the occurring stresses will be analyzed. For the first two parts different load cases will be defined using the requirements from the European standard as indicated in section 3.2. For the stresses no particular load case is specified. Therefore, the worst case scenario from the first two parts will be checked on stresses to indicate where the weak spots are.

5.1 Load cases platform deflection

This analysis will be performed as described in section 3.2.2. In Table 2 all possible load cases are shown.

Platform Deflection	*Load case*	*Boundary condition*	*Gravitation [mm/s²]*			*Operator force [N]*			*Force on load COG [N]*		
			X	**Y**	**Z**	**Fx**	**Fy**	**Fz**	**Fx**	**Fy**	**Fz**
	1	1		-13730	-1030			-330			-343.35
	2	2		-13730	1030			330			343.35
	3	3	-1030	-13730		-330			-228.9		
	4	4	1030	-13730		330			228.9		

Table 2: Load cases platform deflection

In the first two cases the load is applied over half of the length of the platform and with half of the working limit load, in this case 250 [kg]. In the last two cases the load is applied over half of the width of the platform with one third of the working limit load, in this case 167 [kg]. The values are obtained combining the different load components discussed in chapter 3. Results will be discussed in section 6.3.1. Load case 1 is shown in figure 3. The blue element reflects the mass point of 250 [kg]. The mass is applied at the platform by the black connecting element. The operator force and force on the center of gravity are indicated with the green arrows.

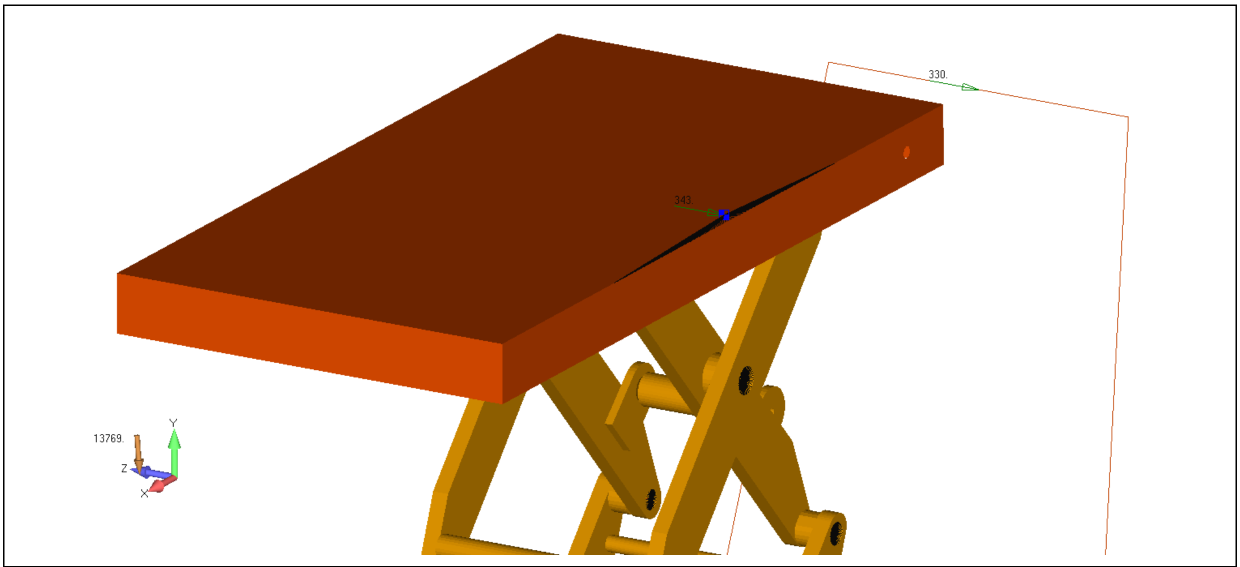


Figure 3: Orientation of loads, Load case 1 deflection

5.2 Load cases stability

The stability analysis will be performed as described in section 3.2.3. In Table 3 all possible load cases are shown.

Stability analysis	*Load case*	*Boundary condition*	*Gravitation [mm/s²]*			*Operator force [N]*			*Force on load COG [N]*		
			X	**Y**	**Z**	**Fx**	**Fy**	**Fz**	**Fx**	**Fy**	**Fz**
	5	5		-9807	-1339			-546			-893
	6	6	1339	-9807		546			893		
	7	7		-9807	-1339			-546			-893
	8	8	-1339	-9807		-546			-893		
	9	9		-9807	1339			546			893
	10	10	-1339	-9807		-546			-893		
	11	11		-9807	1339			546			893
	12	12	1339	-9807		546			893		

Table 3: Load cases stability

The results of the stability analysis will be shown in section 6.3.2. This section also contains schematic pictures to indicate the preferable, worst case, position of the castor wheels as well as the applied constraints on the wheels and the position of the load. The values are obtained by combining the different load components discussed in chapter 3. For the stability analysis stabilizing loads do not have a dynamic load factor or stability factor and overturning loads are taken worst case and include a dynamic load factor and the stability factor of 1.3. Load case 5 is shown in figure 4. The blue element is the mass of 500 [kg]. With the grey element this mass is equally divided over one quarter of the platform. The green arrows again indicated the operator force and the force on the center of gravity of the load.

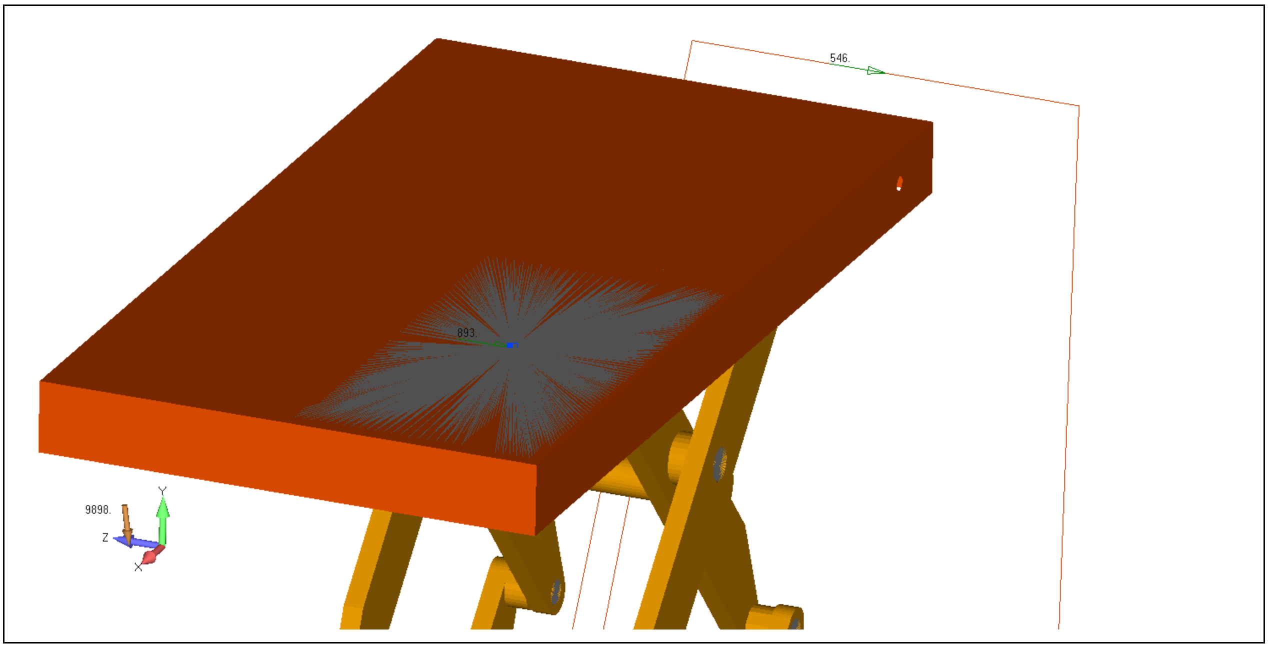


Figure 4: Orientation of loads, Load case 5 stability

6 Finite element method calculations

In this chapter first an overview of the used software is given. Next, the FEM-model is described. Finally the results of the analysis are given and an assessment of the results is done.

6.1 Used software and elements

Type of analysis:
FEM-analyses have been performed using static analysis

Element types:

- **SOLID Elements:**
  The hexahedron element is defined by 8 nodes having three degrees of freedom per node: translations in the nodal x, y, and z directions.
- **PLATE elements:**
  3- and 4-node bending plate elements with 6 degrees of freedom for each node. The elements contain both membrane and bending stiffness.
- **BEAM elements:**
  2-node beam element with 6 degrees of freedom for each node. The beam element is defined by a single directed axis through the first two nodes. The third node defines the plane of the local Y-axis. Element has tension, compression, torsion and bending capabilities. Different properties at each end of the beam can be specified.
- **RBE3 elements:**
  Interpolation elements are used to define the motion at the dependent node as the weighted average of the motions at the independent nodes.

Used software:

Pre- and postprocessor	: FEMAP 11.1.2
Solver	: NX Nastran

6.2 Model description

In this section the used model will be described. The table is modelled in both the high and low configuration. This is done because the different configurations of the table show different load patterns. The cylinder forces and stresses in some parts of the scissors are for example higher in the low configuration. However, the load on the cam wheels are highest in the high configuration. Because of clarity the extended model will be used for this section. The retracted model consists of the same elements. The full model is shown in figure 5. With help of this figure the different components of the lifting table will be described.

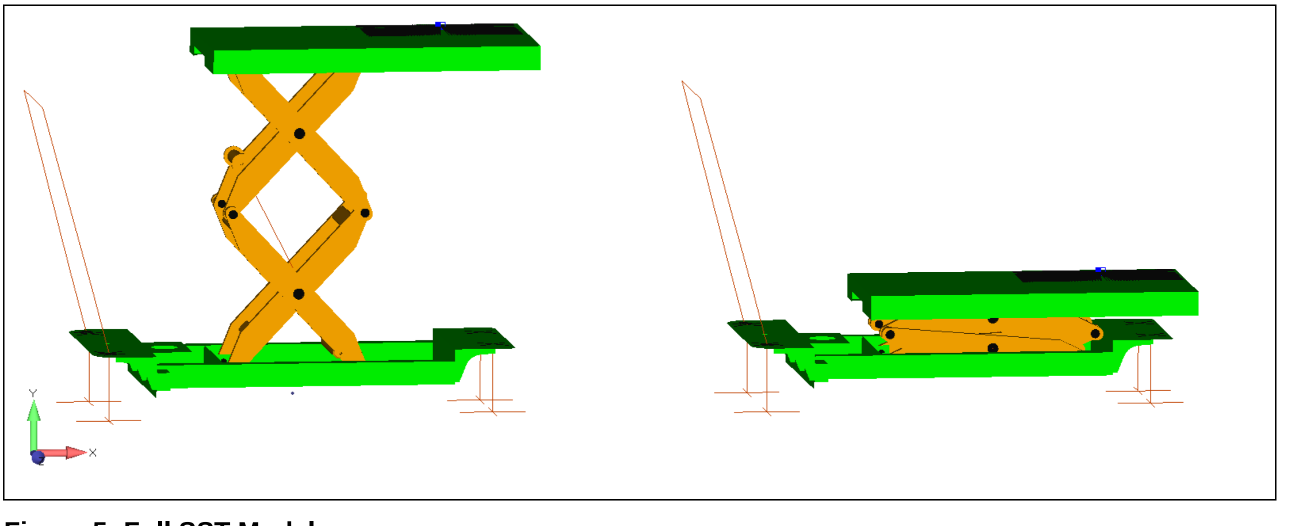


Figure 5: Full SST Model

6.2.1 Solid elements

The scissor legs, cam followers, structural tubes and cylinder axis are modelled with solid elements. These elements are shown in figure 6. The cam followers are connected to the scissors with a glued connection. Doing so the forces between the scissors and the wheel axis can be obtained.

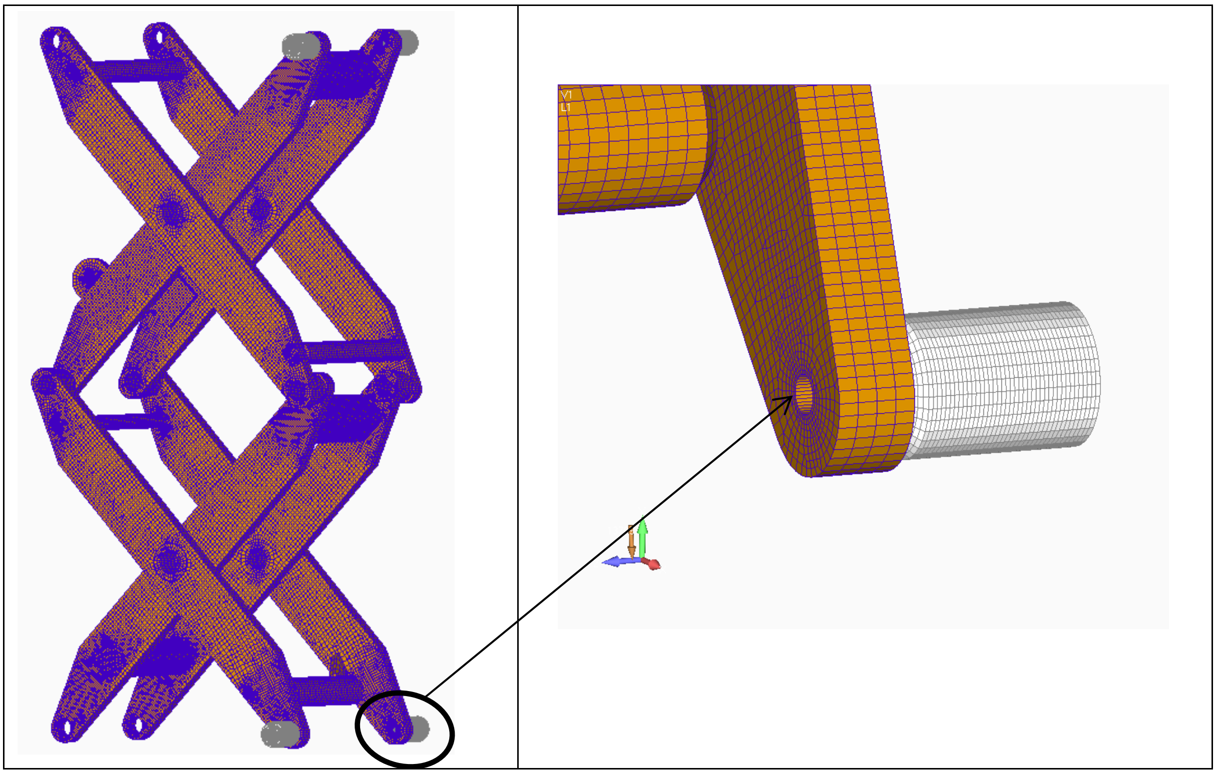


Figure 6: Solid elements with detailed view of glue *connection*

6.2.2 Plate elements

Because of a assumed plane stress state, plate elements are used to model the upper and lower deck. For simplicity and accuracy reasons most of the fillets are removed. Figure 7 shows the mesh of both decks.

