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Abstract

The existing design of marine and offshore elevators are mainly based on the expertise out of land-based elevators, with higher safety factors and all existing requirements being fulfilled. Thyssenkrupp Elevator Marine & Offshore Division believes that their existing guide structure is overdimensioned, which consists of guide rails, brackets and associated fixation. This results in additional weight and increased cost of material and installation. The cost is always important for the customer, but the weight of the components has also been a major focus the recent years. This is because less weight would allow more goods and passengers on the ship, and thus an attractive feature when selecting the elevator supplier. Thyssenkrupp Elevator is therefore seeking for a research opportunity to increase their knowledge about elevator guide structures on ships, as well as being proactive and stay competitive in the maritime and offshore environment.

A thorough literature study on current rules and requirements for elevators on ships, with special attention to the guide structure, has been carried out. Four different methods for calculating the load cases acting on the structure are developed from this. Thyssenkrupp Elevator are usually designing their elevators for ships being classed according to DNV GL, but there is also an upcoming yacht that is to be classed according to Lloyd's Register. Two of the methods are therefore applicable for DNV GL, while the two others are applicable for Lloyd's Register. The first method is based on the theory of free harmonic vibration, while the three others are based on rules, requirements and equations defined by the two classification societies.

A mapping of load cases has been evaluated to find the magnitude of the transverse and longitudinal forces, caused by the different ship types. This was done by dividing different ships into three classes based on similarities in type, dimensions and resulting motions. The result shows that the motions of the expedition vessels are subjecting the guide structure for the largest forces, while the motions of the cruise liners are subjecting the structure for the smallest forces. Based on the result, it is not recommended to define standardized load cases, but instead calculate the structure using customized load cases for each project.

An assessment of two different bracket solutions have been carried out. This includes the standard land-based bracket from thyssenkrupp Aufzugswerke, which is developed at the headquarter in Neuhausen, and the customized offshore bracket from Thyssenkrupp Elevator Marine & Offshore Division, which is developed at the department in Ålesund. Furthermore, it is shown from the structural FEM analysis that the standard solution can be used for offshore applications as well. Its strength capacity is not as great as the customized solution, but applicable for loads up to 4000 kg. As for the customized solution, it is recommended to reduce the plate thickness and width of the bracket, since it still would be within acceptable safety factors after these changes.

Four different combinations of guide rails and brackets have been examined to find the most optimized solution for the guide structure. In terms of installation cost, the result shows that a lot of savings could be achieved with the use of the standard bracket instead of the customized bracket. Reducing the guide rail dimension would also save time for installation, but not as significant as the change of brackets. The opposite applies for the cost of material, since the savings are dependent on the reduction of guide rail dimensions. Comparing the cost of material and installation, it is shown that the savings are achieved in the installation phase. As for the weight of the components, this is influenced by the reduction of guide rail dimensions, and not by using one or the other of the brackets.

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Acknowledgements

This thesis is submitted for concluding the degree of Master of Science in Mechanical and Structural Engineering and Material Science, with specialization in Mechanical Engineering, at the University of Stavanger. The content in this thesis is based on a development project for thyssenkrupp Elevator Marine & Offshore Division, because there was a need for research on offshore elevator guide structures. The research presented in this thesis has been carried out at their department in Ålesund during the spring semester of 2017.

Working in the environment of thyssenkrupp Elevator has been a great experience, and it has given me a lot of motivation. I have been in contact with their suppliers regarding my findings, on-site inspections at the shipyard, meetings with the customers and provided my suggestions on upcoming projects regarding guide rail dimensions and bracket spacing.

I would like to express my appreciation to my supervisors from the company, Samuel Bauer and Geir Bøstrand, and the rest of the employees at thyssenkrupp Elevator for their hospitality, guidance, support and sharing of knowledge. I would also like to express my gratitude to my supervisor at the University, R.M. Chandima Ratnayake, for suggesting the thesis and for his guidance, advises and feedback on the project. In addition, the rest of the professors and employees at the Department of Mechanical and Structural Engineering and Material Science for knowledge and learning.

Øie in Stavanger, 14.06.2017

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Abbreviations

All abbreviations that are used in the thesis are listed here alphabetically. Most of the symbols are on the other hand defined in the main text and just a selection is listed here.

ABS	American Bureau of Shipping
AE	Aft End
BL	Baseline
CAD	Computational Aided Design
CL	Centerline
DAF	Dynamic Amplification Factor
DNV GL	Det Norske Veritas Germanischer Lloyd
EN	European Standard
FE	Fore End
FEM	Finite Element Method
IACS	International Association of Classification Societies
ISO	International Organization for Standardization
LR	Lloyd's Register
TBN	To Be Named
TK	thyssenkrupp Elevator Marine & Offshore Division, Ålesund in Norway
TKA	thyssenkrupp Aufzugswerke, Neuhausen in Germany
WL	Waterline

Symbols

A	Heave amplitude
E	Young's modulus, 206 000 MPa used for steel
f	Stress factor
g/g_0	Gravitational acceleration, 9.81 m/s^2
σ_{all}	Allowable stress
σ_b	Bending stress
σ_c	Compression stress (= buckling stress)
σ_{cr}	Critical compression stress
σ_k	Buckling stress (= compression stress)
σ_t	Tension stress
σ_y	Yield stress
T_θ	Roll period
T_φ	Pitch period
θ	Roll angle
φ	Pitch angle
x	X-coordinate
y	Y-coordinate
z	Z-coordinate, defined by DNV GL
z	Vertical coordinate of the ship's rotation center
$\mathrm{Z_p}$	Z-coordinate in calculation of pitch, defined by LR
$\mathrm{Z_r}$	Z-coordinate in calculation of roll, defined by LR
´	Arcminutes

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1 Introduction

The first two sections present the background for the thesis, both in terms of the company and their objectives for the project. The structure of the thesis, with the content of each chapter, is then described in the last section.

1.1 Background

Thyssenkrupp Elevator Marine & Offshore (hereby referred to as TK) is a division in the multinational thyssenkrupp Elevator AG. They design, install and maintain passenger and service elevators for marine and offshore applications, such as cruise liners, expedition vessels, luxury yachts, supply & service ships, ferries and other vessels in their interest. Oil rigs, offshore fish farms, offshore windmills and special applications, such as gangway concept for service ships, are also a part of their line of business [1], [S. Bauer, Head of Marine & Offshore Division Norway, thyssenkrupp Elevator. Personal communication via mail, 18.12-16]. Most of the elevators, or lifts, are delivered from the New Installation department of thyssenkrupp Aufzugswerke in Neuhausen (hereby referred to as TKA), which is a subsidiary of thyssenkrupp Elevator AG. The elevator systems are developed and manufactured by TKA, before being customized and installed on different floating constructions by TK.

The existing design of marine and offshore elevators are mainly based on the expertise out of land-based elevators, with higher safety factors and all existing requirements being fulfilled. TK believes that their existing guide structure is overdimensioned, based on the weight of the structure and the cost of material and installation. The guide structure is in this thesis defined as the structure consisting of the guide rails, brackets and associated fixation. In recent years, there has been a lot of attention to the weight of the components, and something the shipyard is always trying to reduce. Less weight would allow more passengers and goods aboard, and thereby increase their income. TK is therefore seeking for a research opportunity to increase their knowledge about elevator guide structures on ships, as well as being proactive and stay competitive in the maritime environment.

1.2 Objective

TK have been working with the current guide structure for years, and the objective is defined based on their experience and feedback from earlier projects. It is to develop an optimized guide structure, mainly for centrally guided and suspended elevators, which fulfills all existing requirements and avoids overengineering based on the weight of the structure and cost of material and installation. The structure should be optimized for the best possible solution of guide rails, brackets and associated fixation.

The process of developing a more optimized solution is based on a mapping of load cases, a literature study on standards, rules and requirements, investigating the state of land-based and offshore elevator design and an assessment of brackets. The objective is further divided in two goals:

1) Is it possible to use smaller guide rail dimensions and larger distances between the brackets, and still be within the requirements from the classification societies?
2) Can the standard land-based bracket from TKA be used for TK's offshore and maritime applications?

The assessment of the brackets should also include the current bracket solution from TK. This bracket is redesigned for each project, and it is desirable to optimize the design towards a more standardized solution, such that the configure to order would increase.

1.3 Scope of work

Most of the fundamental theoretical background for elevators on ships is presented in chapter 2. This involves an introduction of the main elevator components to give the reader the required knowledge on elevator technology, and a more detailed presentation of the guide structure to give the reader the understanding of the components that are being evaluated.

Chapter 3 of this thesis covers the rules and requirements for the guide structure in elevators on ships, defined by two different classification societies. This includes requirements on forces resulting from ship motions, how the load cases are defined and restrictions on allowable stresses and deflections.

The ship motions and methods of calculating the accelerations and forces are described in chapter 4. This includes illustrations of how the forces are transmitted from the car, via the guide structure, and to the trunk. A description of the guide rail calculations with associated governing equations is also presented.

Chapter 5 presents the research and development phase of the thesis, which includes the different calculation methods, an assessment of the two bracket solutions based on trunk layout and forces from the ship motions, and the ship classes used to map the load cases. The cost savings are also presented, which includes an estimated cost of installation for the four selected combinations of guide structure.

The results are presented in chapter 6, including an example on guide rail calculation, FEM analysis of the brackets, proposed changes to the customized bracket solution and cost savings based on weight reduction and shorter installation time. The results are thoroughly discussed in chapter 7 and the conclusion is given in chapter 8. Lastly, the calculation methods and FEM analysis are then presented in the appendices.

2 Theoretical background

TK delivers either hydraulic lifts or electric lifts, also called traction lifts. A hydraulic lift is power by an electrically driven pump unit that transmits hydraulic fluid to a jack. This pump unit is usually located in the room next to the pit. The jack is the hydraulic actuating unit that acts directly on the car, i.e. direct acting lift, or connected through ropes, i.e. indirect acting lift as illustrated to the right in Figure 1. The traction lift consists of an elevator that is balanced with the use of a counterweight, as shown to the left in Figure 1. This counterweight is equal to the weight of the elevator plus 40-60 % of the rated load, and the system will therefore be in equilibrium when the elevator is half full. It is powered by an electric motor that is connected to a driving sheave, which rotates the system by friction. Each sheave is made with grooves for the rope that increases the friction and guiding. The car and the counterweight are connected with suspension ropes, allowing the traction drive to lift or lower the car. This reduces the need for machine capacity compared to an elevator with a winch drum. All traction lifts are delivered with either an integrated machine room in the headroom, so called “machine room-less solution”, or a separate machine room either located besides, on top or below the trunk. A controller unit is also installed nearby the driving machine to control the running and stopping of the elevator.

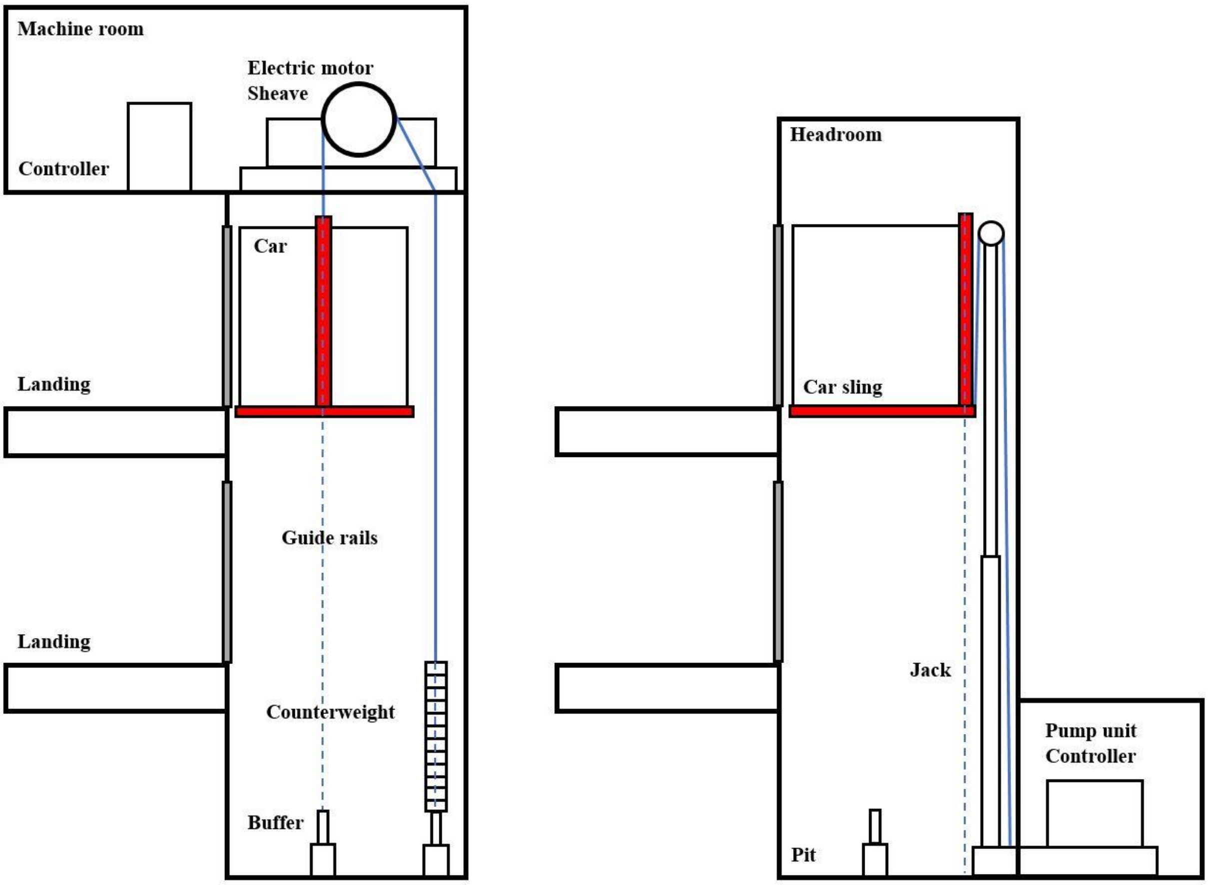

Figure 1. Traction vs. hydraulic lift.