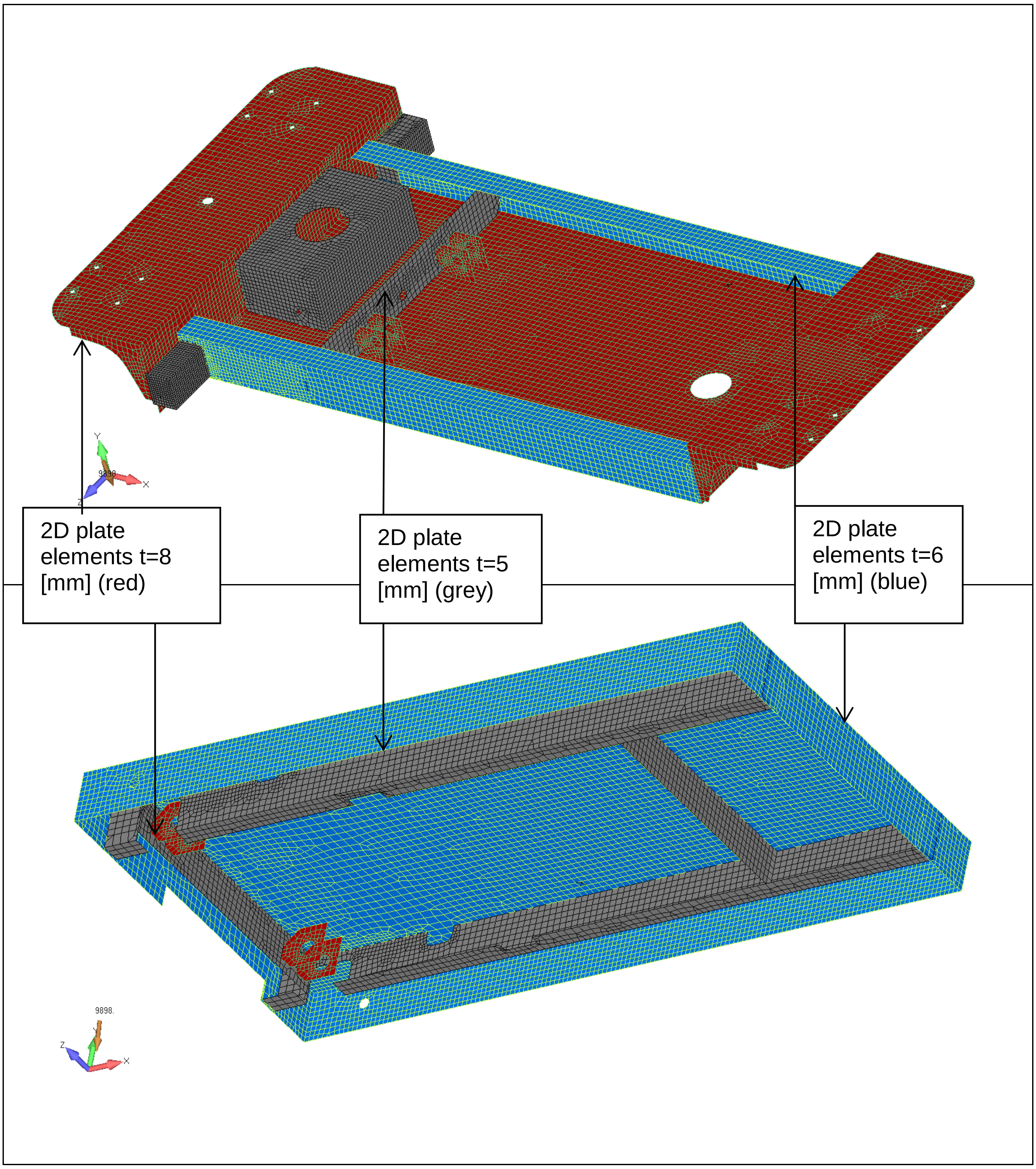


Figure 7: Plate elements of upper and lower deck

6.2.3 Beam and RBE3 elements

To keep the model simple, some components are not meshed in detail, but are modelled using beam elements. These beam elements have realistic stiffness and are connected to the solid and plate elements with RBE 3 interpolation elements. The beam elements are displayed with a red color and the RBE 3 elements with a grey color in figure 8. The castor wheels, operator, cylinder, axis between scissors and connection of the cam followers with the deck are modelled with beam elements. To give a good indication of the forces, stresses and contact pressures at the hinge mechanism the axis should be meshed with solid elements and surface to surface contact should be implemented into the model. This could be a good next step in the engineering. However, the computational demands are much bigger and therefore contact surfaces are not implemented in this design.

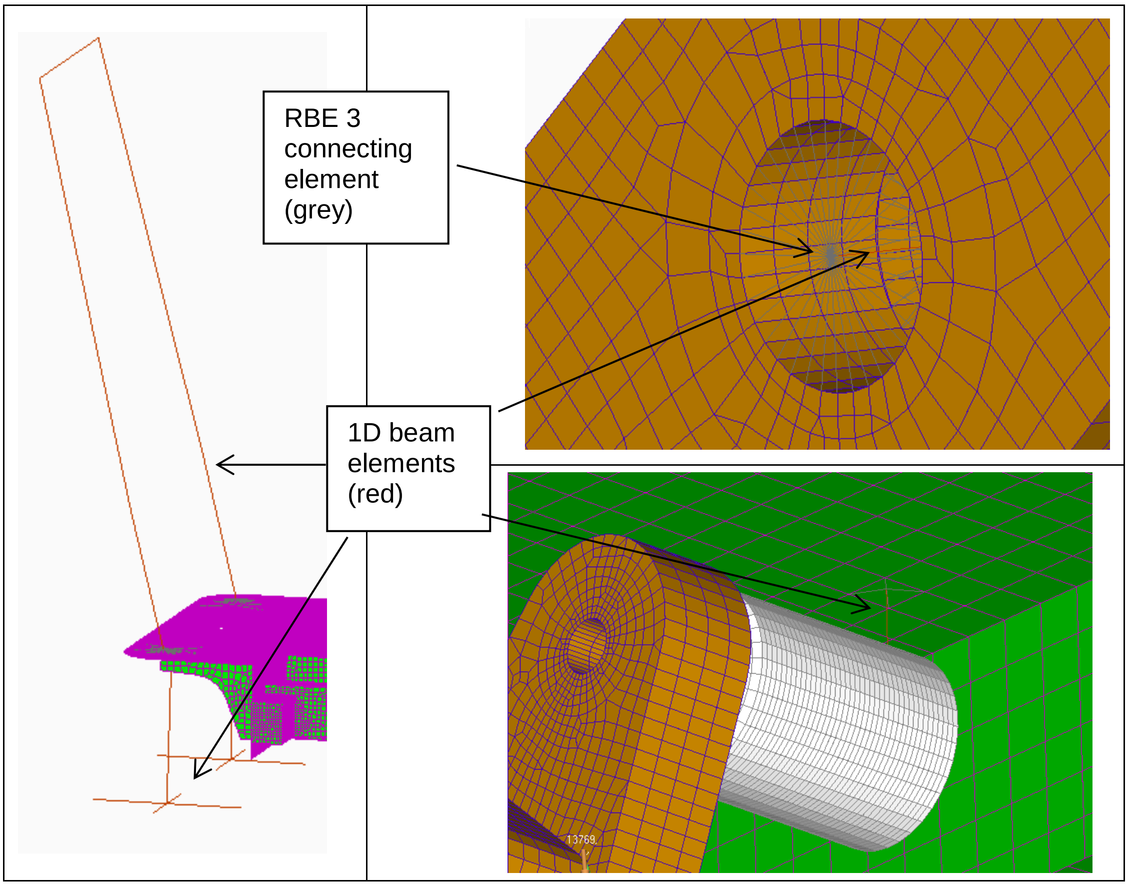


Figure 8: Beam elements for wheels, operator, axis and cam follower

6.2.4 Mass element

To apply a certain weight on the table a mass element is used which is connected to the deck with a RBE3 interpolation element. For one platform deflection case this configuration is shown in figure 9.

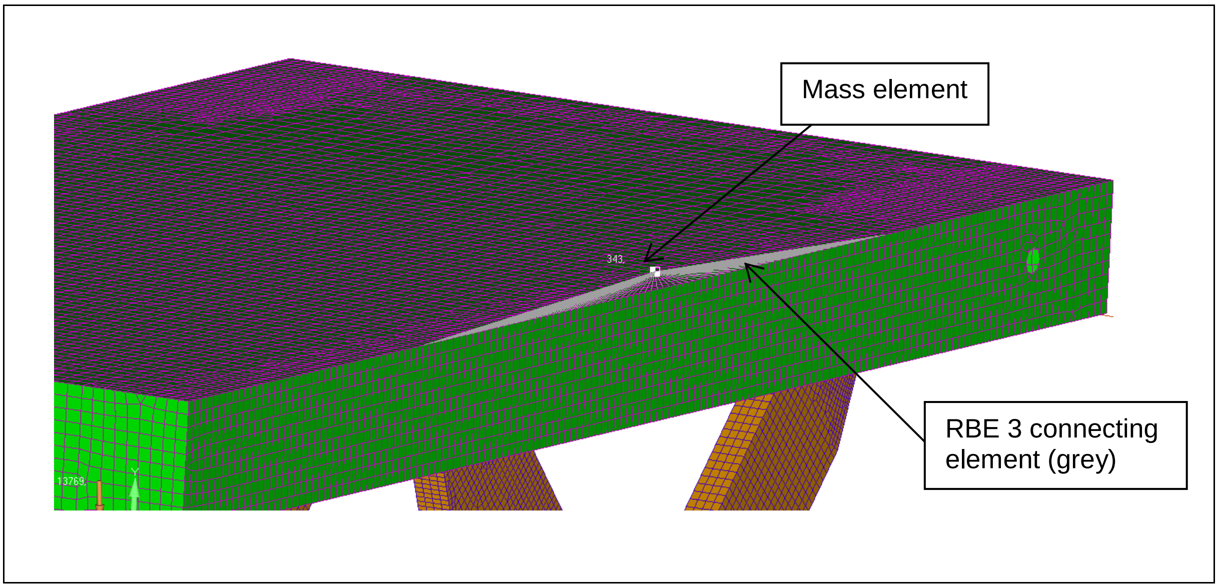


Figure 9: Mass element with RBE3 connection

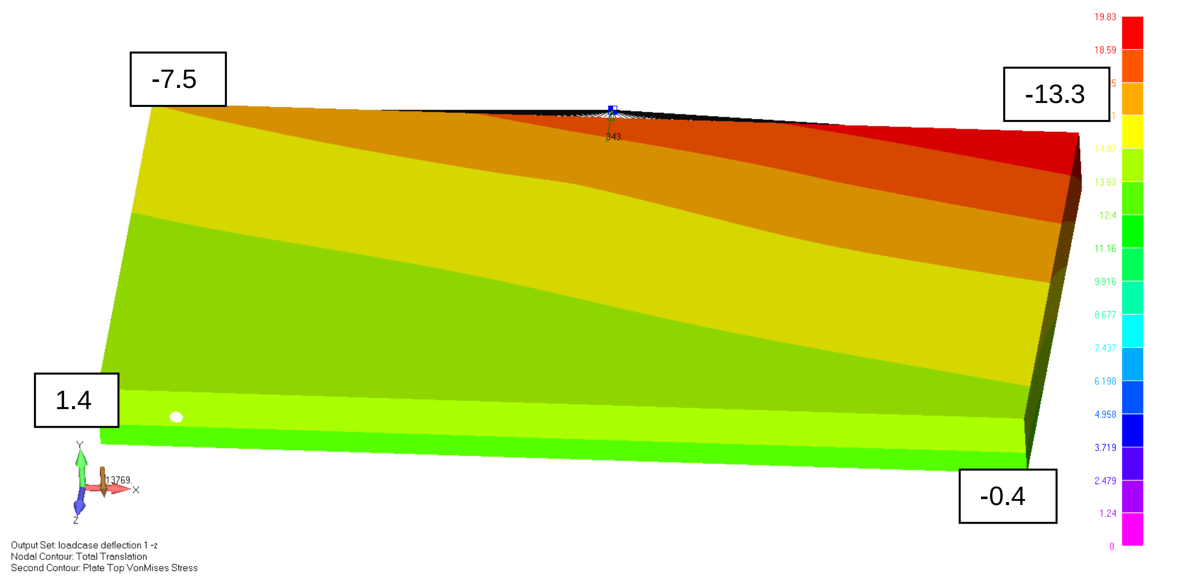
6.3 Results and assessment

In this section the results of the previously described analysis are given.

6.3.1 Platform deflection

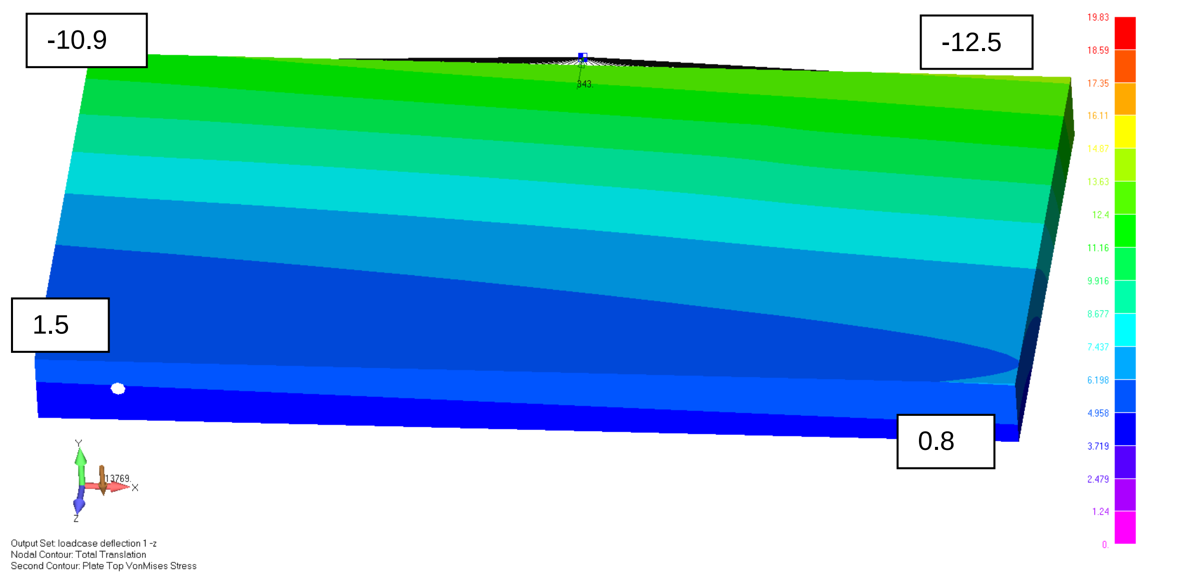
First, the maximum allowable platform deflection is checked. Total deflections from the high and low model are plotted and deflection in y-direction is indicated, see the figures below. The slope is calculated, green is within the specification and red is too much deflection.

Load case 1: Mass on -z side. High



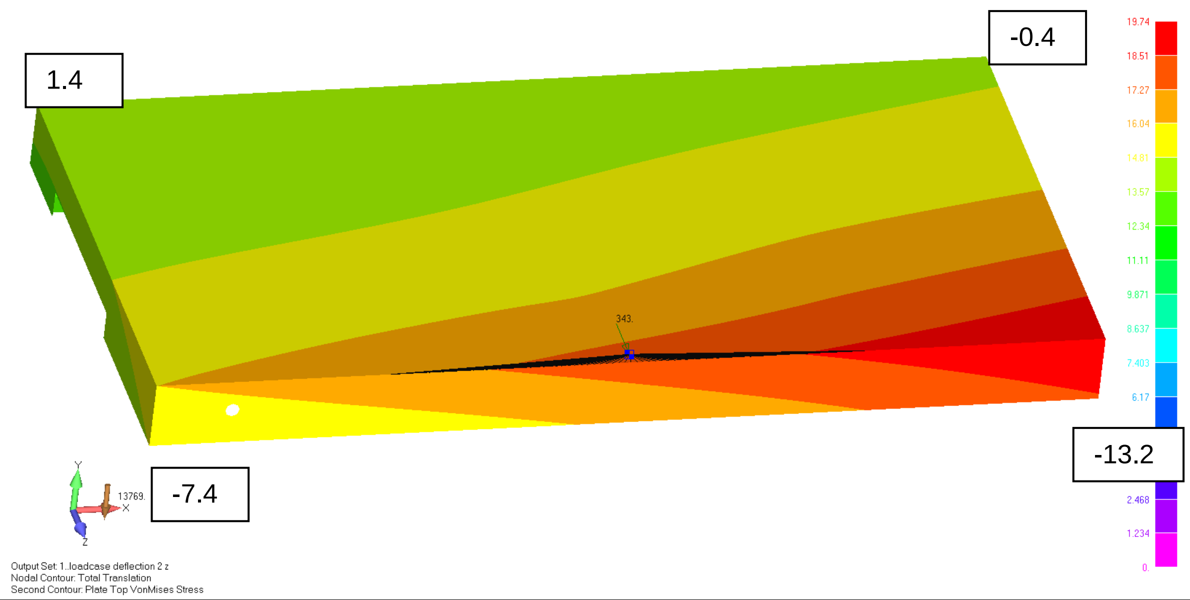
Max. deflection over length: 5.8 / 1000 = 0.58 %
Max. deflection over width: 12.9 / 520 = 2.48 %