The term elevator denotes the lifting equipment consisting of a car (1), which is supported by a car sling (2) and running between guide rails (3) that are attached to the trunk (4) with brackets (5). The car is an enclosed compartment that carries passengers, cargo and/or staff between each deck, while the car sling is the supporting frame connecting the car to the drive system. It consists of beams assembled into a frame, usually with a platform positioned in the bottom. There are many different car sling solutions, but the most common is the centrally guided frame with suspension ropes (6) attached to the sling and counterweight (7). The car sling is installed with either guide shoes (8) or roller shoes (9) at each side, both on the upper and lower part of the sling. Guide and roller shoes are meant for guiding the sling onto each guide rail, where roller shoes are used for a larger rated speed due to more travel comfort. All elevators are also installed with a overspeed governor (10), which is the mechanism that causes the safety gear to be activated if the predetermined speed is exceeded.

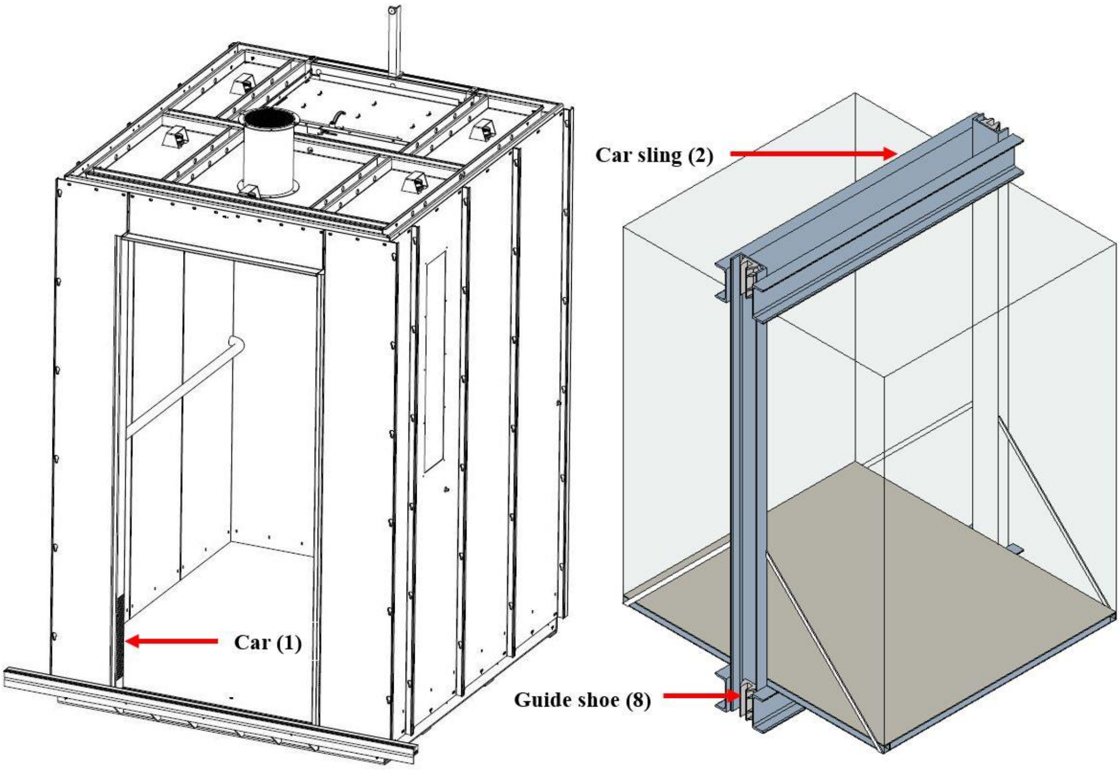

Figure 2. Elevator car and car sling [internal document, TK].

Figure 3. Car with stainless steel interior [internal document, TK].

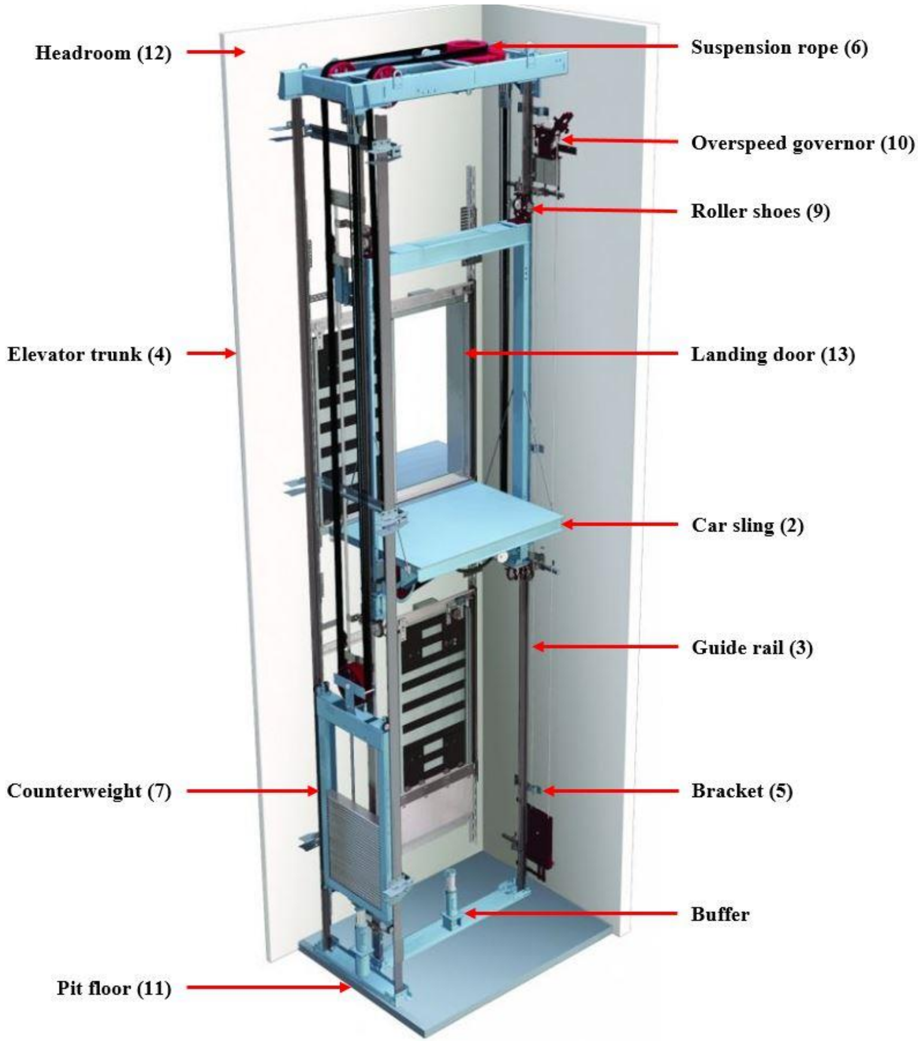

Figure 4. Main components [2].

The safety gears are installed on each side of the car sling, at a certain distance above the two guide shoes installed in the lower part of the sling. TK uses two different safety gears; bi-directional progressive gears for their offshore traction lifts and single-acting instantaneous gears for their offshore hydraulic lifts. The progressive safety gear is a braking action with an absorbing mechanism to increase the elastic behavior of the system, while the instantaneous safety gear is an almost immediate gripping action with short breaking length. Instantaneous safety gears are also delivered with captive rollers, which makes the braking action more elastic compared to ordinary clamps.

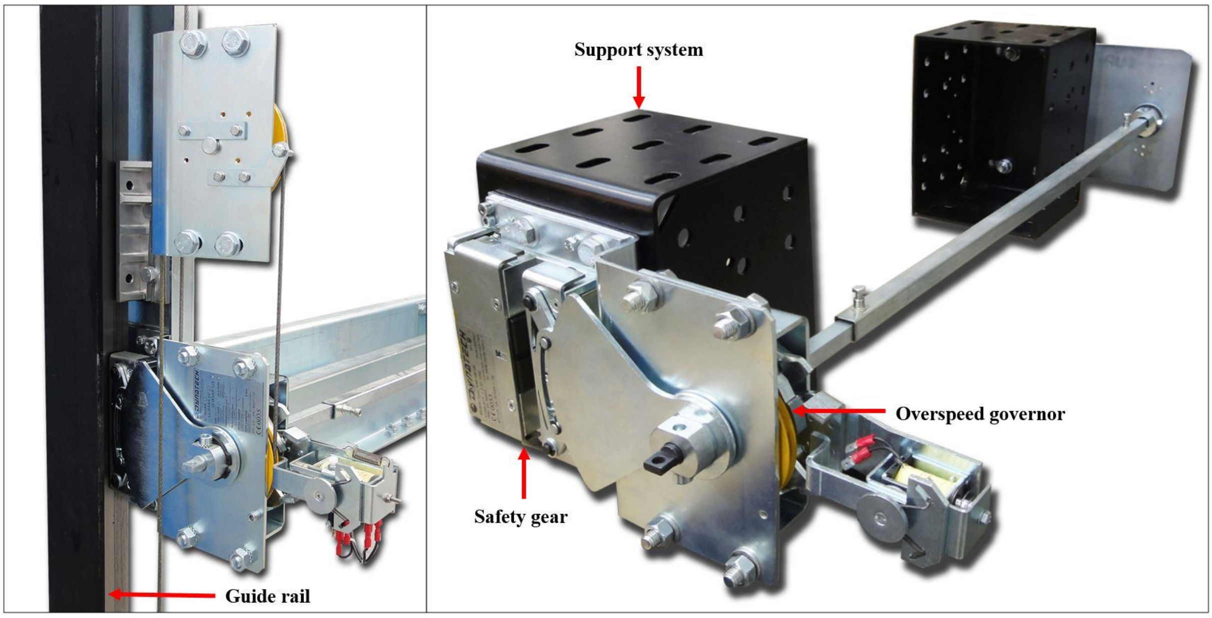

Figure 5. Safety device [3].

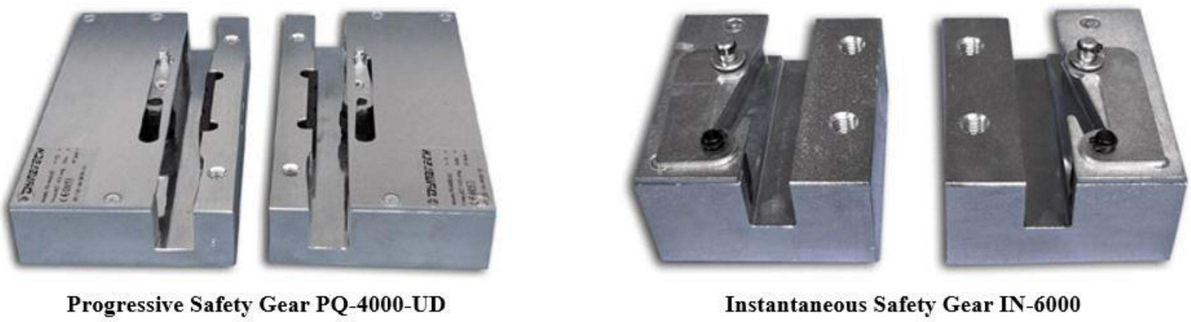

Figure 6. Progressive vs. instantaneous safety gears [3].

A trunk is an opening through several ship decks, enclosed by bulkheads and supported by beams and stiffeners. The trunk extends from the pit floor (11) to the roof of the headroom (12), consisting of openings at each deck for landing doors (13), and a ladder covering the whole travel for emergency measures. A buffer is also installed in the pit, which is a resilient mechanism to absorb the impact in the end of the travel, both for the elevator and the counterweight. Figure 7 shows an example of different trunk placements inside a cruise ship, and the size relative to the ship, while Figure 8 illustrates two trunks with cut outs for the landing doors, the machine room at the top and the pit in the bottom.

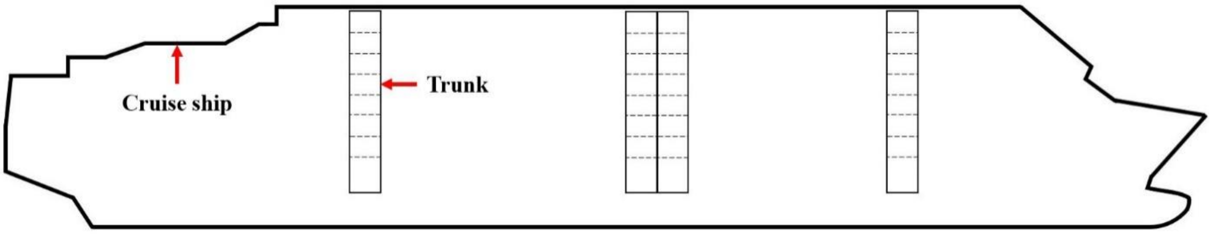

Figure 7. Example of trunk placements in a ship.

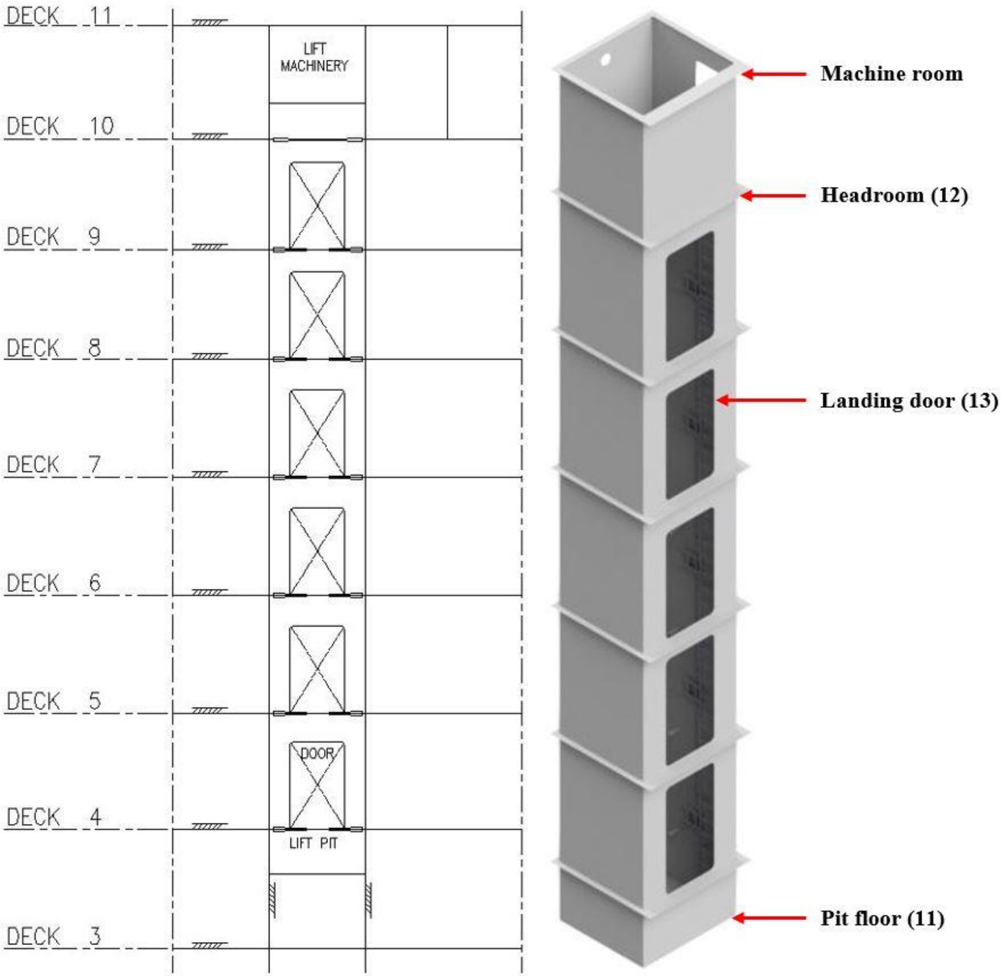

Figure 8. Offshore steel trunks [internal document, TK].

2.1 Guide structures for elevators

The guide structure is the assembly of guide rails, brackets and associated fixation as shown in Figure 9. Guide rails are usually made from drawn steel in standard dimensions and cut into equal lengths of 5 meters. They are connected by fishplates and associated parts in the installation of the structure, as shown in Figure 10. The material quality is of either S275JR, which refers to machined steel (denoted as B), or S235JR, which refers to cold drawn (denoted as A). They are attached to the trunk by brackets and guided all the way from the pit to the headroom. Both the car and the counterweight, with associated overspeed governors, are guided with at least two guide rails. An additional guide rail is often used on cantilevered hydraulic car slings, which prevents the rotation that occurs when it is only supported at one end. This guide rail is placed on the opposite side of the jack.

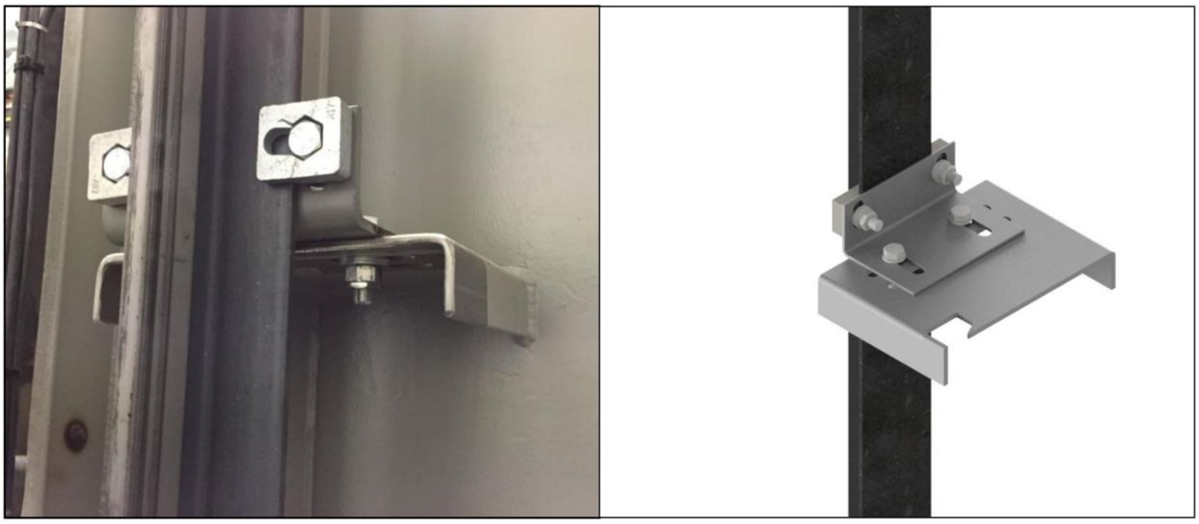

Figure 9. Guide structure of elevator cars [internal document, TK].

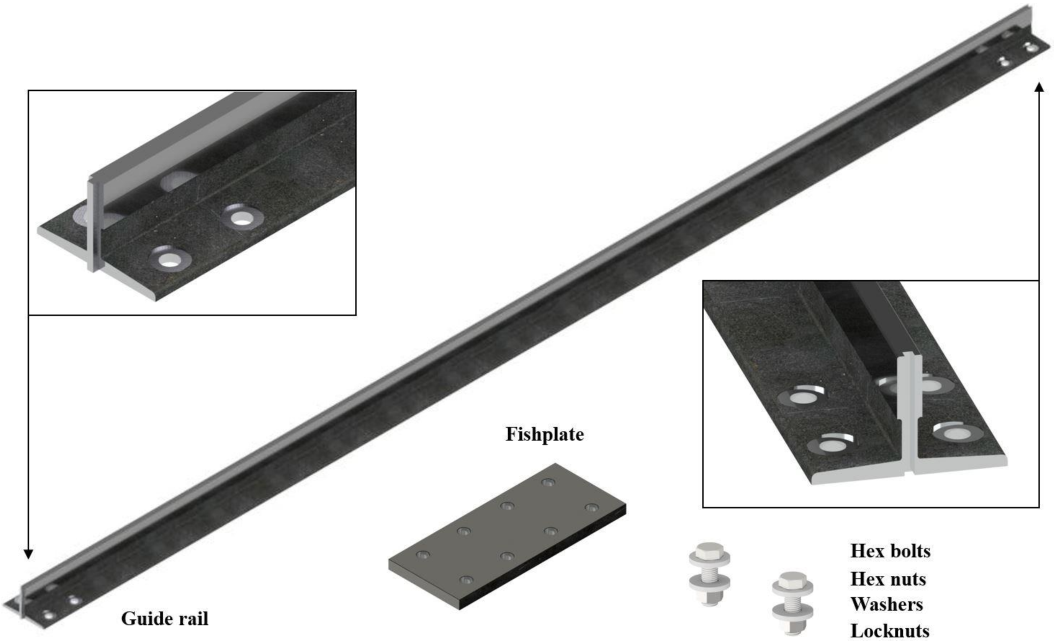

Figure 10. Guide rail with fishplate and accessories.

The research in this thesis is mostly directed to the guide rails for the car, since these are assumed to be overdimensioned. However, the calculation methods presented in section 5.2 are applicable for all types of guide rails given that the distance to the mass center is known. A lot of attention should also be directed to the guide rails for the counterweight due to its mass. Furthermore, four sets of guide rails are currently being used for TK's traction lifts with centrally guided car slings; the largest for the car, the second-largest for the counterweight and the two smallest for the overspeed governors. And these are the most used guide rail dimensions based on their offshore projects:

- Car T127-2/B
- Counterweight T90/B
- Overspeed governor, car T70/B
- Overspeed governor, counterweight T50/A

The bracket is the assembly of beams, profiles and/or channels that constrain the guide rails to the trunk. A bracket for the car, could for instance be an assembly of a L-profile bolted on top of a U-channel as seen in Figure 11. There are many different brackets in an elevator system, depending on the component they are supporting and how the trunk is built. The car, counterweight and governors are all supported by different brackets due to their application. There are also two ways of supporting the car brackets, which depends on the orientation of the car and the counterweight as shown in Figure 12. The elevator car brackets are either directly welded to the trunk or assembled on an existing counterweight frame. The latter solution is not included in this thesis.

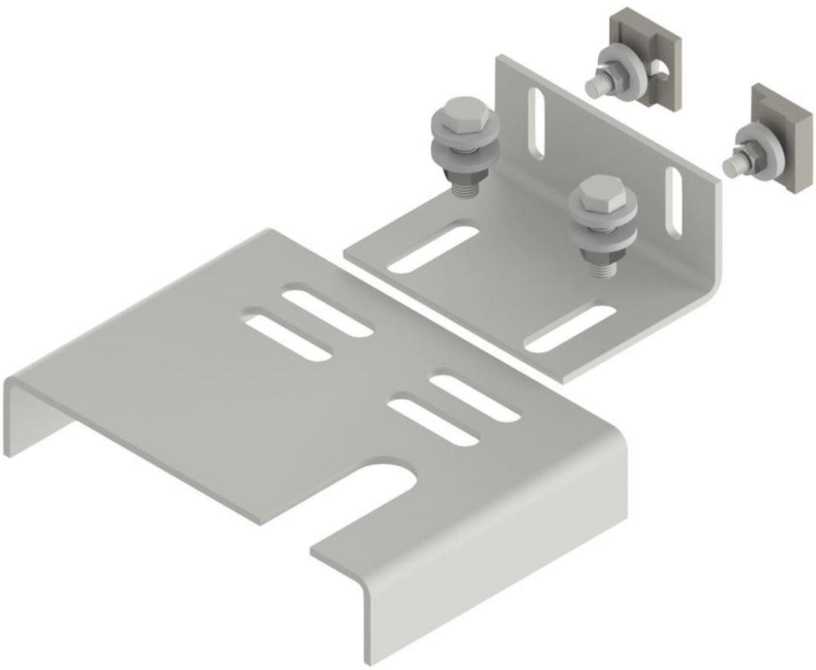

Figure 11. Bracket with accessories.