Load case 1: Mass on -z side. Low



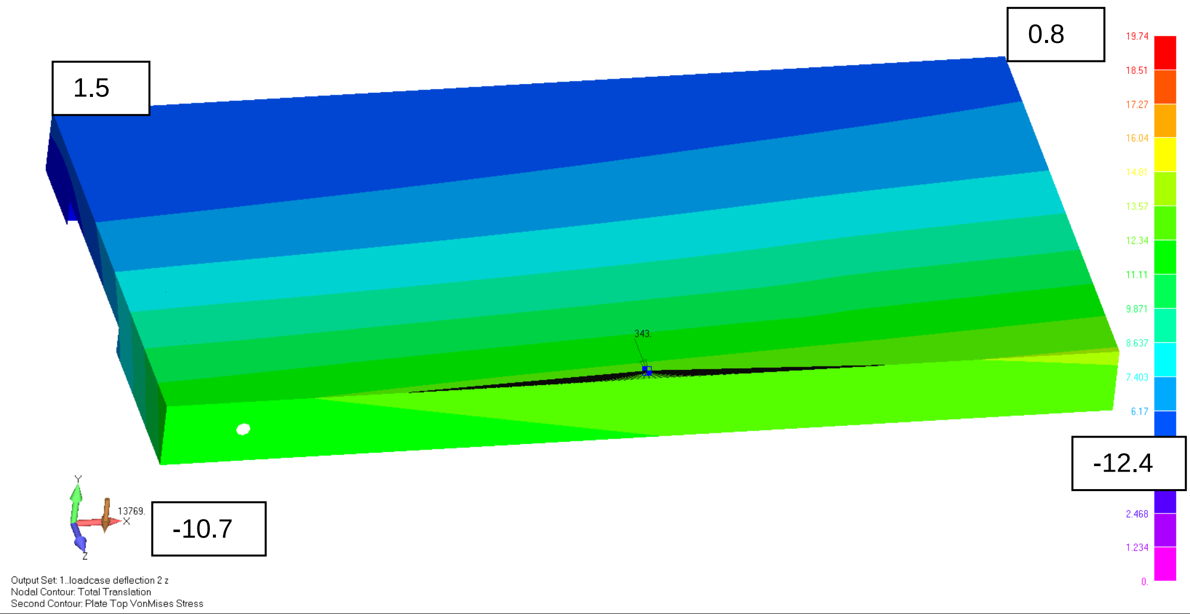
Max. deflection over length: 1.6 / 1000 = 0.16 %
Max. deflection over width: 13.3 / 520 = 2.56 %

Load case 2: Mass on +z side. High



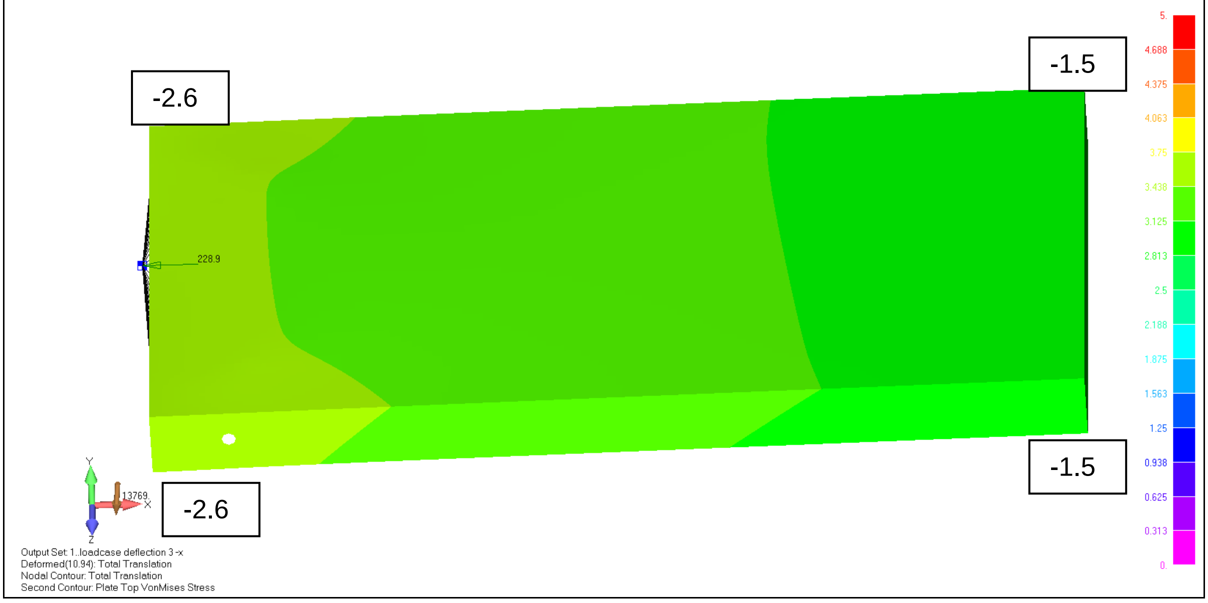
Max. deflection over length: 5.8 / 1000 = 0.58 %
Max. deflection over width: 12.8 / 520 = 2.46 %

Load case 2: Mass on +z side. Low



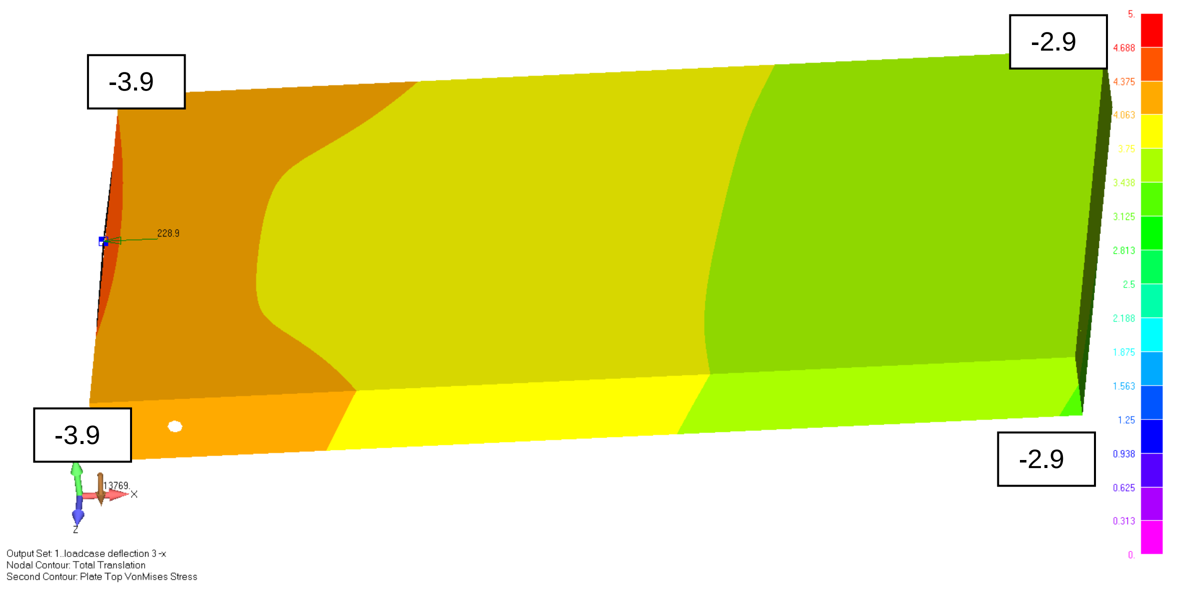
Max. deflection over length: 1.7 / 1000 = 0.17 %
Max. deflection over width: 13.2 / 520 = 2.54 %

Load case 3: Mass on -x side. High



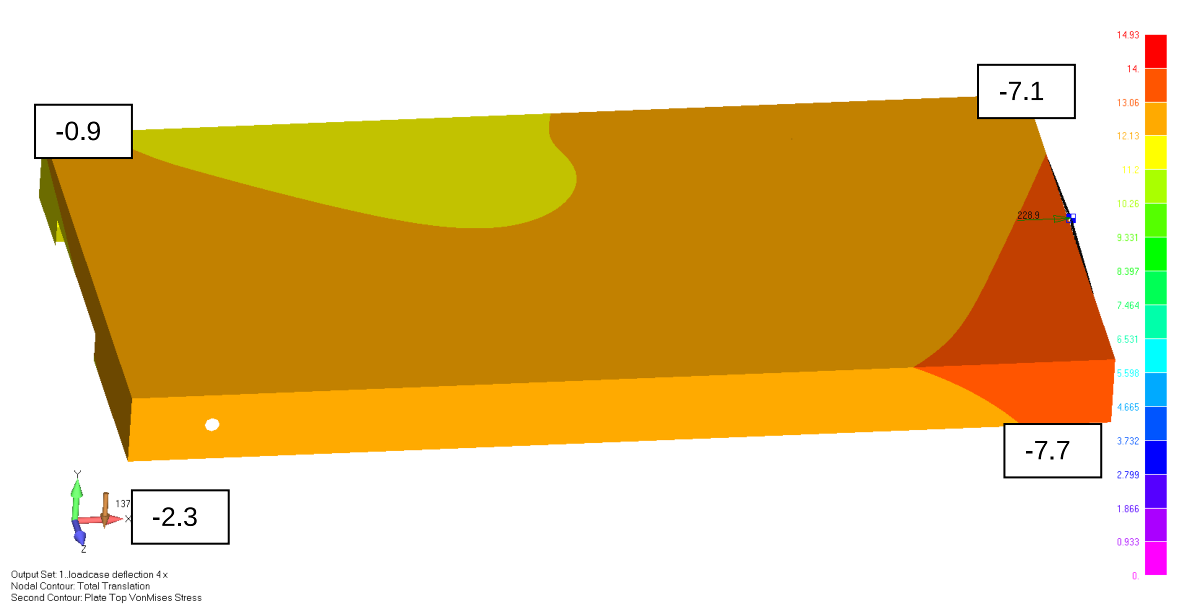
Max. deflection over length: 1.1 / 1000 = 0.11 %
Max. deflection over width: 0 %

Load case 3: Mass on -x side. High



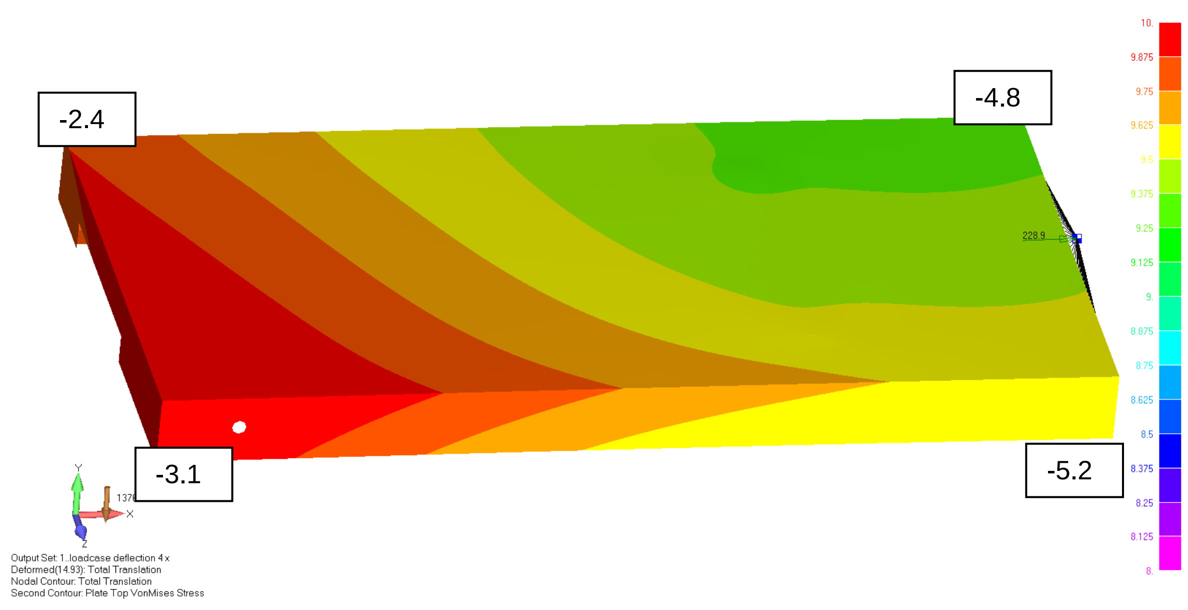
Max. deflection over length: 0 %
Max. deflection over width: 1 / 1000 = 0.1 %

Load case 4: Mass on +x side. High



Max. deflection over length: 6.2 / 1000 = 0.62 %
Max. deflection over width: 1.4 / 520 = 0.27 %

Load case 4: Mass on +x side. Low

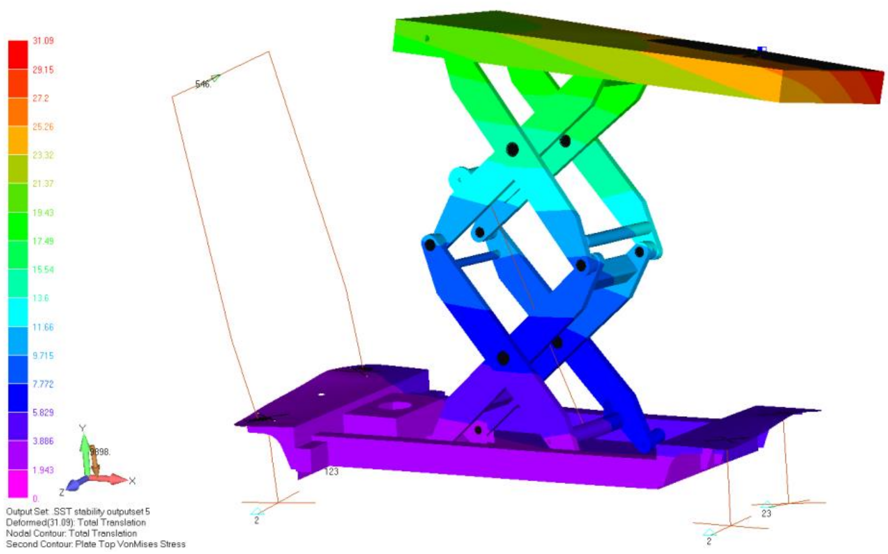


Max. deflection over length: 2.4 / 1000 = 0.24 %
Max. deflection over width: 0.7 / 520 = 0.13 %

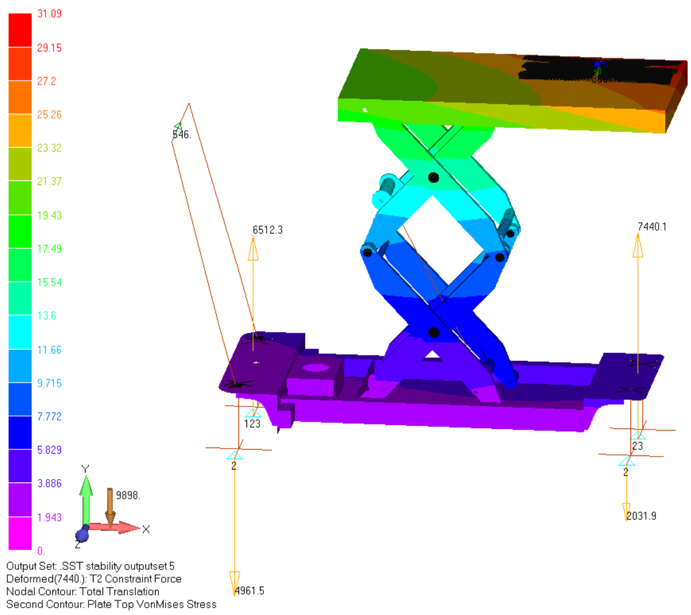
6.3.2 Stability analysis

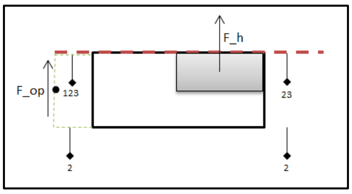
The stability analyses are performed in this section. The figure at the right bottom of each case indicates the configuration. The figure gives a schematic top view of the table. The red dotted line is the expected axis of rotation. The grey block gives the quarter in which the mass is placed. The forces F_h and F_op are respectively the horizontal force on the mass element and the horizontal operator force. Finally the four black beams indicate the orientation of the wheels and corresponding constraints. The number 1, 2 and 3 correspond to constraints on the translation in x, y and z- direction. For the assessment of stability the assumption is made that two wheels on tension always leads to instability. In the case that one wheel is on tension and one compression the moment equilibrium about the axis of rotation is calculated to investigate the table is not turning over.

LOAD CASE 5 - stability



Displacement

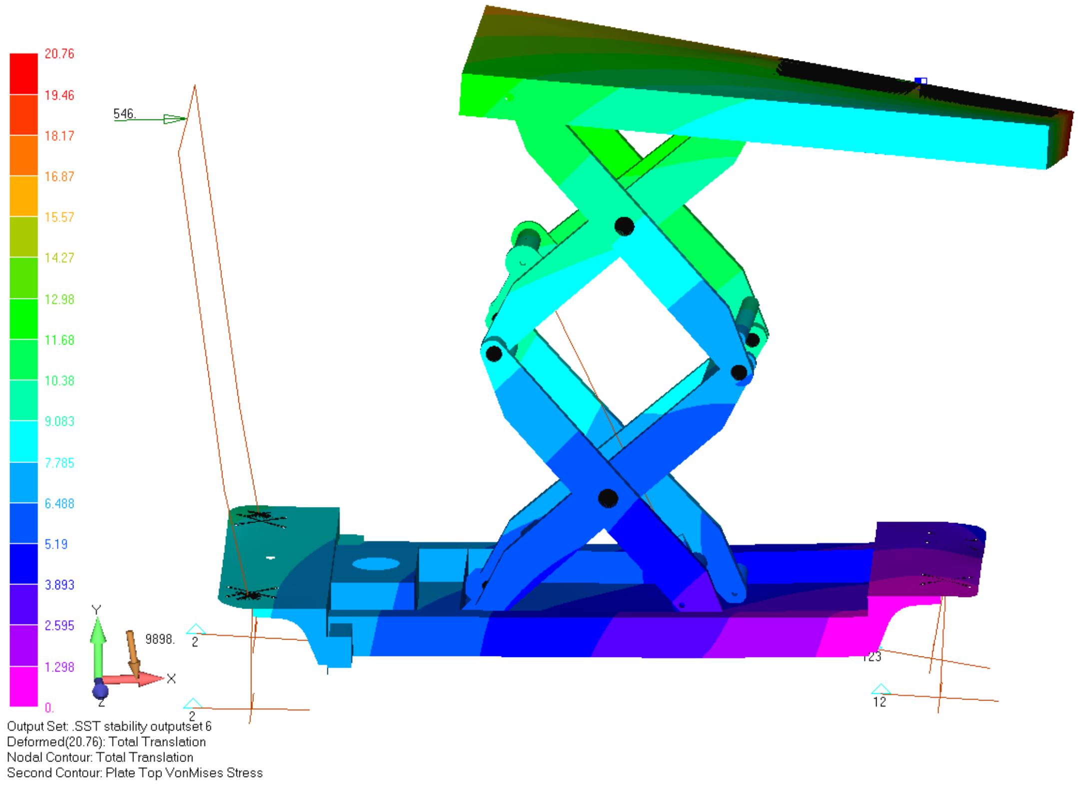




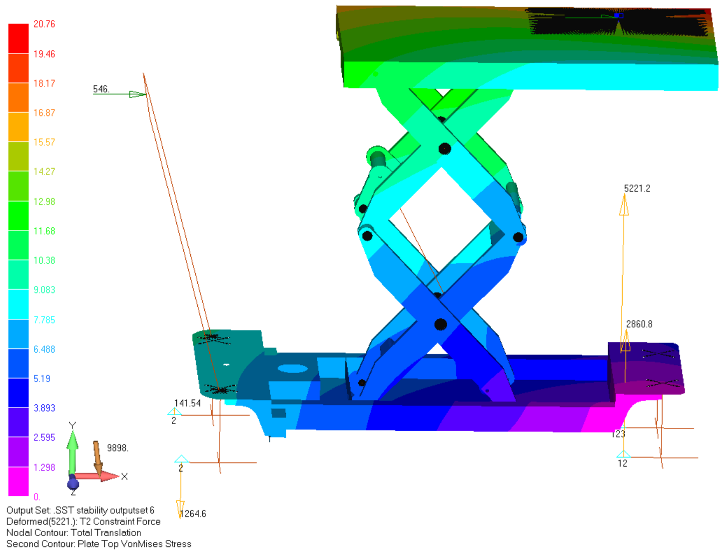
Y – Direction constraint forces

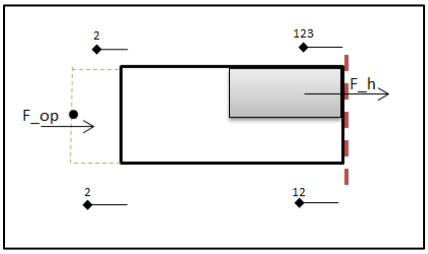
Case 5 is not stable

LOAD CASE 6 - stability



Displacement

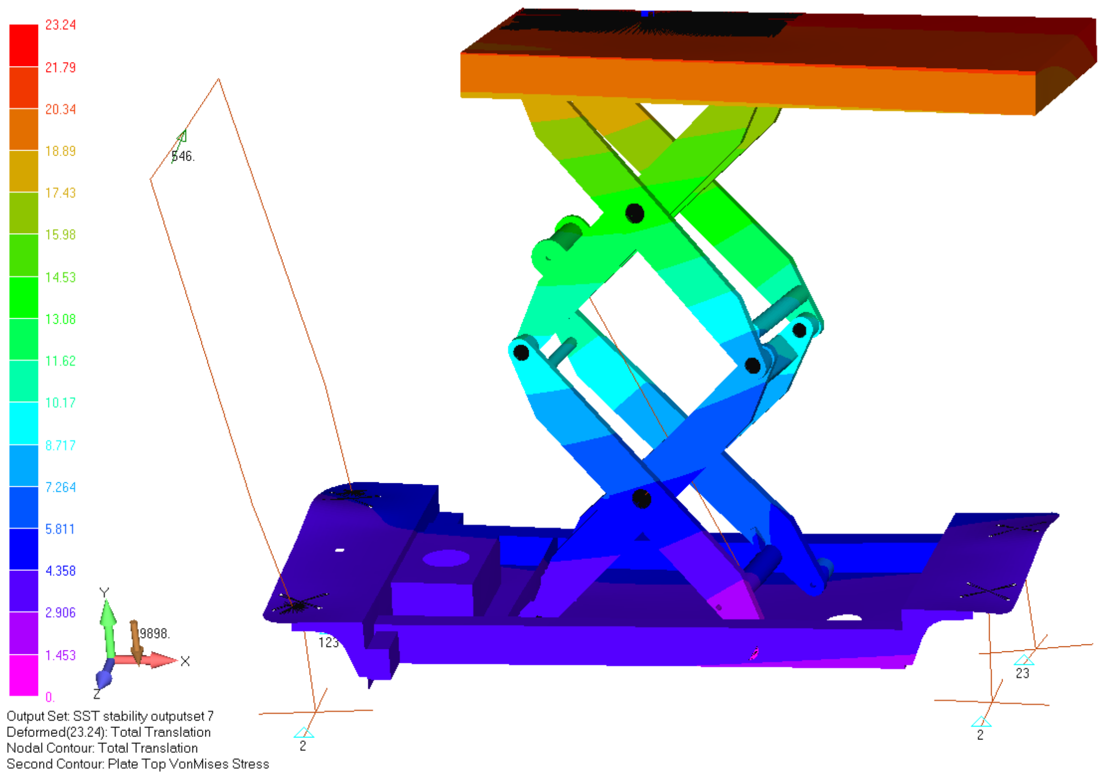




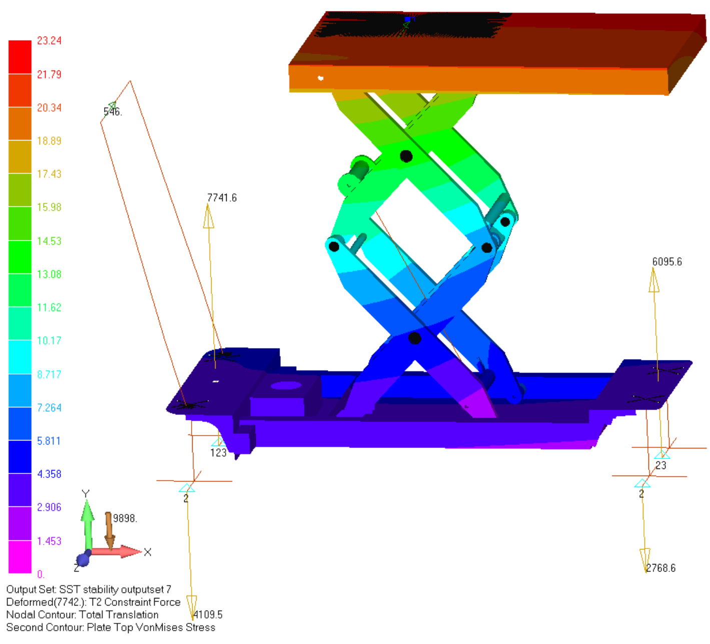
Y – Direction constraint forces

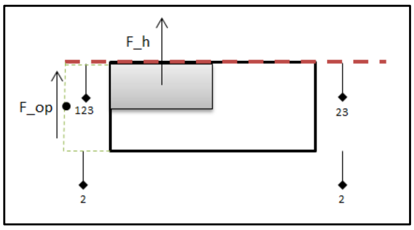
Case 6 is not stable

LOAD CASE 7 - stability



Displacement

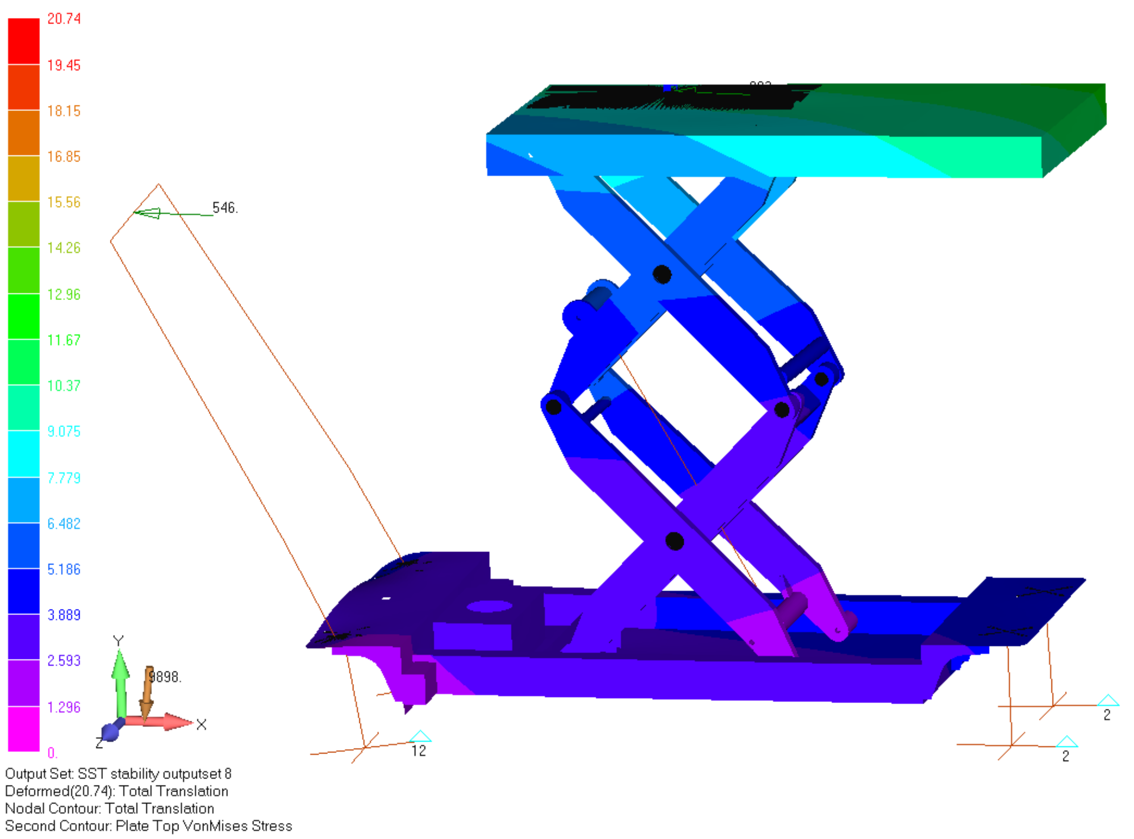




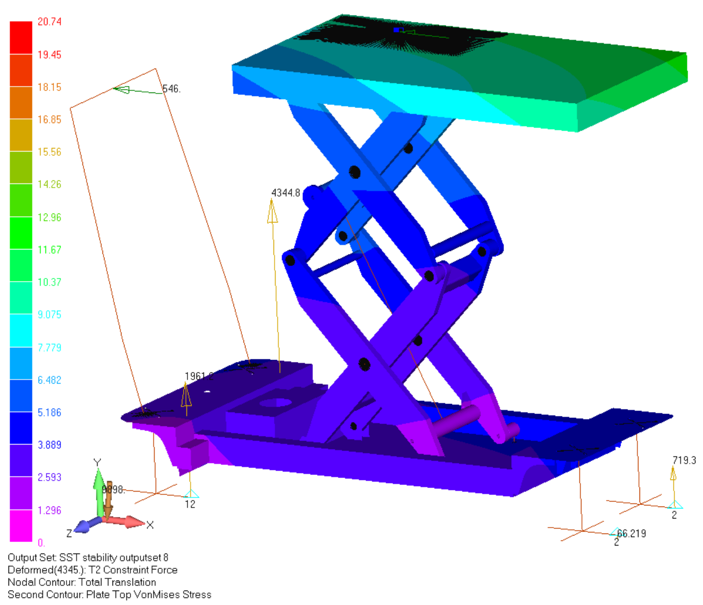
Y – Direction constraint forces

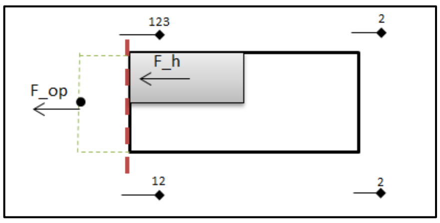
Case 7 is not stable

LOAD CASE 8 - stability



Displacement

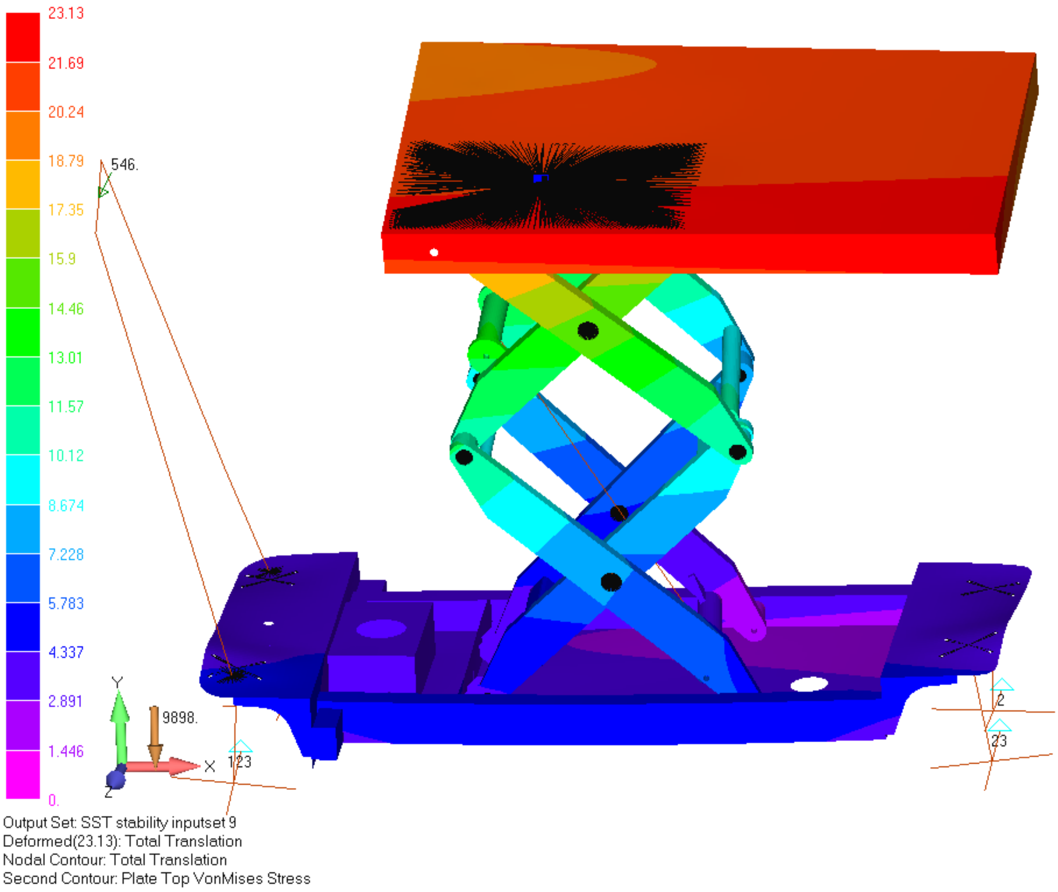




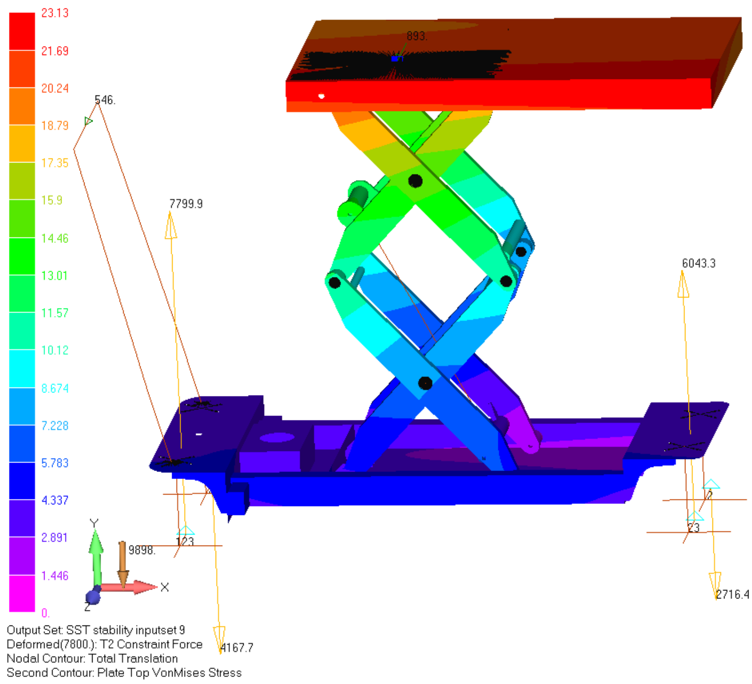
Y – Direction constraint forces

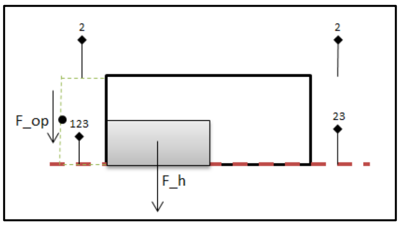
Case 8 is stable

LOAD CASE 9 - stability



Displacement

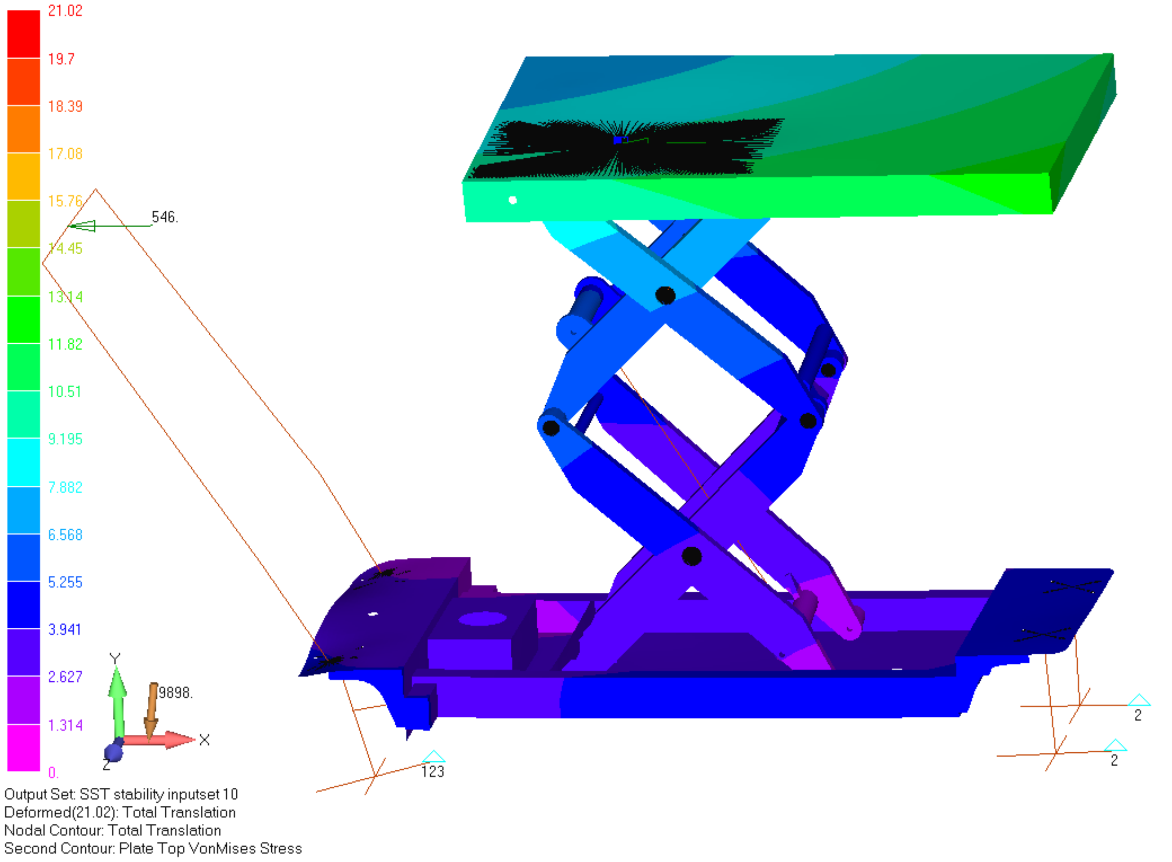




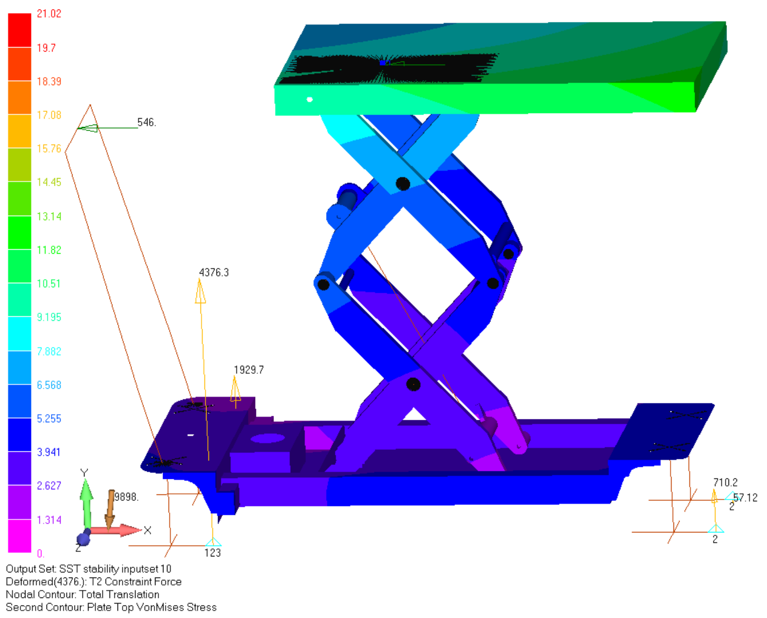
Y – Direction constraint forces

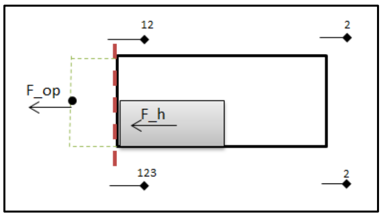
Case 9 is not stable

LOAD CASE 10 - stability



Displacement

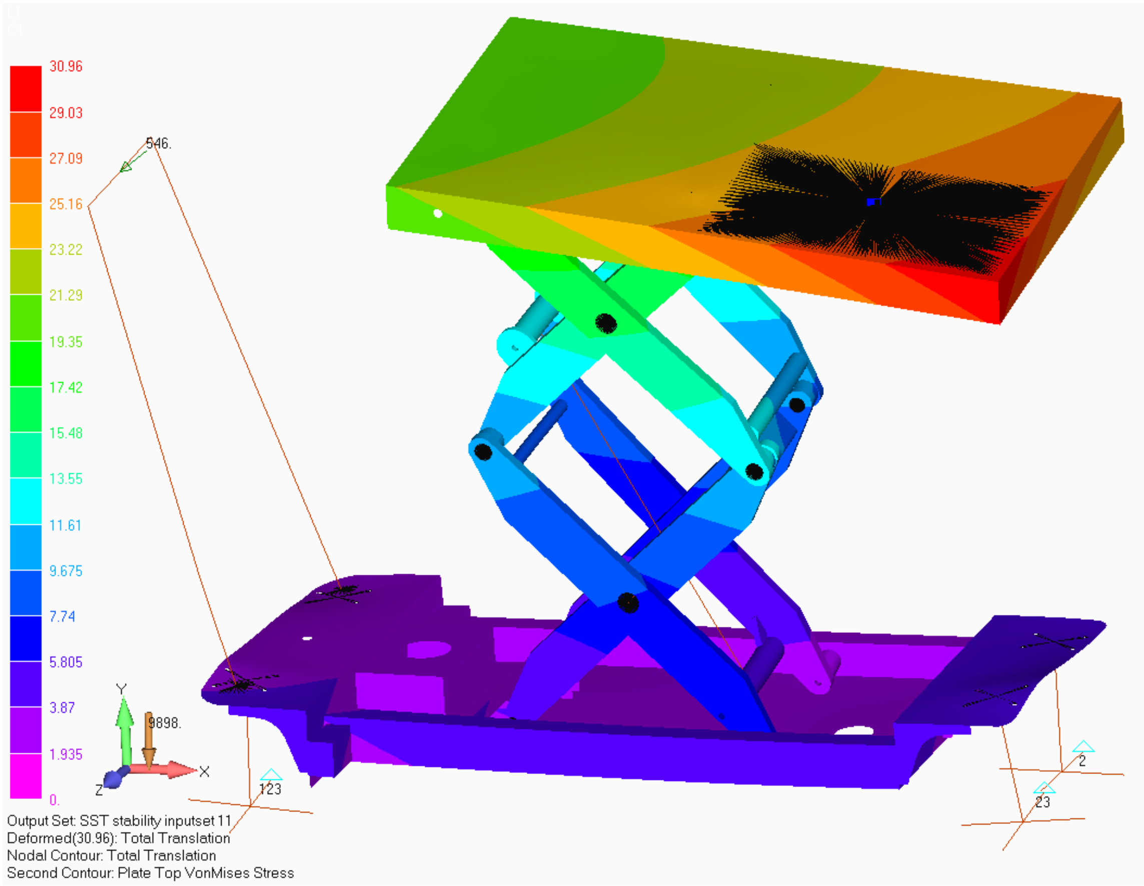




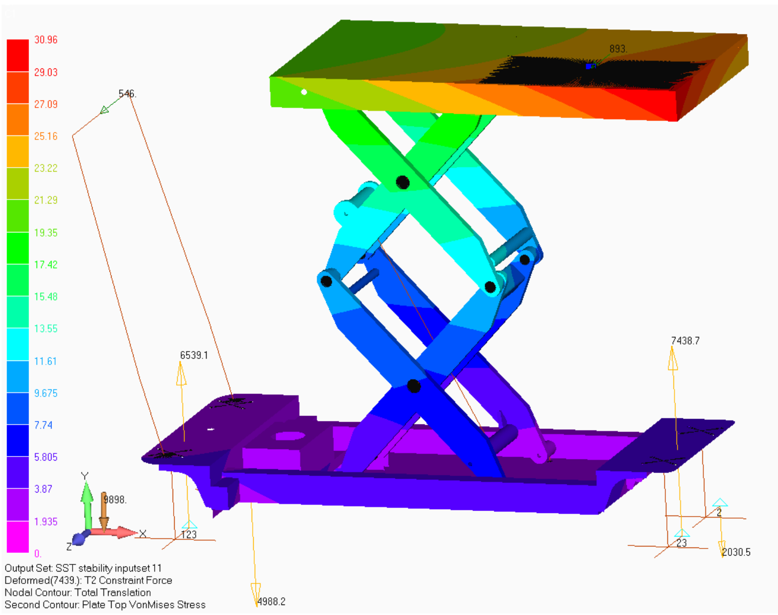
Y – Direction constraint forces