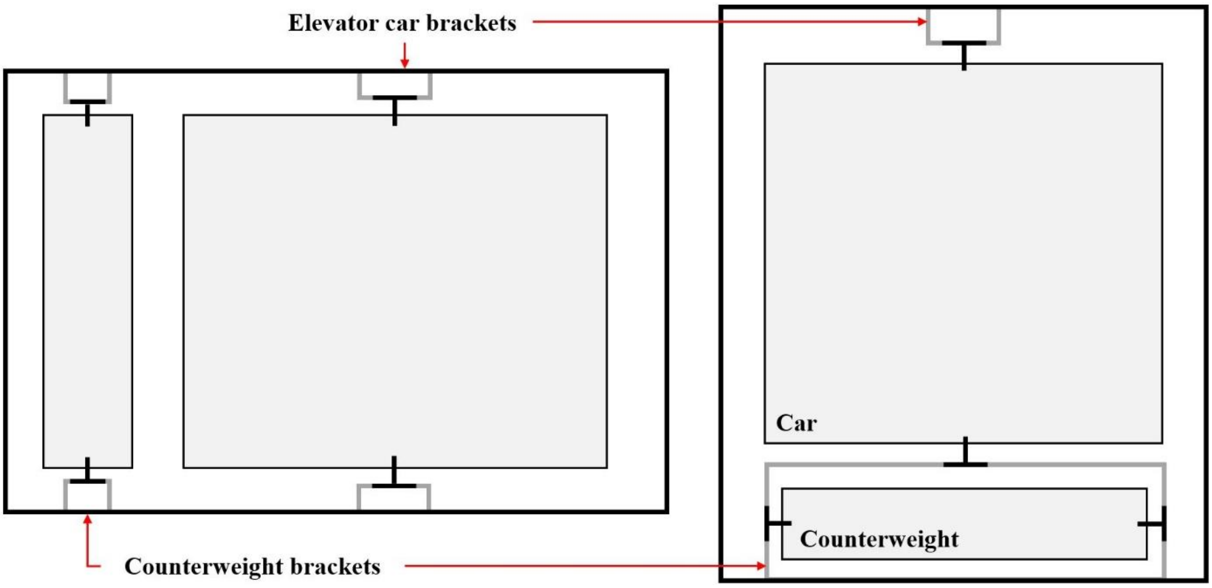

Figure 12. Car brackets and counterweight brackets.

From an engineering perspective, there are also challenges in terms of standard brackets versus customized brackets. Standard bracket dimensions are rarely used in offshore trunks, due to the variety of trunk layouts and orientation of components. Offshore trunks are also more compact than land-based trunks, and built with bulkhead stiffeners on the inside. This makes the surface of the trunk more challenging to work with compared to a plane concrete surface. The distance from the trunk to the car is also an important factor in the design of the brackets, since this varies from project to project. These issues are all considered in the assessment of the brackets, which is presented in chapter 5.3.

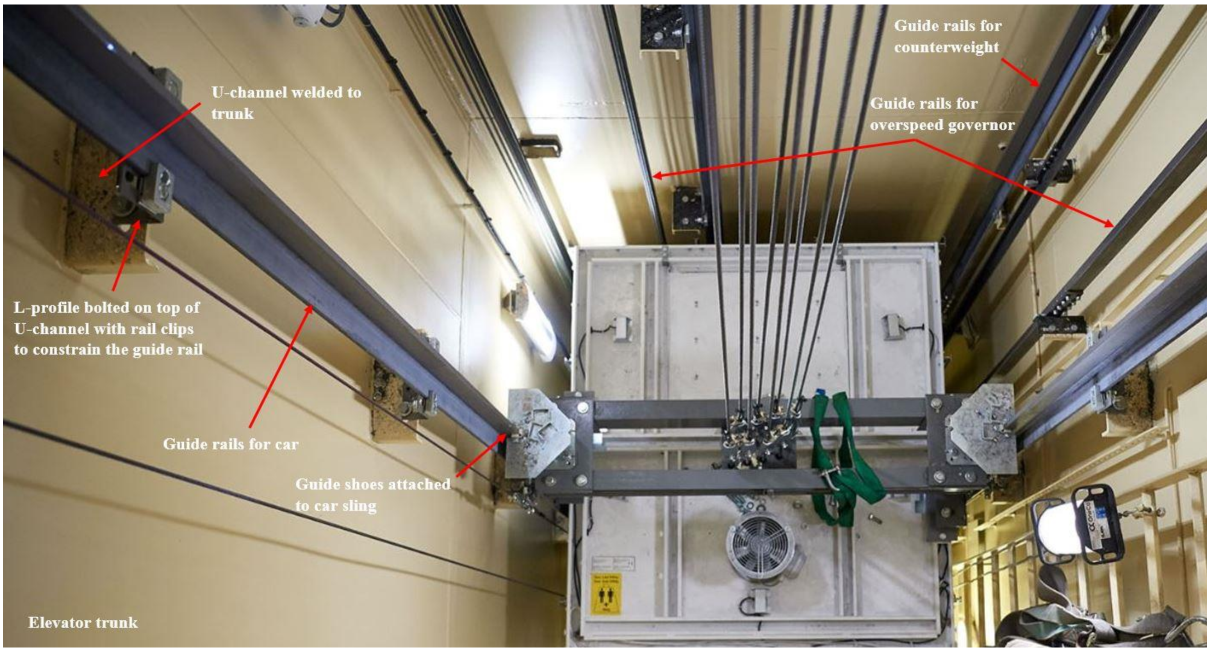

Figure 13. The guide structure inside an elevator trunk [internal document, TK].

3 Rules, requirements and guidelines

All elevators delivered by thyssenkrupp Elevator are designed, manufactured and installed according to DNV GL's standard for Rules for Lifts [4]. The standard builds on ISO 8383 [5], EN 81-1 [6] and EN 81-2 [7]. ISO 8383 contains the requirements for offshore elevators, while EN 81-1 and EN 81-2 are only valid for land-based elevators. EN 81-1 and EN 81-2 are being replaced by EN 81-20 [8] and EN 81-50 [9], which will take effect during the summer of 2017. DNV GL will also replace their existing standard for Rules for Lifts with a new standard, which will build on EN 81-20, EN 81-50 and ISO 8383. The standard is expected take effect in the last half of 2017 and become the state of art on rules, requirements and guidelines for elevators on ships and other offshore constructions [H. Jensen, Senior Engineer, DVN GL Lifting Appliances. Personal communication via mail, 06.02-17]. Figure 14 presents the upcoming changes for the standard.

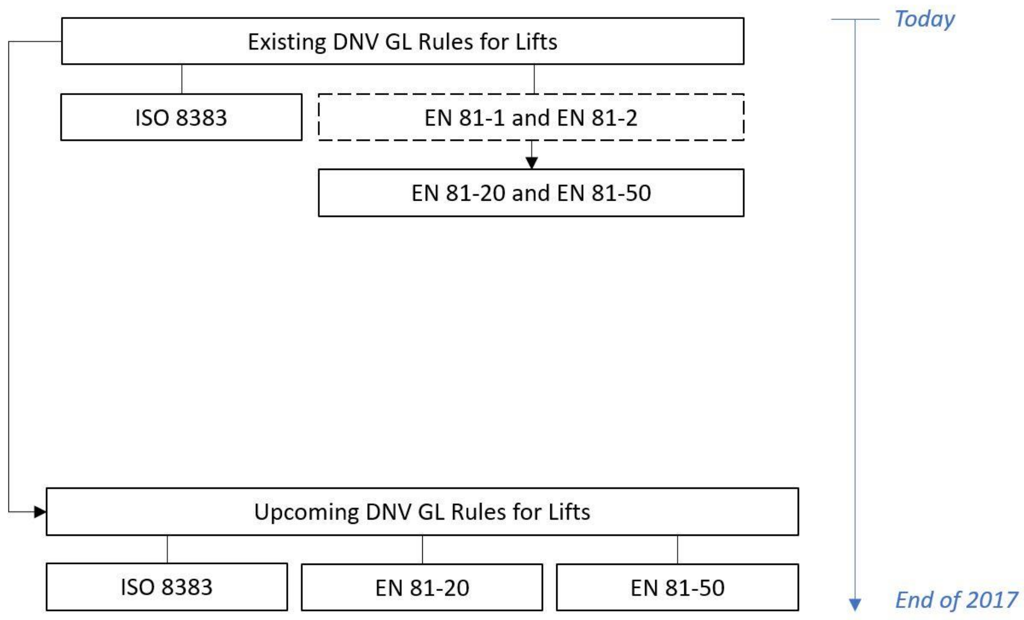

Figure 14. DNV GL's standard for Rules for Lifts.

TK is however dependent on which class the ship is being built according to, something that is decided by the customer. This means that the elevators need to be applicable for most of the classification societies. TK is usually designing their elevators for ships that are classed according to DNV GL, but there is also one example with a yacht which is being classed according to LR. Because of the increasing interest for luxury yachts and expedition vessels, LR will be included in the calculation of guide rail strength. In addition, it is likely that TK will be working on several projects classed by LR, and it is therefore wise to establish a good procedure for approval of documentation.

3.1 Ship motions

Most authoritative classification societies around the world, such as DNV GL, LR and ABS requires an elevator installation that is designed according to certain angles and periods, both for operating and stowed condition. Operating condition refers to an elevator running up or down, while stowed condition refers to an elevator that is parked at a certain deck height. Their requirements states that the installation shall be able to operate under the following environmental conditions listed in the mid column in Table 1. Furthermore, the associated machinery and structure are to withstand the forces resulting from the environmental conditions specified in the right column in Table 1. The requirements from ABS are also included for comparison with the two other societies.

DNV GL’s Lifting Appliances department accepts deviations from the requirements, if the shipyard or the Ship Structures department could provide documentation on different ship motions. Approval for other dynamic motions is usually done with offshore platforms, fixed to the seabed, or offshore fish farms. It is however unusual that the shipyard presents documentation on different ship motions to the Lifting Appliances department. In general, all deviations from the requirements are evaluated individually, and then approved or declined by an “Approval Center” based on risk assessments. [H. Jensen, Senior Engineer, DVN GL Lifting Appliances. Personal communication via mail, 06.02-17].

The requirements in the standard are strict, and it is only possible to calculate for different periods in stowed condition, since this requirement is not specified in the standard. On the other hand, calculating with other periods is only valid if the Ship Structures department of DNV GL approves it. The same period as for operating condition is therefore used.

The standard from LR is also developed by their Lifting Appliances department. However, they do not need to consult to other departments for ship specific calculations. If ship characteristics are known, different angles and periods can be calculated without being approved by any department responsible for ship motions. However, there are certain restrictions for the motions in stowed condition. The pitch angle φ shall not be greater than 8°, while the roll angle θ shall not be less than 22° and not greater than 30°.

Source	***Operating condition***	***Stowed condition***
DNV GL [4]	Roll: ± 10°, period 10 s Pitch: ± 5°, period 7 s Heave: A = 0.0125 L, period 10 s	Roll: ± 22° 30’ (22.5°) Pitch: ± 7° 30’ (7.5°) Note: 30´ = 0.5°
ISO 8383 [5]	Roll: ± 10°, period 10 s Pitch: ± 5°, period 5 s Heave: A < 3.8, period 10 s A = 3.8 - 0.01 (L - 250)	-
LR [10]	Roll: ± 10°, period 10 s Pitch: ± 7.5°, period 7 s	Roll: ± 22.5°, period 10 s Pitch: ± 7.5°, period 7 s Heave: A = 0.0125 L, period 10 s *If ship characteristics are known:* $22° \leq \theta \leq 30°$ $\varphi \leq 8°$
ABS [11]	Roll: ± 10°, period 10 s Pitch: ± 5°, period 7 s Heave: A < 3.8, period 10 s A = 3.8 - 0.01 (L - 250)	Roll: ± 30°, period 10 s Pitch: ± 10°, period 7 s

Table 1. Summary of operating and stowed condition.

3.2 Load cases

DNV GL requires that the strength of the guide rails, brackets and associated fixation shall be able to resist the forces acting from the car or counterweight, due to the motion of the ship as defined in Table 1. For stowed condition, they assume both the car door and landing door to be locked, and the car empty of passengers. This means that one should not include the rated load, i.e. capacity, in the calculation for passenger lifts in stowed condition. The counterweight in stowed condition at a maximum height will therefore be the worst load case.

LR have defined the same requirements and the same components to include in each condition, but their calculation methods do also include the dynamic components of roll and pitch, caused by the heave motion of the ship. They do also combine the forces in the calculation of the exceptional condition, instead of only calculating with the vertical force. An exceptional condition refers to as an unusual condition, such as safety gear operation. Furthermore, there is no condition which stand out as the worst load case due to their way of calculating, meaning that operating, stowed and exceptional condition must always be checked for both car and counterweight. All load cases from DNV GL and LR are presented in Figure 15, and the difference between their interpretation is presented in Table 2.