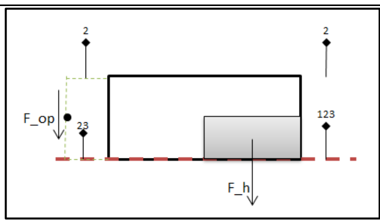
Case 10 is stable

LOAD CASE 11 - stability



Displacement

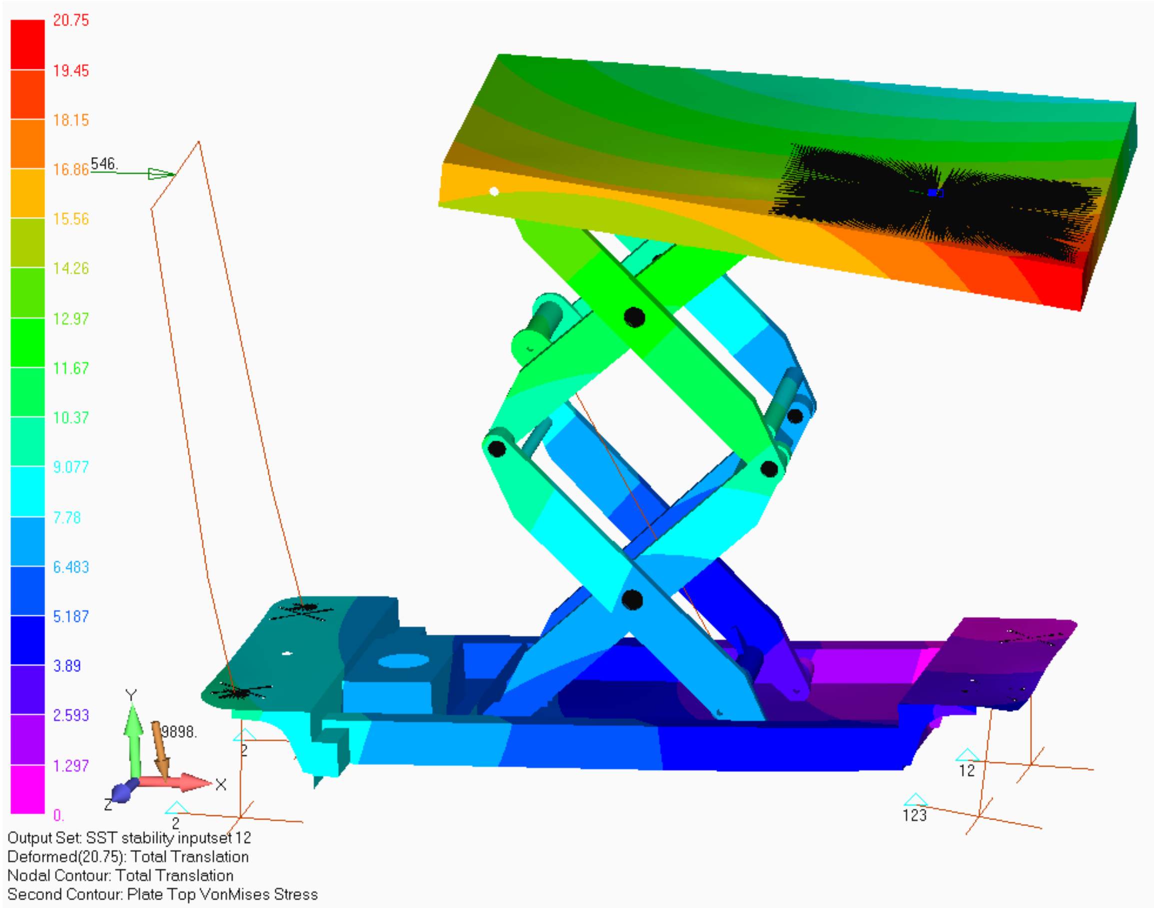




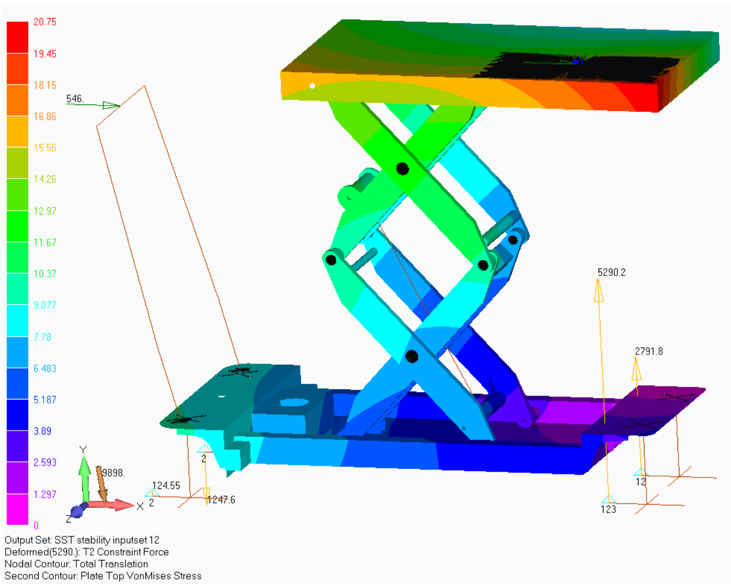
Y – Direction constraint forces

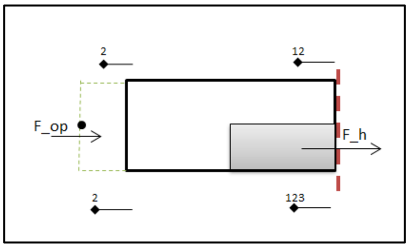
Case 11 is not stable

LOAD CASE 12 – stability



Displacement





Y – Direction constraint forces

Case 10 is not stable

6.3.3 Stress analysis

The stresses are analyzed for one case to give an indication whether the structure is strong enough or not. Stability load case 5 is considered to be worst case, so this case is considered in the stress analysis.
The global stresses in the bottom and top deck are shown in figure 10. It can be seen that the stresses are at some places above the primary allowable stress of 140 [MPa] for stainless steel. These stress peaks cannot be assessed as a secondary stress, because the stress peaks are not self-limiting and therefore, will not disappear in practice. However, the use of 1d RBE 3 elements to connect the wheels to the bottom deck can be a bit too conservative. In practice a plate is connected between the wheels and the deck to spread the stresses. Despite this the performance of the top and bottom deck is not sufficient.