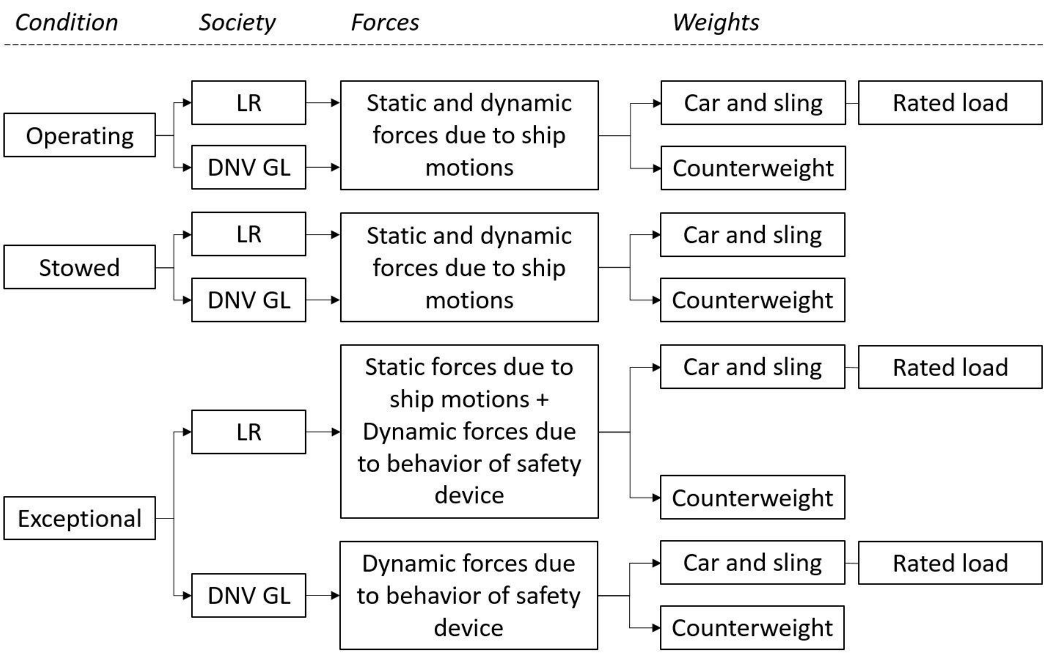

Figure 15. Load cases.

Forces	***LR***	***DNV GL***
Due to ship motions	$F_{\text{static roll}} + F_{\text{dynamic roll}} + F_{\text{dynamic heave (at roll angle)}}$ $F_{\text{static pitch}} + F_{\text{dynamic pitch}} + F_{\text{dynamic heave (at pitch angle)}}$	$F_{\text{static roll}} + F_{\text{dynamic roll}}$ $F_{\text{static pitch}} + F_{\text{dynamic pitch}}$
Due to safety device	$k \cdot F_{\text{static combined}}$ (*k* is the DAF based on safety device)	$k \cdot F_{\text{static vertical}}$

Table 2. Difference between LR and DNV GL.

3.3 Requirements for guide rails

There are no requirements for allowable deflection in DNV GL's standard for Rules for Lifts, other than limiting the deflection to a value that will not affect the normal operation of the car and the counterweight. The requirements in EN 81-1 are often used as guidance due to the diffuse formulation, and they are 5 mm when safety gears are operating or 10 mm without safety gears, which refers to running and loading. ISO 8383 is however used in practice for evaluating the maximal deflection in operating condition, which states that it should not exceed 3 mm. There is no requirement in stowed condition, but 3 mm is used here as well. An external elevator controller must perform guiding tests if greater values are presented in the calculations, and few benefits are therefore obtained due to the time and cost of the controller. However, the new standard from DNV GL will require a maximal deflection of 3 mm for both operating and stowed condition. This will eliminate all doubts and clearly specify what is required. As for LR, they require the same values for permissible deflections as EN81-1, whether the elevator is operating, stowed or in an exceptional condition [H. Jensen, Senior Engineer, DVN GL Lifting Appliances. Personal communication via mail, 06.02-17].

Max. deflection	***DNV GL, existing***	***DNV GL, upcoming***	***LR***
Operating	3 mm	3 mm	10 mm
Stowed	-	3 mm	10 mm
Exceptional	-	-	5 mm

Table 3. Requirements for maximal deflection.

The requirements for allowable bending and buckling stresses are different for the two classification societies. DNV GL have defined two allowable stresses in calculation of buckling. The stresses shall not exceed 140 MPa for guide rails with 370 MPa steel grade, and 210 MPa for guide rails with 520 MPa steel grade. The stress could also be found by linear interpolation for steel graded in between. When calculating according to DNV GL, bending stresses and buckling stresses should be calculated separately. Bending stresses in stowed condition are to be evaluated for the largest bending stress in either roll or pitch direction. LR have on the other hand defined two requirements depending on the stress condition; subjected to pure bending or subjected to both bending and compression. According to LR, roll and pitch are assumed to act simultaneously for both operating and stowed condition. For exceptional condition, only the bending stresses for the static components of roll and pitch are used. All the requirements are listed in Table 4, and the stress factors for each condition are presented in Figure 4.

Source	***Stresses***	***Criteria***
DNV GL	Buckling (= compression)	$\sigma_k \leq \sigma_{all}$
LR	Bending	$\sigma_b \leq f\sigma_y$
	Bending and compression	$\frac{\sigma_b}{\sigma_t} + \frac{\sigma_c}{\sigma_{cr}} \leq f$

Table 4. Requirements on permissible stresses.

Condition	***Operating***	***Stowed***	***Exceptional***
Stress factor, f	0.60	0.75	0.85

Table 5. Stress factors.

4 Ship motions and accelerations

The motion of a ship can be described and measured by the six degrees of freedom with respect to the reference center. All degrees of freedom are defined by the right-hand coordinate system, where positive linear motions are defined as forward, port side and upward direction, and positive rotations are defined in the counter-clockwise direction. With references to Figure 16, the following translations and rotations are used to describe the motions and accelerations of a ship.

- Heave, the linear vertical motion both upwards and downwards.
- Sway, the linear transverse motion either towards port side or starboard.
- Surge, the linear longitudinal motion defined in aft or forward direction.
- Roll, the rotation about the longitudinal axis.
- Pitch, the rotation about the transverse axis.
- Yaw, the rotation about the vertical axis.

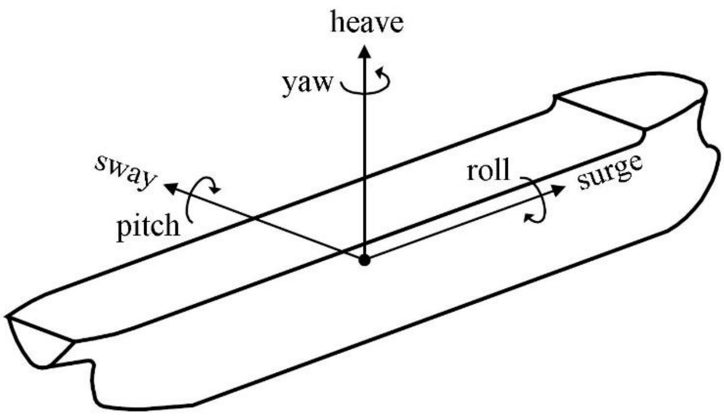

Figure 16. Definition of ship motions [12].

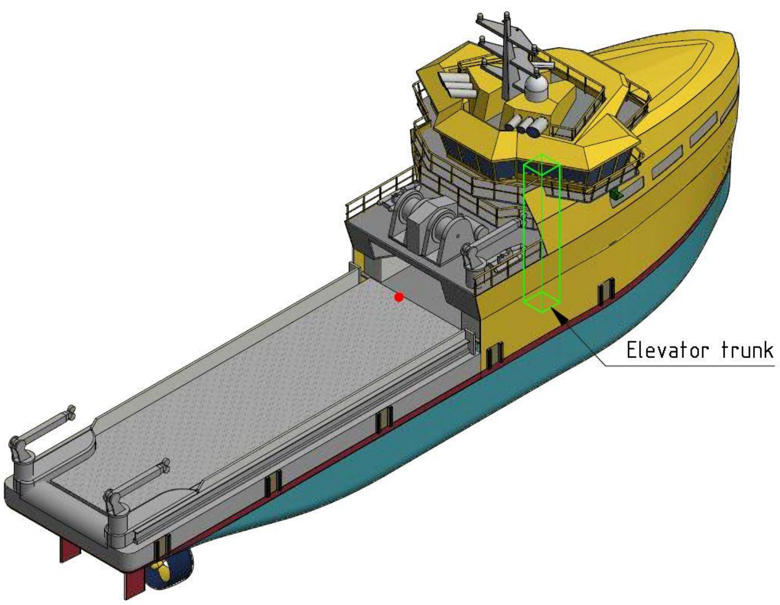

Figure 17. Elevator trunk in relation to reference center [internal document, TK].

4.1 Free harmonic vibration

Figure 18 presents a typical placement of an elevator trunk in a supply vessel. The trunk is illustrated by the green rectangular extending from the waterline to the bridge deck. It will always follow the motion of the ship, and can therefore be calculated with the same environmental conditions. The car is illustrated by the hatch area and located at maximal travel height. The dashed semicircle describes the oscillating path of the car's center of gravity (COG). For simplicity, one could assume that the COG acts in the middle of both guide shoes. It is on the other hand located more towards the floor of the car, due to the mass distribution of the car sling. An estimate is however needed, since the COG changes depending on the elevator type and associated car sling. For instance, a panorama lift with glass interior would give a different COG than a hydraulic lift with steel interior. A more detailed COG could be found using an CAD-software to analyze the properties and mass distribution of the selected elevator type. Furthermore, the distance from the ship's rotational center to the COG is denoted as R, which is used both for roll and pitch calculations. It is hard to estimate the exact location of the rotation center, since the relation between the waterline, the ship's center of gravity (G) and the ship's center of buoyancy (B) changes depending on the load condition. It is therefore calculated with the same formulation as DNV GL uses, which is presented in section 4.2. This gives a value approximately near the ship's center of gravity.

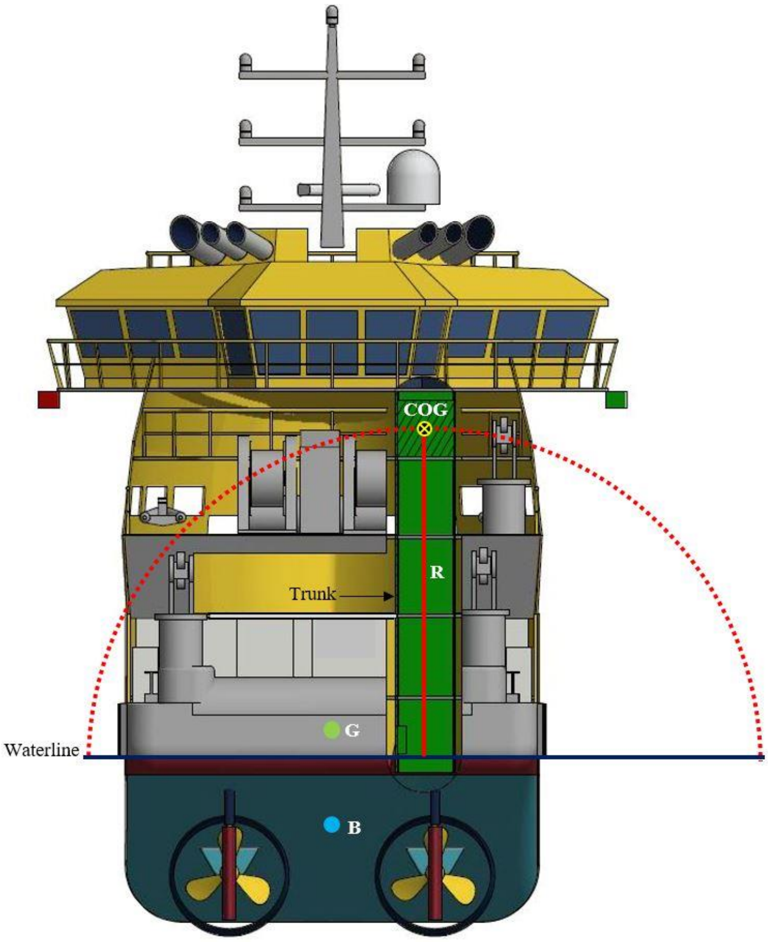

Figure 18. Rear view of a supply vessel [internal document, TK].

The accelerations caused by roll and pitch can be found by assuming the elevator motion as a regularly repeating oscillatory motion, as shown in Figure 19. An object that experiences this periodic motion is said to be harmonic, known in the literature as free harmonic vibration. The motion is assumed to oscillate from positive angle (green position) to negative angle (white position) and back again to positive angle, which is equal to one period. As seen from the dashed semicircle in Figure 18, one could assume that the elevator follows a circular path due to the motion of the ship. The amplitude must therefore be defined by the arc length, and not the linear length [13].