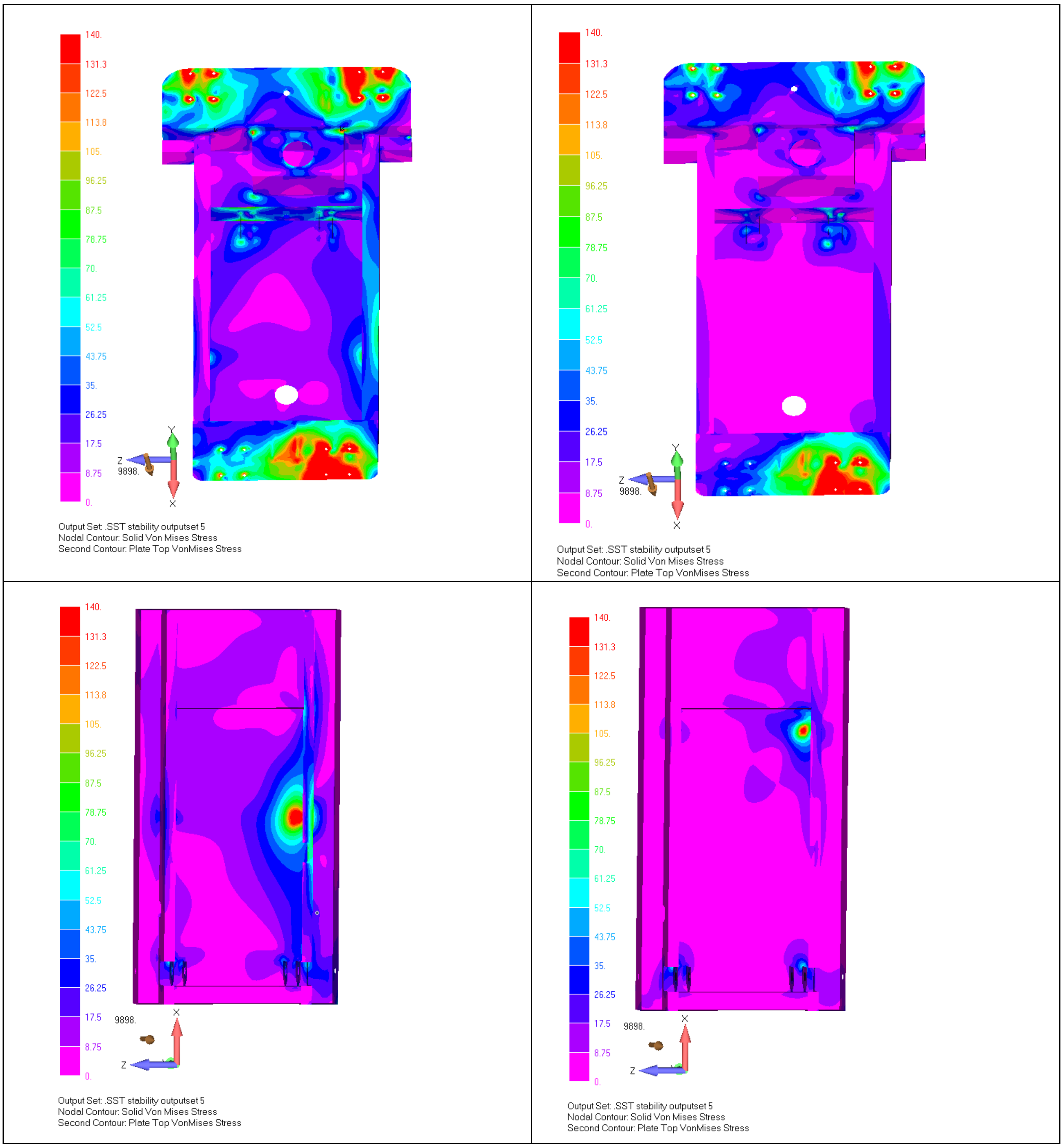


Figure 10: Bottom and top deck of high (left) and low (right) table

The global stresses in the scissors in the high and low configuration are below the allowable primary stress of 140 [Mpa]. See figure 11 and 12.

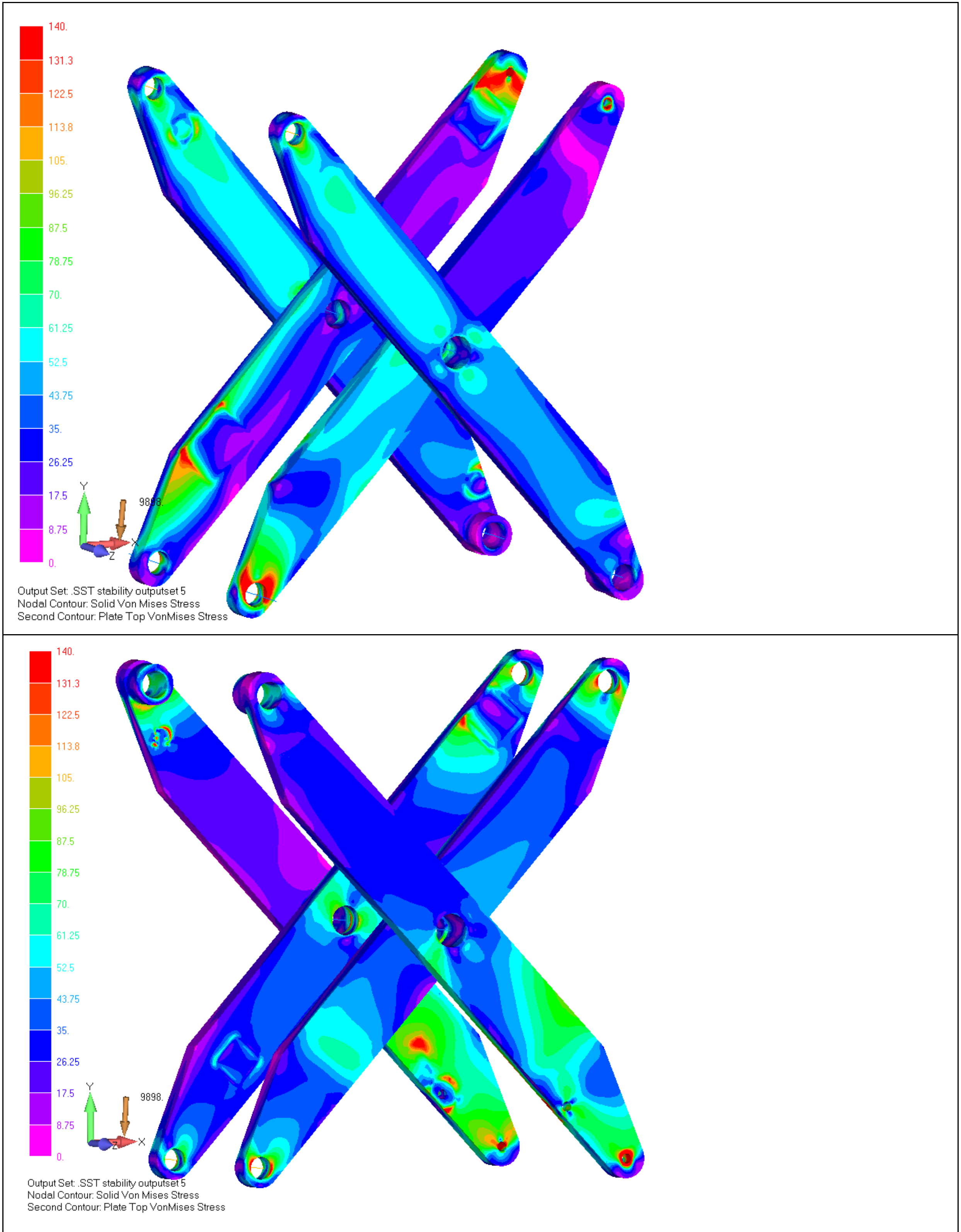


Figure 11: Von mises stress; scissors in high position, scaled to primary stress of 140 [Mpa]

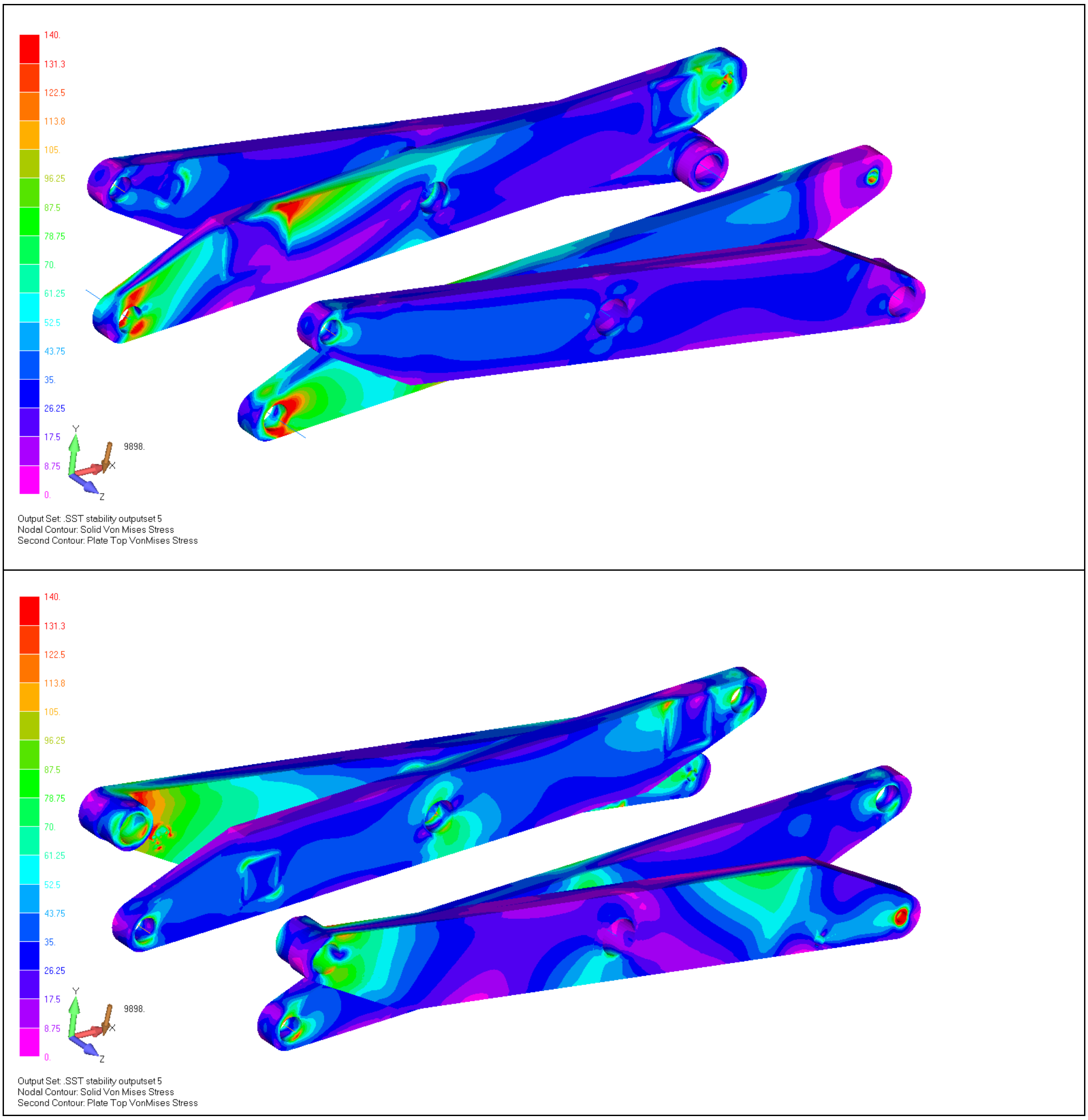


Figure 12: Von mises stress; scissors in low position, scaled to primary stress of 140 [Mpa]

Local exceeding of the allowable primary stress occurs in the regions where the scissors are connected with each other or where structural and cylinder tubes are connected. Most of these stress peaks can be considered as self-limiting and therefore this peaks will disappear in practice due to local non-detrimental deformation. To give more insight the connections should be implemented into the FEM model and surface to surface contact should be applied. Doing so the more detailed peaks can be assessed and the stresses can be checked for secondary values. In figure 13 the high configuration of the scissors are shown scaled to the secondary stress. It can be seen that all peaks disappear except the stresses at the connections of the cam wheels. These connections are considered next.

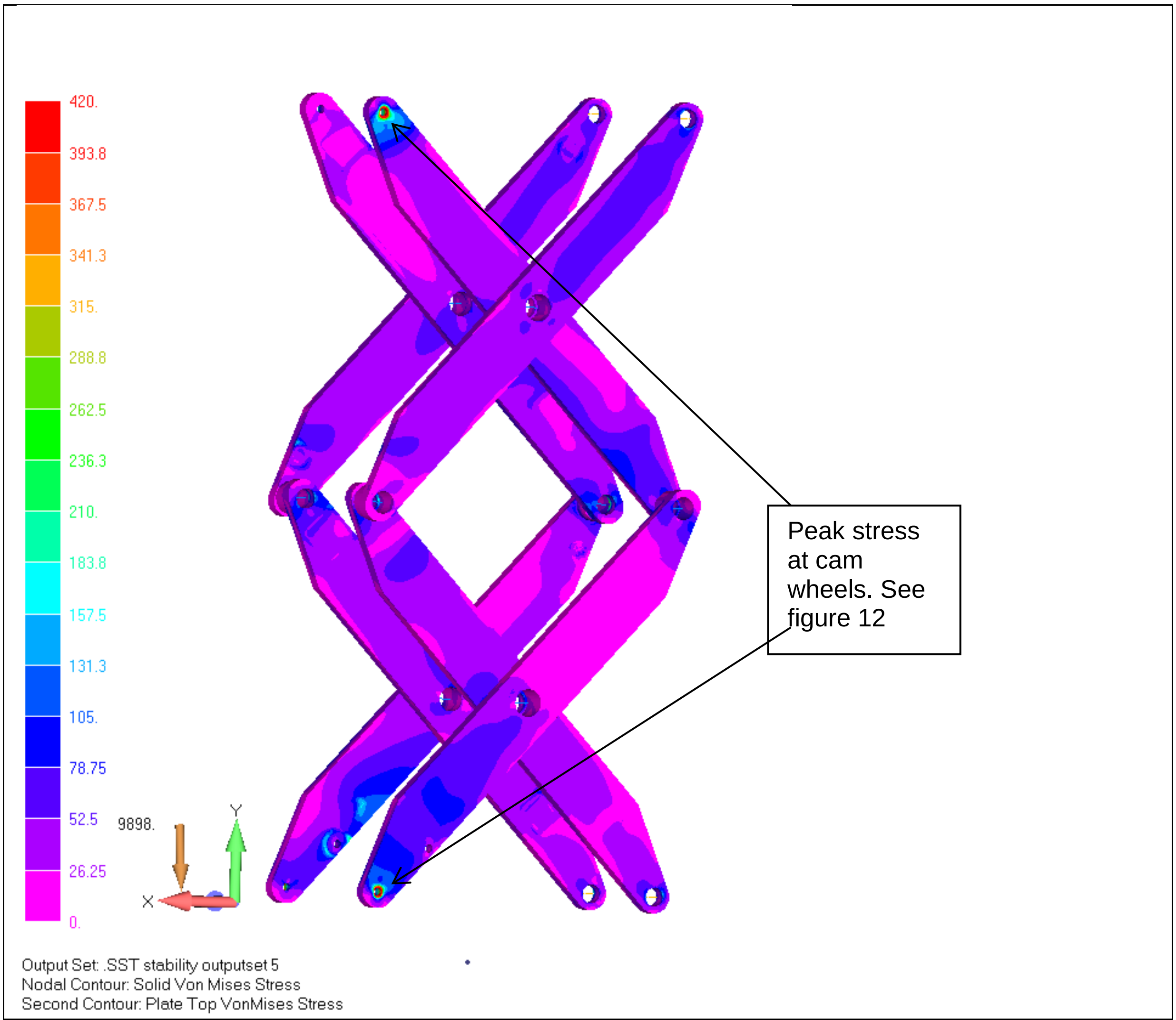


Figure 13: Von mises stress of scissor in high configuration, scaled to secondary stress of 420 [MPa].

In figure 14 the connection of the cam wheel on the scissor leg is shown in bigger detail. It can be seen that the stress exceeds the secondary stress for stainless steel. This connection needs to be revised, for example by making the radius of the axis bigger or connect the wheels on both side of the table with each other in order to make the bending stiffness higher, and increase the bearing surface.

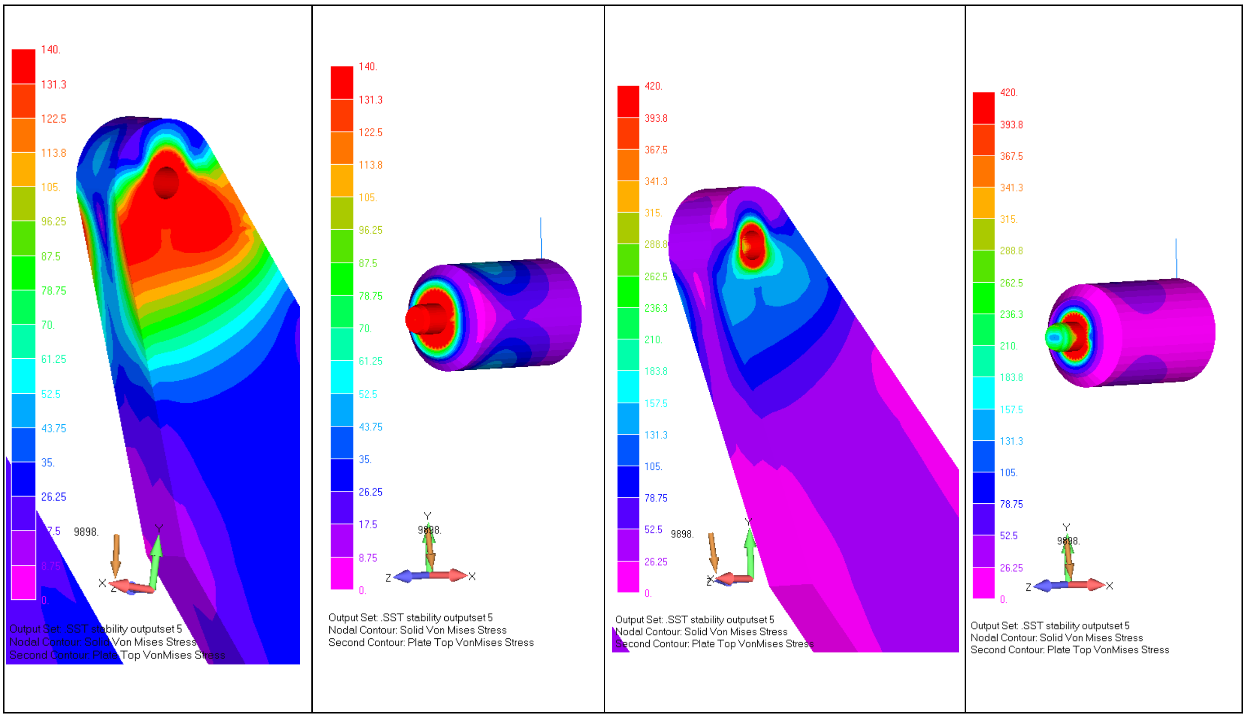


Figure 14: Connection of cam wheel and scissor leg, scaled to primary (left) and secondary (right) stress.

The global stresses in the shafts between the scissors are below the allowable primary stress of 140 [MPa]. See figure 15. Some areas show local exceeding of the primary stress. These peaks can be considered as self-limiting and therefore, be assessed as secondary stresses.

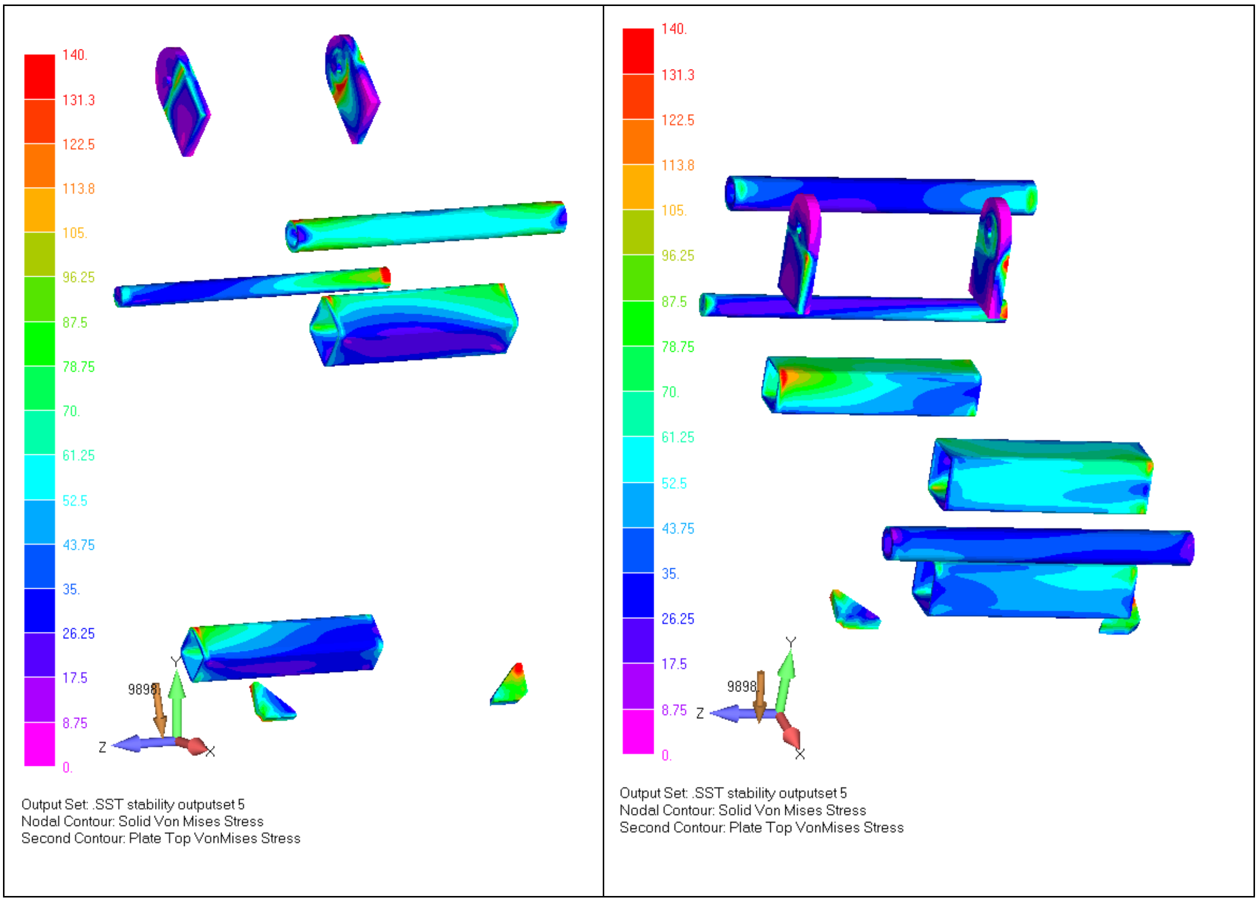


Figure 15: Von mises stress of shafts, high position (left), low position (right); scaled to allowable primary stress of 140 [MPa].

The stresses in the cylinder shafts (Cr42Mo4) are below the allowable primary stress of 433.3 [MPa], see figure 16.

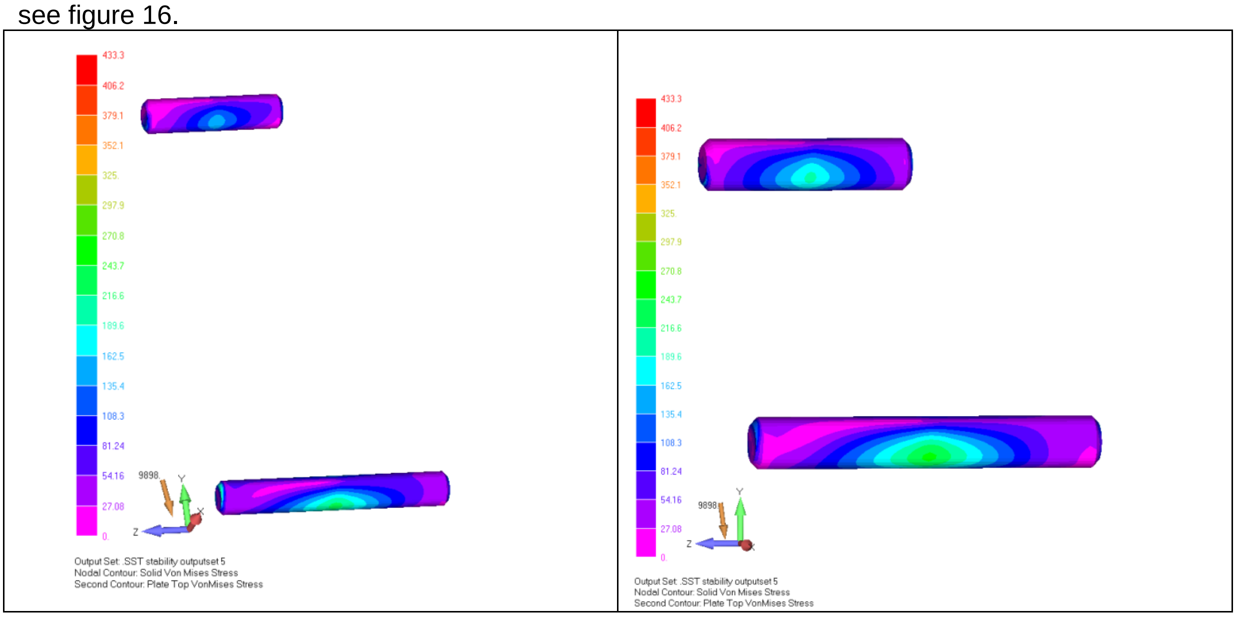


Figure 16: Von mises stress cylinder axis, high position (left) and low position (right), scaled to primary stress of 433.3 [MPa].

7 Analytical verification

In this chapter the lifting table is simplified and different forces are calculated analytically in order to verify the FEM model. For simplicity the table is considered in 2D, as shown in figure 17. A force of 4905 [N] (500 [kg]) is placed at one quarter of the length of the table and in the middle of the width of the table. This means that both scissors take the same amount of force and therefore, the analytical model and the FEM model can be compared in a 2D situation.

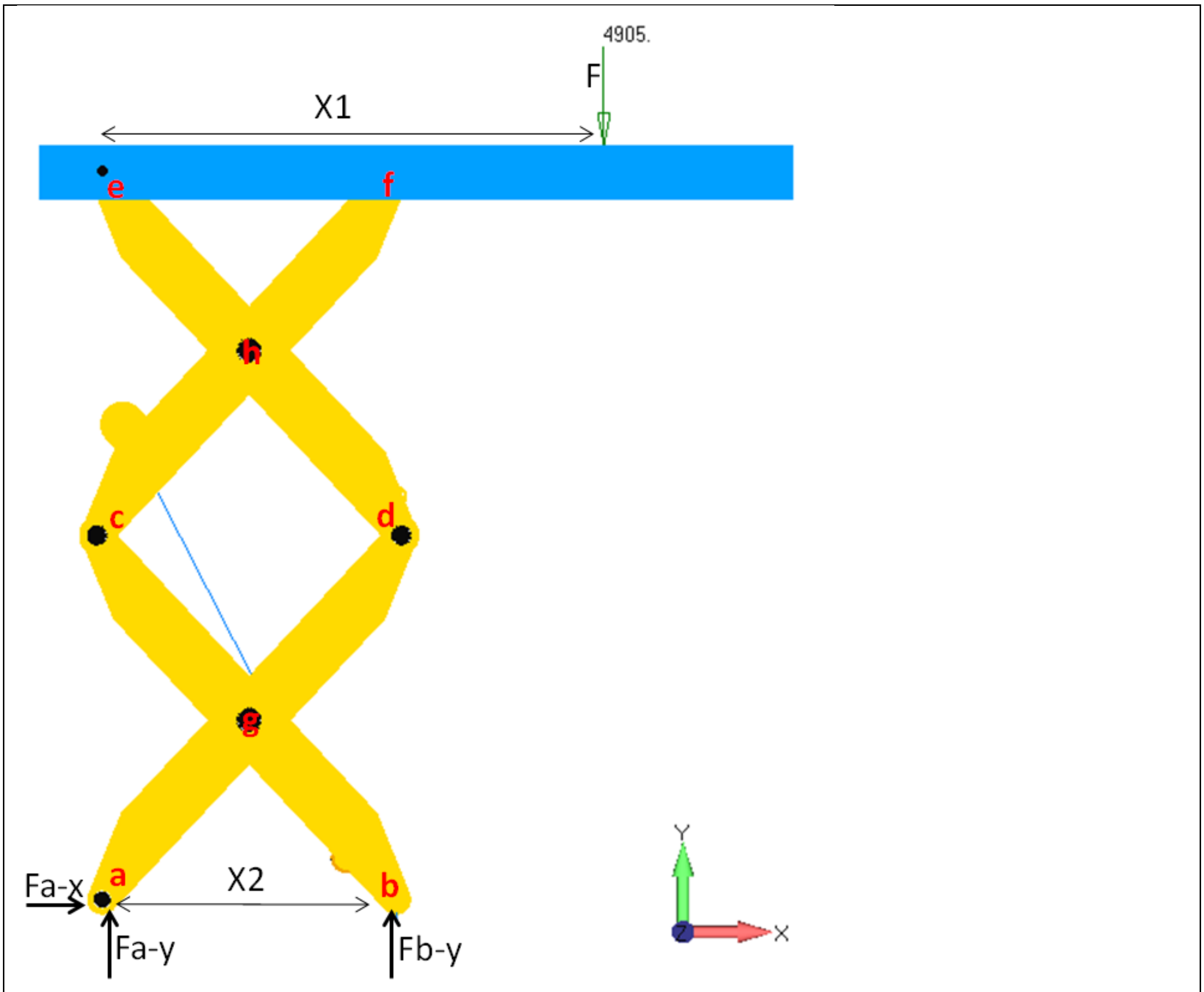


Figure 17: Determine forces at point A and B

The first step is determining the forces at point a and b in figure 17. The force and moment equilibrium is considered for this situation:

$$\sum F_x = Fa_x = 0$$

$$\sum M_a = X_1 * F - X_2 * Fb_y = 0,$$

$$Fb_y = \frac{X_1 * F}{X2} = \frac{750 * 4905}{400} = 9197\ N$$

$$\sum F_y = Fa_y + Fb_y - F = 0,$$

$$Fa_y = -Fb_y + F = -9197 + 4905 = -4292\ N$$

The next step is to remove the upper deck and determine the forces at point e and f. This situation is indicated in figure 18. Due to the cam wheels the forces at point B and F only show a y-component.