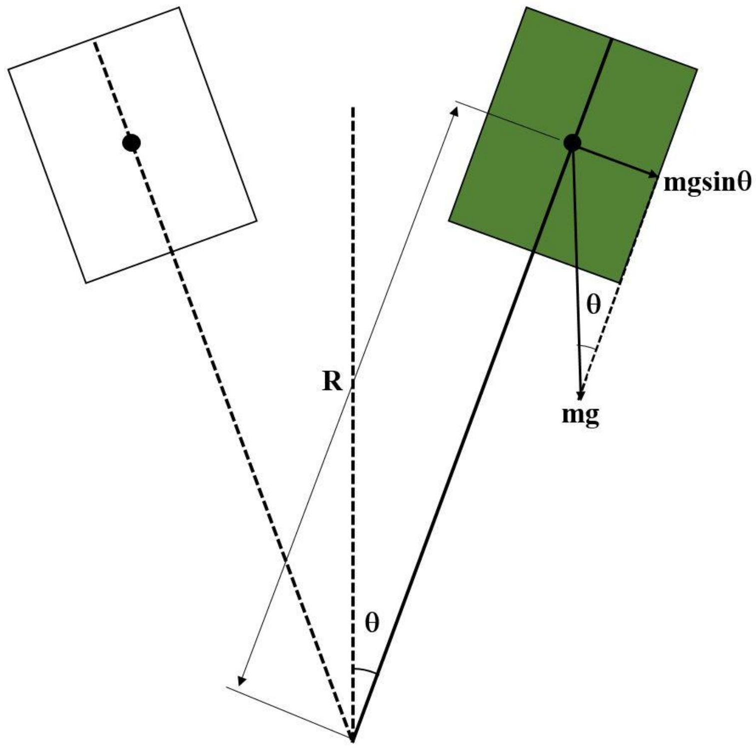

Figure 19. Elevator car in free harmonic vibration.

Once the period, angle and amplitude are known, the following relationships could be used to express the position, velocity and acceleration of the elevator:

$$x(t) = A\,cos(\omega t) \tag{1}$$

$$v(t) = -A\omega\, sin(\omega t) \tag{2}$$

$$a(t) = -A\omega^2\, cos(\omega t) \tag{3}$$

where;

$$A = R\alpha \tag{4}$$

(α is either θ or φ)

$$\omega = 2\pi f \tag{5}$$

$$f = \frac{1}{T} \tag{6}$$

$x(t)$	position as function of time	*m*
$v(t)$	velocity as function of time	*m/s*
$a(t)$	acceleration as function of time	m/s^2
ω	angular speed	$rads^{-1}$
f	frequency	s^{-1}
A	amplitude	*m*
R	distance from the ship's rotation center to COG	*m*

4.2 Accelerations from classification societies

The alternative approach for calculating the ship motions and accelerations is based on rules and procedures defined by the classifications societies. It is presented in both DNV GL's rules for Classification of Ships and LR's Code for Lifting Appliances in a Marine Environment. Both methods are based on the similar approach for calculating the ship motions and accelerations, but only DNV GL's method is presented in this section. The difference between the two is how the hydrodynamic equations are formulated, for instance how the constants are found. Further investigation on the technical background for these equations are not included.

DNV GL's Rules for Classification of Ships is based on IACS Common Structural Rules for Bulk Carriers and Oil Tanker, but the rules are however applicable for all ship types. The North Atlantic wave environment is used to represent the conditions for the ship specific calculations. These conditions are in strength assessment based on extreme operation at sea, associated to loads encountering the ship once in her lifetime. This is the same as a return period of 25 years, which corresponds to an approximate probability of exceedance of 10^{-8}. This is then reduced to a "daily" level, with the aid of reduction factors for calculating the strength related to normal accept criteria [14].

The method calculates ship motions based on dimensions and characteristics, as presented in Table 6. It is also necessary to specify different correction factors, for instance if the ship is built with bilge keels. These factors are specified in Appendix B. The ships accelerations at the center of gravity are calculated once the dimensions and correction factors have been defined.

Symbol	***Meaning***	***Description***	***Unit***
L	Rule length	The distance from AE to FE	*m*
B	Moulded breadth	The greatest breadth measured amidships at the scantling draught	*m*
D	Moulded depth	The vertical distance amidships from BL to main deck	*m*
T_{SC}	Scantling draught	The vertical distance amidships from BL to summer WL at full load condition	*m*
T_{LC}	Midship draught	The vertical distance amidships from BL to summer WL at a considered load condition	*m*
C_B	Block coefficient	The ratio of moulded displacement (underwater volume) of a ship to the product of seawater density, T_{SC}, L and B.	-

Table 6. Ship dimensions and characteristics.

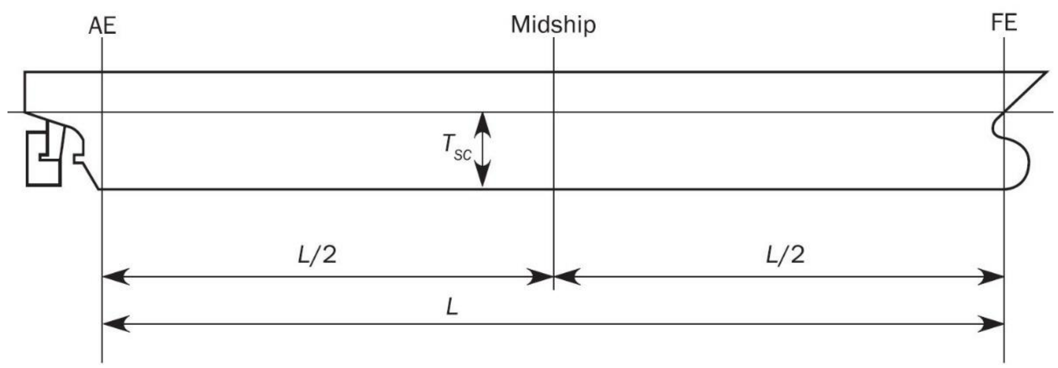

Figure 20. Side view of ship [15].

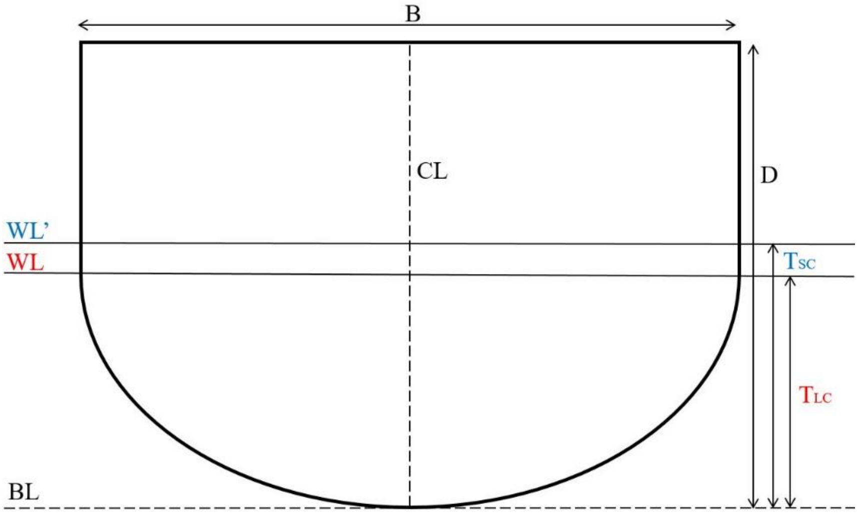

Figure 21. Rear view of ship.

An estimation of the rotation center is needed to perform further calculations. The location of this center varies from ship to ship, since it is influence by the ship dimensions and the load distribution. A suitable estimate is however defined by DNV GL, which is applicable for most ship types. By their definitions, the center is assumed to be 0.45 L from AE, at CL and z from BL. The first and second coordinates are easy to relate to, while the latter coordinate must be calculated using equation (7). According to IACS, the vertical rotational center is assumed to be the smaller of ($D/4 + T_{LC}/2$) and D/2, which results in a value approximately to the ship's vertical center of gravity.

$$z = \min[\frac{D}{4} + \frac{T_{LC}}{2}, \frac{D}{2}] \quad (7)$$

After defining the ship dimensions, the coordinates to the elevator are implemented to calculate the envelope accelerations at this point. Envelope accelerations are often used when maximum design accelerations are required, for example in calculation of machinery foundation strength. They are expressed as the resulting accelerations in longitudinal, transverse and vertical direction. Accelerations from surge and pitch are included in the longitudinal component, while sway and roll are included in the transverse component. As for the vertical component, heave is the most determinant component, but the contributions from roll and pitch are also included. The magnitude of the accelerations will also vary depending on the position relative to the ship, it may either increase, decrease or be constant with the ship's height, breadth or length. The longitudinal acceleration is for instance constant along the ship's length, while the transverse acceleration is constant along the ship's breadth, and the yaw term is therefore neglected in both expressions [14].

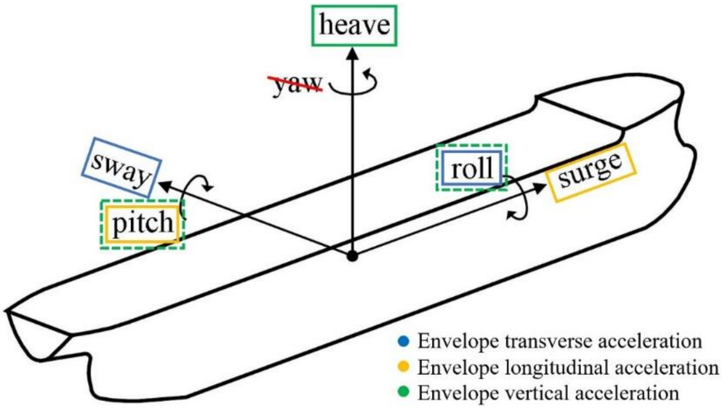

Figure 22. Envelope accelerations.

Figure 23 shows a plot of the roll acceleration, the envelope transverse acceleration and how they both vary with the height. The solid line represents the roll acceleration, while the dashed line represents the envelope transverse acceleration. The direction of the roll acceleration changes from negative to positive at the rotation center, while the envelope is constant in one direction since this includes roll, sway and the static inclination as well. A similar plot would also be obtained for the envelope longitudinal acceleration. In other words, both are influence by the height, and increasing it would result in higher transverse and longitudinal accelerations.

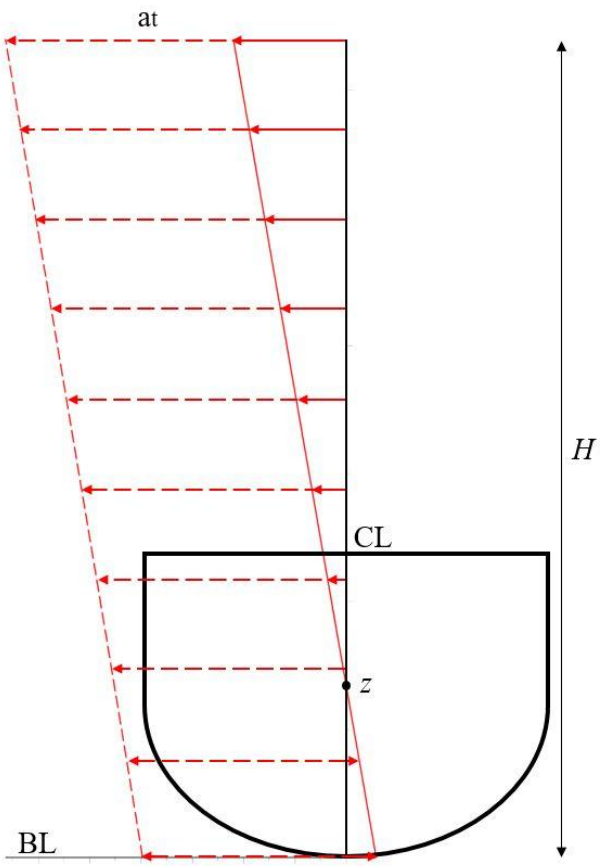

Figure 23. The envelope transverse acceleration (dashed) and roll acceleration (solid).

Figure 24 illustrates the envelope vertical acceleration and how it varies with the length of the ship. The vertical acceleration will also vary with the breadth of the ship, but this has almost no influence on the resulting acceleration. As expected, the acceleration reaches its minimum at 0.45 L from AE, but keep in mind that it is not equal to zero at this point. The maximum value is on the other hand obtained in the FE.

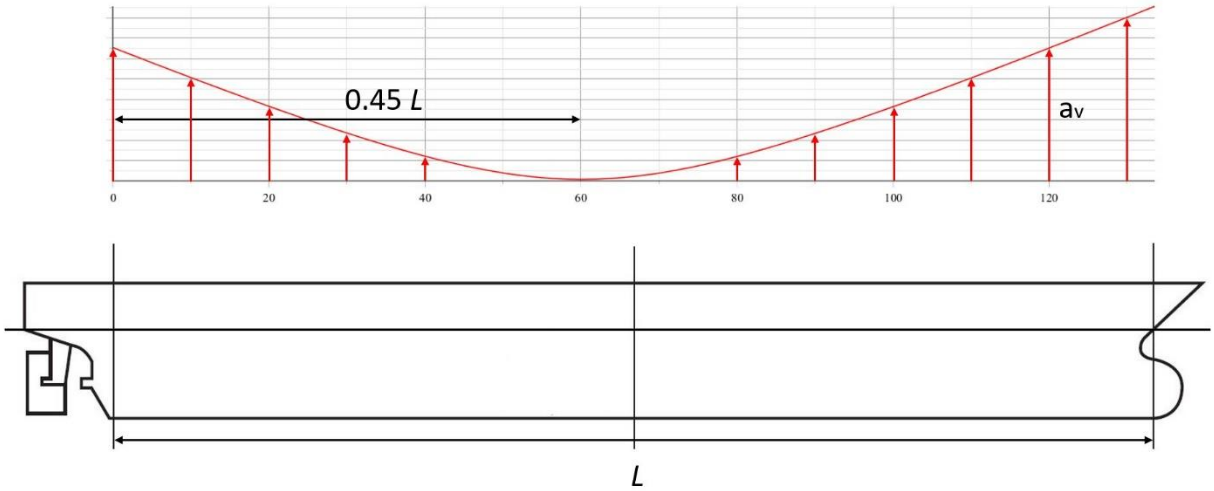

Figure 24. The envelope vertical acceleration at CL.

4.3 Calculation of guide rail strength

When calculating the strength of the guide rails, the forces acting on both axes need to be taken into consideration. The direction of the guide rail axes will vary depending on the orientation of the elevator, as shown in Figure 25. If the elevators are being installed with transverse oriented guide rails, they would in a roll motion distribute the force on the faces of two guide shoes and then further onto one guide rail. However, if the elevators are being installed in the opposite direction, the force would be distributed on the faces of four guide shoes and then further onto two guide rails. The worst case when calculating according to DNV GL, is if they are installed in the transverse direction, since the force from roll is of greater magnitude, and because of the number of guide shoes. As for LR, all cases must be checked.

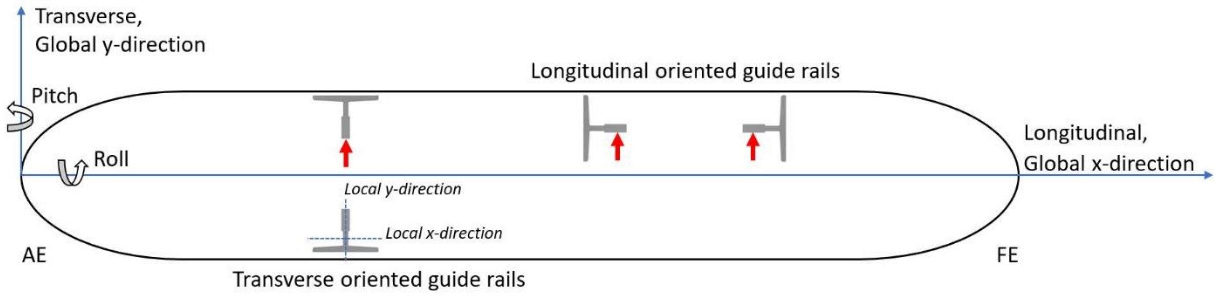

Figure 25. Guide rail orientations.

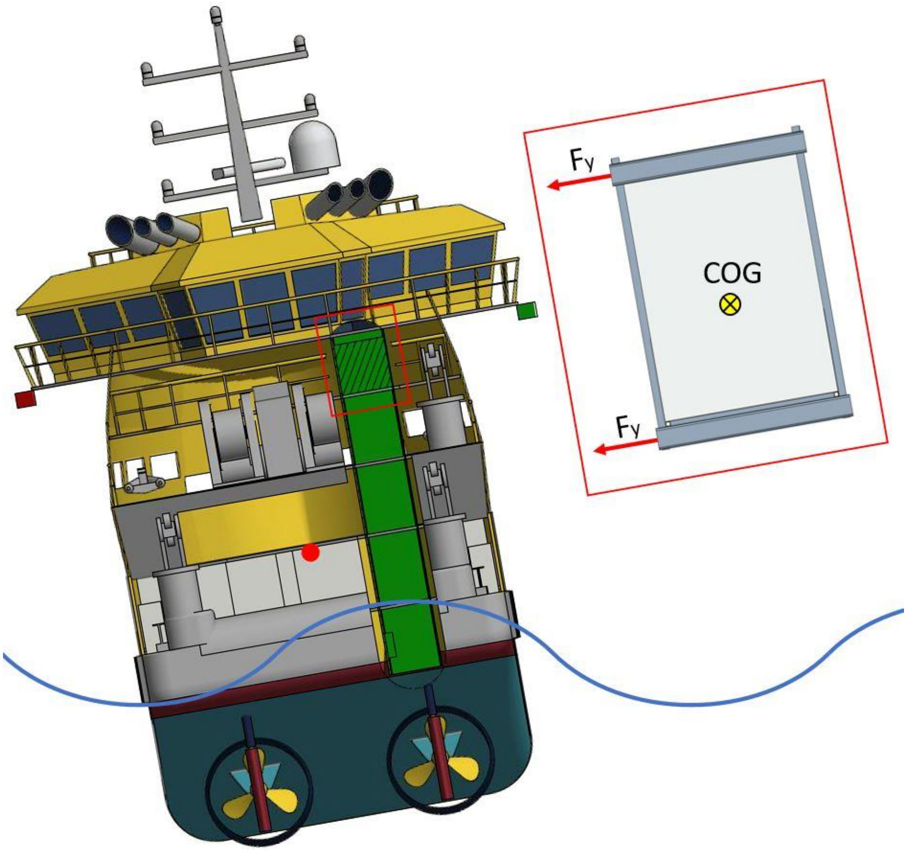

Figure 26. Load distribution from roll motion and transverse oriented guide rails.

An example of a car sling with associated guide shoes, assembled in the upper and lower part of the sling, is shown in Figure 27. The car, which is represented by the transparent box, is fixed inside of the car sling and it is assumed that they will both act as one mass. The force from the car and car sling will firstly be transmitted to the guide shoes, and then to the guide rail as shown in Figure 28.

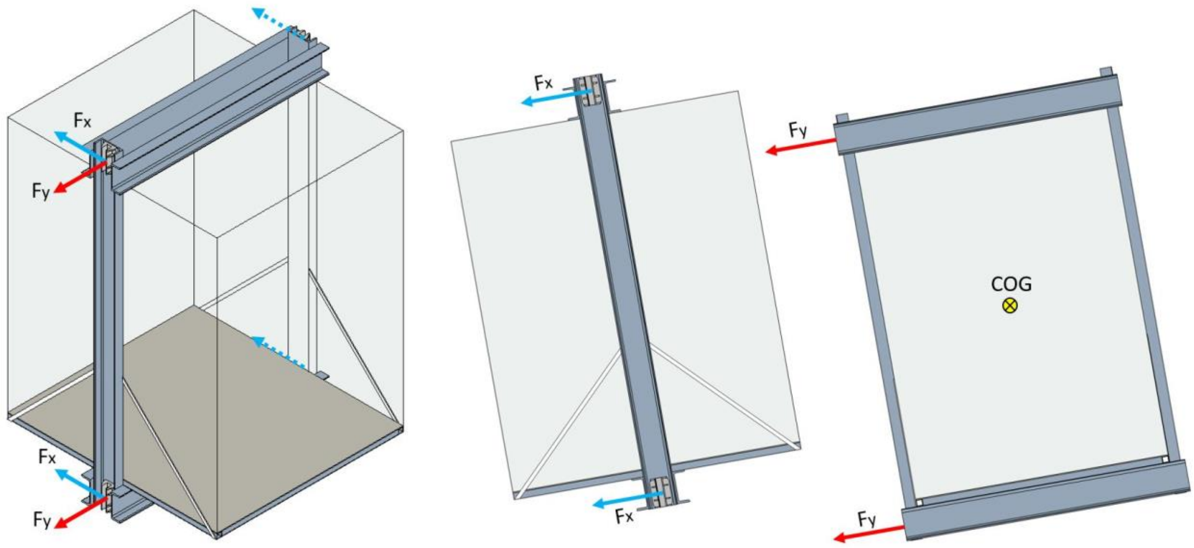

Figure 27. Load distribution from car and car sling.

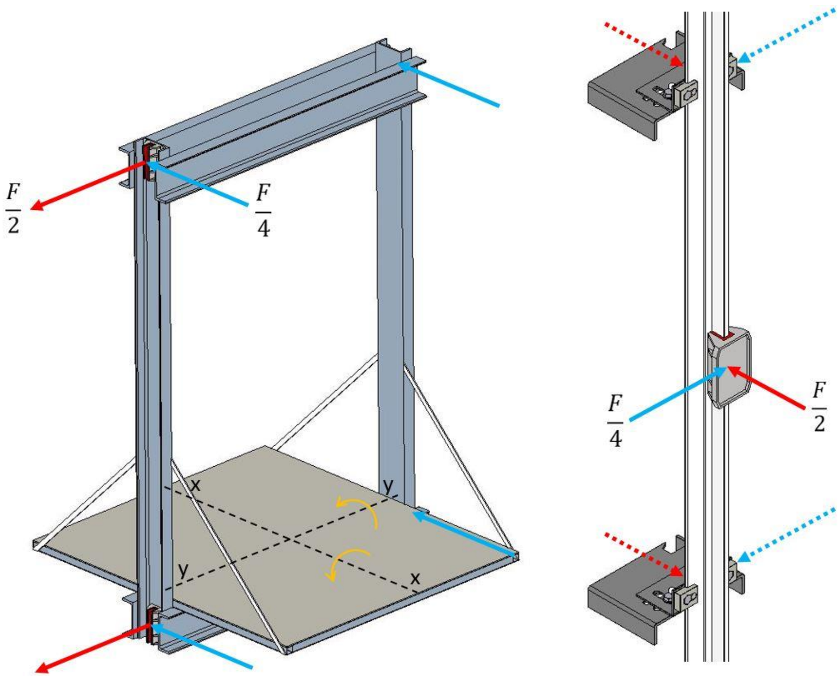

Figure 28. Load distribution from car sling to guide rail.

Figure 29 shows how the forces in both directions are transmitted from the guide shoe to the guide rail. The force F_y will bend the guide rail about the x-axis, which is the strong axis, while the force F_x will bend about the y-axis, which is the weak axis. This applies to all T-profiles except T125-L1/A, T125B and T127-2/B guide rails. These guide rails have more strength about the y-axis.

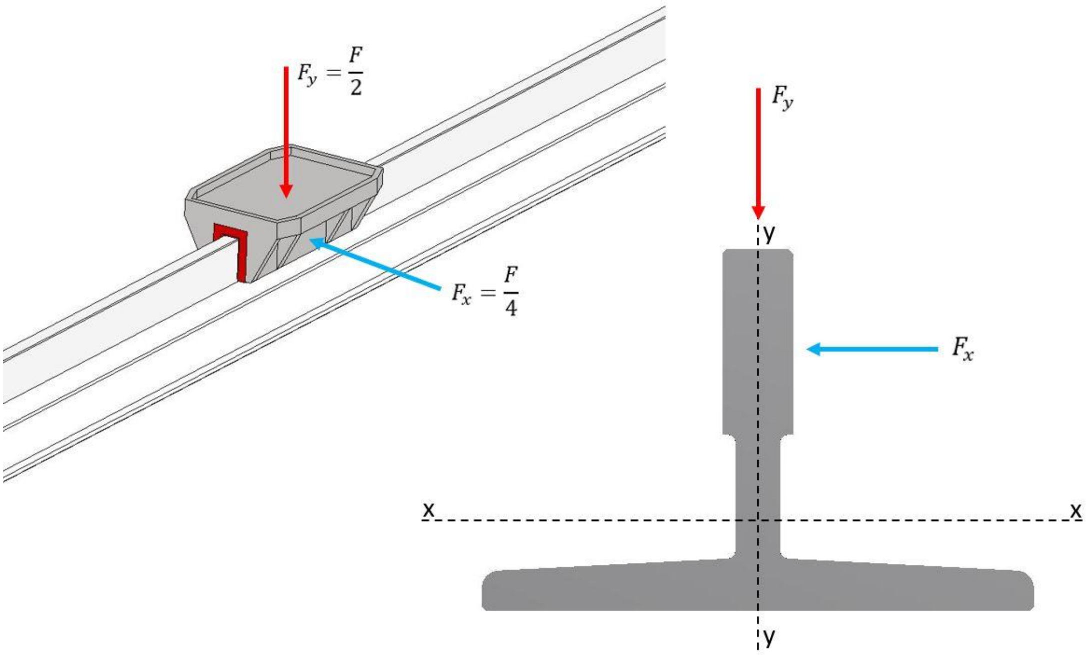

Figure 29. Load distribution from guide shoe to guide rail.