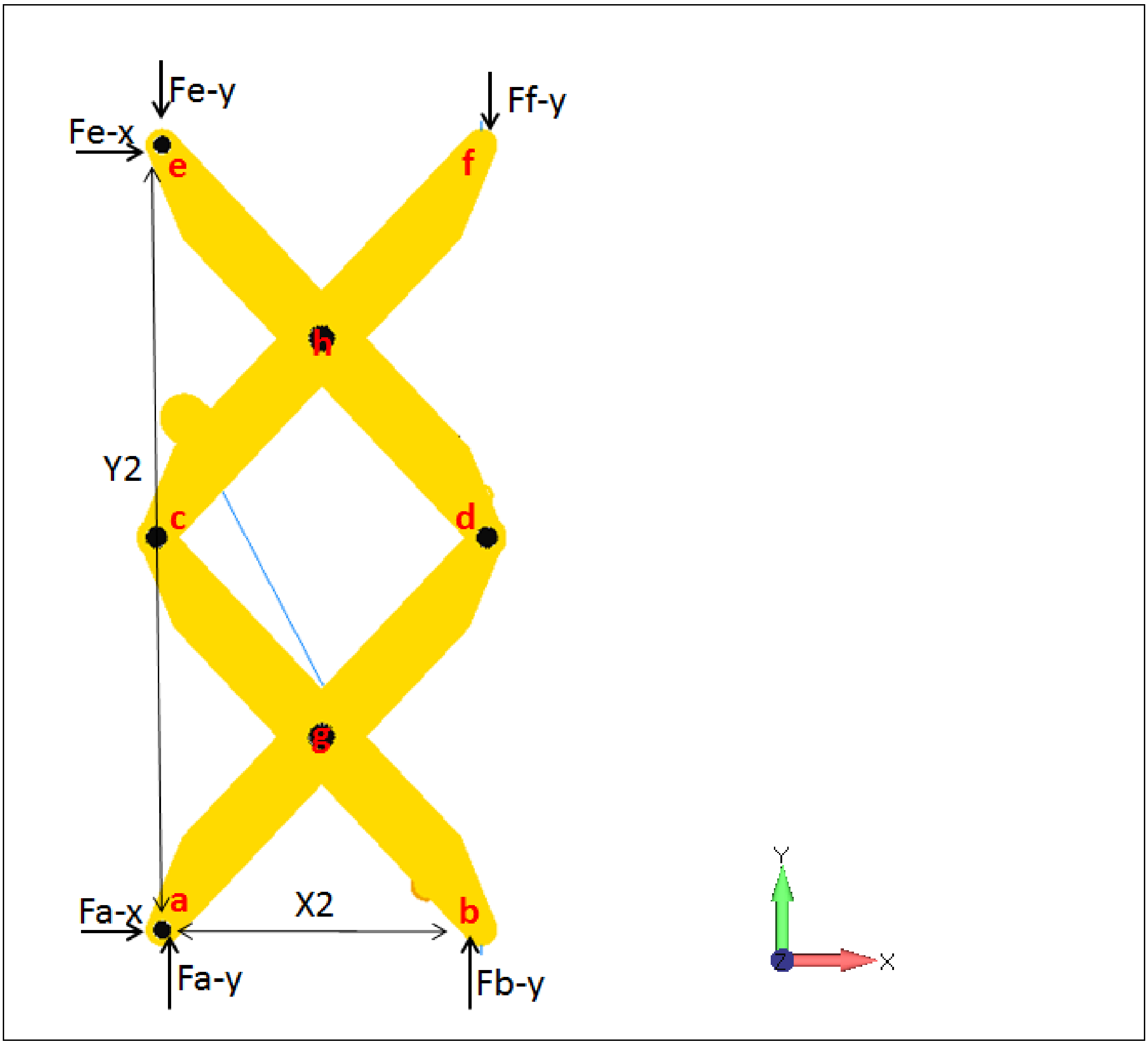


Figure 18: Determine forces at point E and F

The force and moment equilibrium are considered for this situation:

$$\sum F_x = Fa_x + Fe_x = 0,$$
$$Fe_x = 0\ N$$
$$\sum M_a = X_2 * Ff_y - X_2 * Fb_y = 0,$$
$$Fb_y = Ff_y = 9197\ N$$
$$\sum F_y = Fa_y + Fb_y - Fe_y - Ff_y = 0,$$
$$Fe_y = Fb_y + Fa_y - Ff_y = 9197 - 4292 - 9197 = -4292\ N$$

Next, the hinge forces at point D, H and G are determined. This can be done by considering the force and moment equilibriums of the two FBD's shown in figure 19.

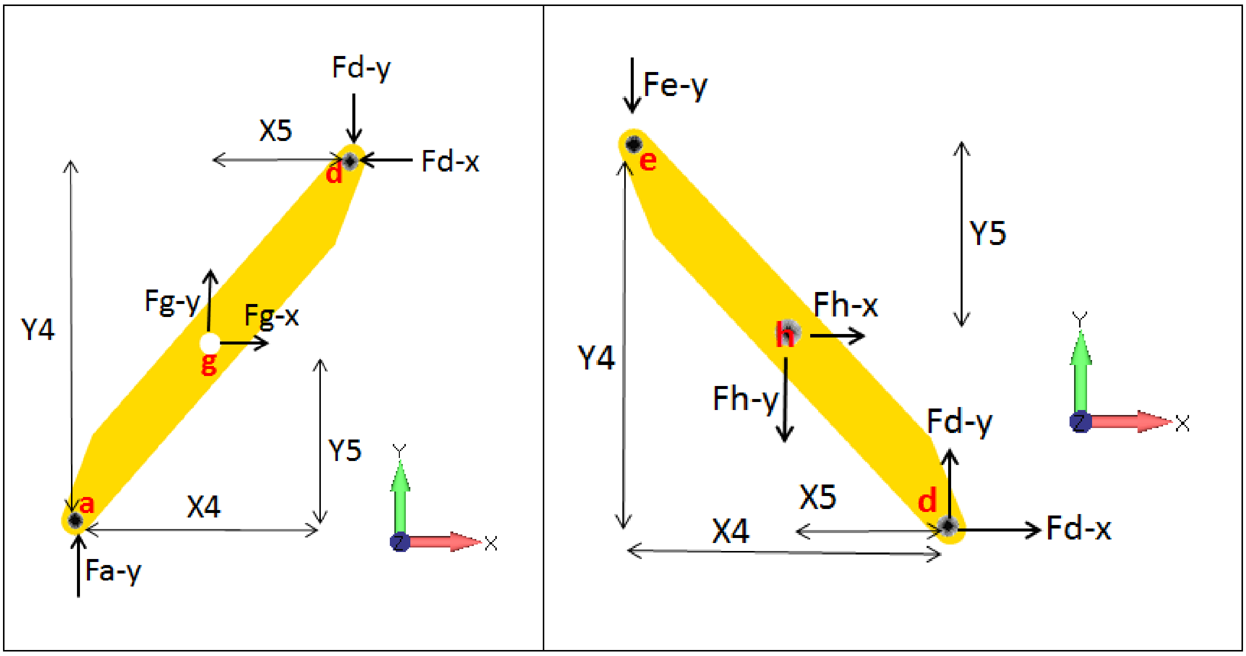


Figure 19: Determine forces at point D, H and G

The six equations of equilibrium are:

$$\sum F_x = Fg_x - Fd_x = 0 \quad (FBD\ left)$$
$$\sum F_x = Fh_x + Fd_x = 0 \quad (FBD\ right)$$
$$\sum F_y = Fa_y + Fg_y - Fd_y = 0 \quad (FBD\ left)$$
$$\sum F_y = Fd_y - Fh_y - Fe_y = 0 \quad (FBD\ right)$$
$$\sum M_d = X_5 * Fg_y - Y_5 * Fg_x + X_4 * Fa_y = 0 \quad (FBD\ left)$$
$$\sum M_d = Y_5 * Fh_x - X_5 * Fh_y - X_4 * Fe_y = 0 \quad (FBD\ right)$$

This system of equations can be solved using matrix-vector multiplication.

$$\begin{bmatrix} -1 & 0 & 0 & 0 & 1 & 0 \\ 1 & 0 & 1 & 0 & 0 & 0 \\ 0 & -1 & 0 & 0 & 0 & 1 \\ 0 & 1 & 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & -Y_5 & X_5 \\ 0 & 0 & Y_5 & -X_5 & 0 & 0 \end{bmatrix} * \begin{bmatrix} Fd_x \\ Fd_y \\ Fh_x \\ Fh_y \\ Fg_x \\ Fg_y \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ -Fa_y \\ Fe_y \\ -X_4 * Fa_y \\ X_4 * Fe_y \end{bmatrix}$$

$$\begin{bmatrix} Fd_x \\ Fd_y \\ Fh_x \\ Fh_y \\ Fg_x \\ Fg_y \end{bmatrix} = \begin{bmatrix} 0 \\ 4292 \\ 0 \\ 8584 \\ 0 \\ 8584 \end{bmatrix} N$$

The only remaining unknown forces are the hinge forces at point C and the cylinder forces. These forces can be obtained by considering the equilibrium equations from the FBD in figure 20.

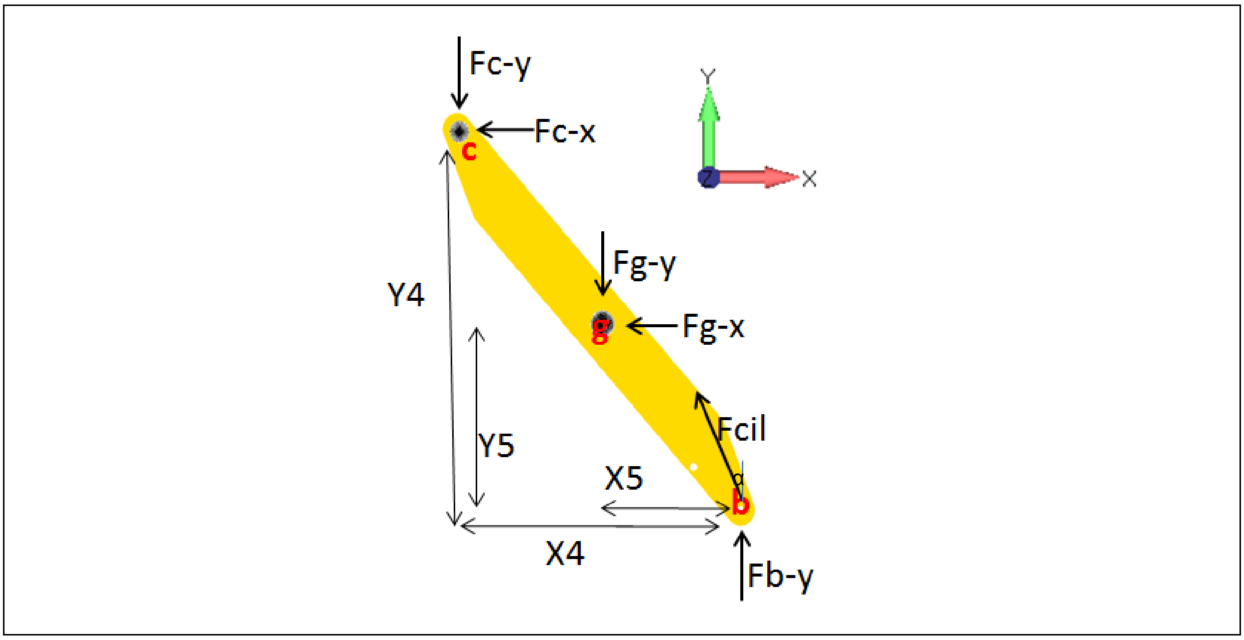


Figure 20: Determine forces at point C and cylinder force

The three equations of equilibrium are:

$$\sum F_x = -Fg_x - Fc_x - F_{cil} * \sin(\alpha) = 0$$
$$\sum F_y = -Fc_y - Fg_y + F_{cil} * \cos(\alpha) + Fb_y = 0$$
$$\sum M_b = -X_5 * Fg_y - Y_5 * Fg_x - X_4 * Fc_y - Y_4 * Fc_x = 0$$

These equations can also be solved with matrix-vector multiplication.

$$\begin{bmatrix} -1 & 0 & -\sin(\alpha) \\ 0 & -1 & \cos(\alpha) \\ -Y_4 & -X_4 & 0 \end{bmatrix} * \begin{bmatrix} Fc_x \\ Fc_y \\ F_{cil} \end{bmatrix} = \begin{bmatrix} Fg_x \\ Fg_y - Fb_y \\ X_5 * Fg_y + Y_5 * Fg_x \end{bmatrix}$$

$$\begin{bmatrix} Fc_x \\ Fc_y \\ F_{cil} \end{bmatrix} = \begin{bmatrix} 9932 \\ -16707 \\ -19965 \end{bmatrix} N$$

In figure 21 the calculated forces in this chapter are compared with forces from the FEM model. The FEM model is a 3D model. Therefore, some points have more components in the Z-direction. These forces are summed and compared with the calculated values. Also the ratio between the two obtained values are given in green. Point B, E, F and C deviate around the 10 %. Point A, H,G and D deviate between the 20 and 30 % and the cylinder forces are exactly the same. It can be concluded that, although there are some deviations, the forces are in the same order of magnitude and therefore, that the used FEM model is correct.

Punt A

3018..Beam EndA Pl1 Shear Force	3019..Beam EndA Pl2 Shear Force	total shear
1132	138	1141
503	167	530
498	-45	500
1337	-75	1339
	Total	3509

Analytical	ratio
4292	1.22

Punt B

	3022..Beam EndA Axial Force
	-4216
	-4158
Total	-8375

Analytical	ratio
-9297	1.11

Punt E

3018..Beam EndA Pl1 Shear Force	3019..Beam EndA Pl2 Shear Force	total shear
-1480	318	1514
-316	317	448
-237	-282	369
-1436	-282	1464
	Total	3794

Analytical	ratio
4292	1.13

Punt F

	3022..Beam EndA Axial Force
	-4016
	-4359
Total	-8375

Analytical	ratio
-9297	1.11

punt H

3018..Beam EndA Pl1 Shear Force	3019..Beam EndA Pl2 Shear Force	total shear
-3597	-10	3597
-3224	-10	3224
	total	6821

analytical		
0	Plane 1	
8584	Plane 2	ratio
8584	Total	1.26

Punt G

3018..Beam EndA Pl1 Shear Force	3019..Beam EndA Pl2 Shear Force	total shear
-3404	15	3405
-3417	15	3417
	Total	6821

Analytical		
0	plane 1	
8584	Plane 1	ratio
8584	Total	1.26

Punt D

3018..Beam EndA Pl1 Shear Force	3019..Beam EndA Pl2 Shear Force	total shear
1565	4	1565
-1786	4	1786
	Total	3352

Analytical		
0	plane 1	
4292	plane 2	ratio
4292	Total	1.28

punt C

3018..Beam EndA Pl1 Shear Force	3019..Beam EndA Pl2 Shear Force	total shear
8109	4557	9302
-8084	-4584	9293
	Total	18595

Analytical		
9932	plane 1	
-16707	plane 2	ratio
19436	Total	1.05

Cylinder force

	3022..Beam EndA Axial Force
	-19962

Analytical	ratio
-19965	1.00

Figure 21: Excel sheets to verify FEM model

8 Conclusion and recommendations

In this report the performance of a mobile lifting table is assessed. The assessment is done using the requirements specified in chapter 3. In this section the conclusions are drawn and recommendations are given to increase the performance of the table.

In chapter 4 the safety requirements with respect to gaps are discussed. It can be concluded that the table does not conform to the European standard with respect to most of the gaps. It is recommended that the geometry of the table will be revised to make it conformal.

In paragraph 6.3.1 the platform deflection is calculated for different load cases as specified earlier in this report. The tilting angle is too big in 2 of the 16 considered cases. It is recommended to increase the stiffness of the table, so the platform deflection will be less and the tilting angle will be conformal with the European standard.

In paragraph 6.3.2 a stability analysis is performed for different load cases as specified earlier in this report. In 6 of the 8 stability cases the lifting table does not conform with the European standard. The stability can be increased be making the stabilizing moment bigger or the overturning moment smaller. This can be done for example by making the center of gravity lower or by adding more weight to the bottom deck. Another possibility is making the bottom deck wider and placing the wheels on the corners of the deck. Taking into account the amount of instability of the table both of these measures will be required.

At last the stresses are analyzed in paragraph 6.3.3. From this analysis it follows that the stresses are below the allowable stress for most cases. High stresses occur in the bottom deck where the wheels are attached and the location where the cam wheels are attached to the scissors. It is recommended to change the geometry at this locations the lower the stress level.