The forces, in both transverse and longitudinal direction, can be found by summing the static inclination together with the acceleration resulting from the ship motions, and then multiplying with the total weight acting in the COG as defined in equation (8).

$$F = m(g + a) \tag{8}$$

where;

$$g = g_0 \sin\alpha \tag{9}$$

(α is either θ or φ)

$F_{x/y}$	transverse or longitudinal force in the evaluated direction	N
m	total weight of component	kg

The bending moment, bending stress and deflection can be found by assuming the guide rail as a simply supported beam, with a point load acting between both brackets as illustrated in Figure 30. One could discuss if the brackets are of fixed support due to the end constraint condition. However, both DNV GL and LR assumes it as a simply supported beam, and it is therefore calculated according to this.

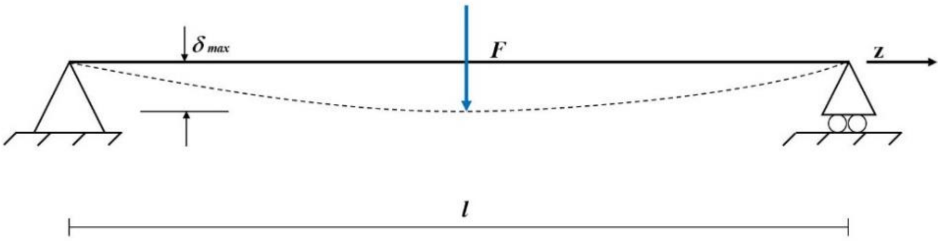

Figure 30. Simply supported beam.

$$M_b = \frac{Fl}{4} \tag{10}$$

$$\sigma_b = \frac{M_b}{W} \tag{11}$$

$$\delta_{max} = \frac{Fl^3}{48EI} \tag{12}$$

M_b	bending moment about the evaluated axis	*Nmm*
l	distance between each bracket	*mm*
σ_b	bending stress about the evaluated axis	*MPa*
δ_{max}	maximal deflection	*mm*
W	section modulus about the evaluated axis	mm^3
I	second moment of inertia about the evaluated axis	mm^4

With the aid of the buckling factors given in DNV GL's standard for Rules for Lifts, the buckling stress in each guide rail may be calculated using equation (13). The buckling factor is dependent on the coefficient of slenderness, which again is based on the material properties of the guide rail.

$$\sigma_k = \frac{F_z\,\omega}{A} \tag{13}$$

$$\lambda = \frac{l_k}{i} \rightarrow \omega(\lambda) \tag{14}$$

$$i = \sqrt{\frac{I}{A}} \tag{15}$$

where;

$$F_z = k(P+Q) \tag{16}$$

- If progressive safety gears are used k = 10
- If instantaneous captive roller safety gears are used k = 15
- If instantaneous safety gears are used k = 25

The vertical force is calculated based on the rated load and the weight of the car plus car sling. A safety gear coefficient is also added to account for the type of safety device. This coefficient is based on a DAF of either 2, 3 or 5, the gravitational acceleration 9.81 m/s^2 and the number of guide rails the force is divided on.

σ_k	buckling stress	*MPa*
F_z	axial force in vertical direction	*N*
ω	buckling factor	-
λ	coefficient of slenderness	-
l_k	effective buckling length	*mm*
k	safety gear coefficient	m/s^2
P	weight of car plus car sling	*kg*
Q	rated load	*kg*

LR calculates their buckling stress in a different way, by combining it with the bending stresses as well. The buckling stress is found using the *Perry Robertson formula*, with the aid of an end constraint condition and the Robertson constant. For further reading, it is referred to *Ch. 4 Sec. 2.18* in LR's Code for Lifting Appliances in a Marine Environment [10].

4.4 Guide rail specifications

Table 7 shows the relation between weight and cost for the guide rails that TK uses the most. Per guide is equal to 5 meters of guide rail and cost per guide is calculated with the 30 % discount, which is normally included in the quotation from their supplier. Cost for accessories are also included, which includes the fishplate with mountings needed to connect each guide rail. Seen from an installation perspective, one could characterize T50 and T70 as the light, T89 and T90 as the medium, and T125 and T127 as the heavy guide rails. Heavy guide rails would require more resources both in terms of transportation and installation. It is however the transportation from the shipyard to the trunk that consumes the most time. If the guide rails are too heavy, a crane is usually required to lift them aboard the ship. It is also easier to transport lighter guide rails through the corridor and into the trunk. As for the installation, there is just a small difference in time spent on the various guide rails.

Type	***Code***	***Weight per guide***	***Cost per guide***
Cold drawn	T50/A	18.7 kg	33 EUR
Cold drawn	T70/A	36.9 kg	68 EUR
Machined	T89/B	61.9 kg	100 EUR
Machined	T90/B	67.7 kg	115 EUR
Machined	T125/B	89.6 kg	143 EUR
Machined	T127-2/B	112.8 kg	189 EUR

Table 7. Guide rail specifications [16], [internal document, TK].

5 Research and development

This chapter presents the research and development phase of the thesis. Section 1 presents the different ship classes with examples on ship dimensions from each class. Four calculation methods are presented in section 2, applicable for both DNV GL and LR, with the complete procedures given in Appendix A-C. Furthermore, examples of guide rail dimensions on existing offshore projects are then presented in section 3. Section 4 presents the assessment of the two different bracket solutions, while section 5 describes the different guide structure combinations with estimated cost of installation. Lastly, the FEM analysis of the customized bracket solution is then presented, including a description of how the modelling is defined. The complete procedure and results for both bracket solutions are given in Appendix D-E.

5.1 Ship classes

TK delivers elevators to ships independent of dimensions, height of trunk or number of elevators. A mapping of different load cases is therefore developed to evaluate the forces resulting from the motion of the ship. Both transverse and longitudinal forces are included to evaluated their impact on the guide structure. The purpose is to see if similar forces are obtained in each class, such that it is possible to use the same guide structure for all similar projects, or if the forces vary too much and new load cases must be calculated in the beginning of each project. The load cases are divided into three classes, where class 1 denotes the ships with the largest dimensions and highest elevator travel, but also with least ship motions due to their stability when subjected to the environmental conditions. Class 3 represents the ships with smallest dimensions and elevator travel, but with most ship motions and accelerations. Lastly, class 2 represent a combination with respect to both ship dimensions and motions. How the ship classes are defined, with examples on typical ship dimensions for each class, are presented in Table 8 and Table 9.

Class	***Type of ship or vessel***	***Examples***
1	Cruise	Oasis of the Sea, Mein Schiff, Viking Line
2	Supply, service, construction	Skandi Açu, Deep Explorer
3	Expedition	Hurtigruten, Vard Ponant, Hapag Lloyd, LMG 200-PC

Table 8. Ship classes.

Ship dimensions	***Class 1***	***Class 2***	***Class 3***
Ship	Mein Schiff	Skandi Açu	Hurtigruten
Length over all, L_{OA}	293.2 m	145.9 m	140 m
Moulded breadth, B	35.8 m	30 m	23.6 m
Moulded depth, D	11.2 m	13 m	16 m
Design draught, T_{LC}	8.05 m	8.5 m	5.5 m

Table 9. Example of ship dimensions for each class.

Figure 31. Class 1, Mein Schiff 4 [17].

Figure 32. Class 2, Skandi Açu [18].

Figure 33. Class 3, Hurtigruten [19].

5.2 Calculation methods

Four different methods can be used to calculate the forces acting on the guide rails and brackets, where two of them are applicable for DNV GL and the two others for LR. The first method is based on the theory of free harmonic vibration, which is the same procedure as DNV GL uses in the verification of the guide rail calculations. It is also possible to calculate other periods for roll and pitch in stowed condition, given that it is documented from the shipyard or the Ship Structures department of DNV GL. An alternative method that combines their requirements with calculated periods is therefore possible. It would however not represent the actual motions of the ship, and is therefore neglected from the thesis. The second method has not previously been presented to the Lifting Appliances department, and need to be taken into consideration when this happens. The method is however widely known and used in the Ship Structures department, and therefore reliable in terms of strength calculations. Method 1 requires the distance in z-direction (denoted as R in method 1), while the other methods require the coordinates in x- and y-direction as well. Figure 34 shows how the coordinates are defined with respect to AE, CL and BL.

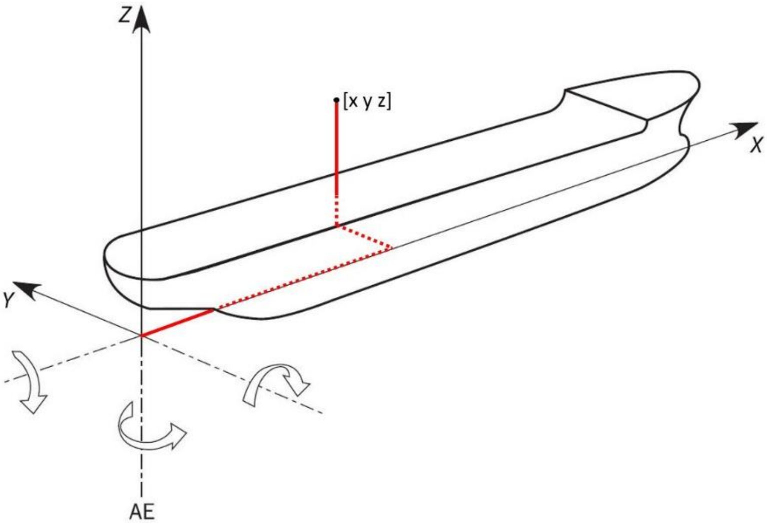

Figure 34. Arbitrary location of elevator.

Method 1: DNV GL Rules for Lifts, 2008 + Free harmonic vibration

1) Calculate transverse and longitudinal accelerations based on requirements for periods and angles, and the distance from the ship's rotation center to COG.

$$\theta, T_\theta, \varphi, T_\varphi, R \rightarrow a_t, a_l$$

2) Combine the accelerations with the static inclination of the ship, and then calculate the resulting forces acting in transverse and longitudinal direction.

$$a_t, a_l, gsin\theta, gsin\varphi \rightarrow F_t, F_l$$

Method 2: DNV GL Classification of Ships, Jan 2017 + Jan 2016 [20]

1) Calculate the ship accelerations at the COG based on ship dimensions and characteristics.

$$L, B, D, T_{SC}, T_{LC}, C_B \rightarrow a_{surge}, a_{sway}, a_{heave}, a_{roll}, a_{pitch}$$

2) Calculate the envelope accelerations based on the coordinates to the elevator.

$$a_{surge}, a_{sway}, a_{heave}, a_{roll}, a_{pitch}, x, y, z \rightarrow a_x, a_y, a_z$$

3) Calculate the resulting forces in transverse and longitudinal direction based on load combinations.

$$(0.67)a_y, (0.67)a_x \rightarrow F_t, F_l$$

Method 3: LR Code for Lifting Appliances, Requirements on ship motions from standard, 2016

1) Calculate the static and dynamic forces, both normal and parallel to deck in transverse and longitudinal direction, based on requirements for periods and angles, and the coordinates to the elevator.

$$\theta, T_\theta, \varphi, T_\varphi, x, y, Z_r, Z_p \rightarrow F_{static}, F_{dynamic}$$

2) Combine the static and dynamic force into three load combinations; roll motion only, pitch motion only and combined motion.

$$F_{static}, F_{dynamic} \rightarrow F_{roll}, F_{pitch}, F_{combined}$$

Method 4: LR Code for Lifting Appliances, Other ship motions defined by standard, 2016

1) Calculate other angles and periods, including the heave amplitude and period, based on ship dimensions and characteristics.

$$L, L_{pp}, B, GM \rightarrow \theta, T_\theta, \varphi, T_\varphi, A, T_h$$

2) Calculate and combine the static and dynamic forces into three load combinations, as described in method 3.

The procedure of calculating the guide rails is presented in Figure 35. Firstly, it is necessary to know which society the ship is being classed according to. Further, it is then possible to calculate with or without ship specific data, as illustrated by the yellow boxes. The customer is usually interested in elevator capacity and what TK can deliver with the given trunk area they are offered. At this stage, guide rail dimensions and distances between brackets could be calculated according to EN81-20/50. These standards calculate normal operation and safety gear operation based on weight distribution, dimensions of the car and placement of guide rails. It assumes both the rated load and self-weight to act at a given distance away from the center of the elevator. This tilts the car and the car sling with respect to the neutral axis, which introduces forces normal to the guide rails. Furthermore, the four methods could then be used to do ship specific calculations, based on additional information on the location of the elevator and the ship dimensions. The output from the offshore calculations should be used in the verification, but it is also interesting to compare both outputs.

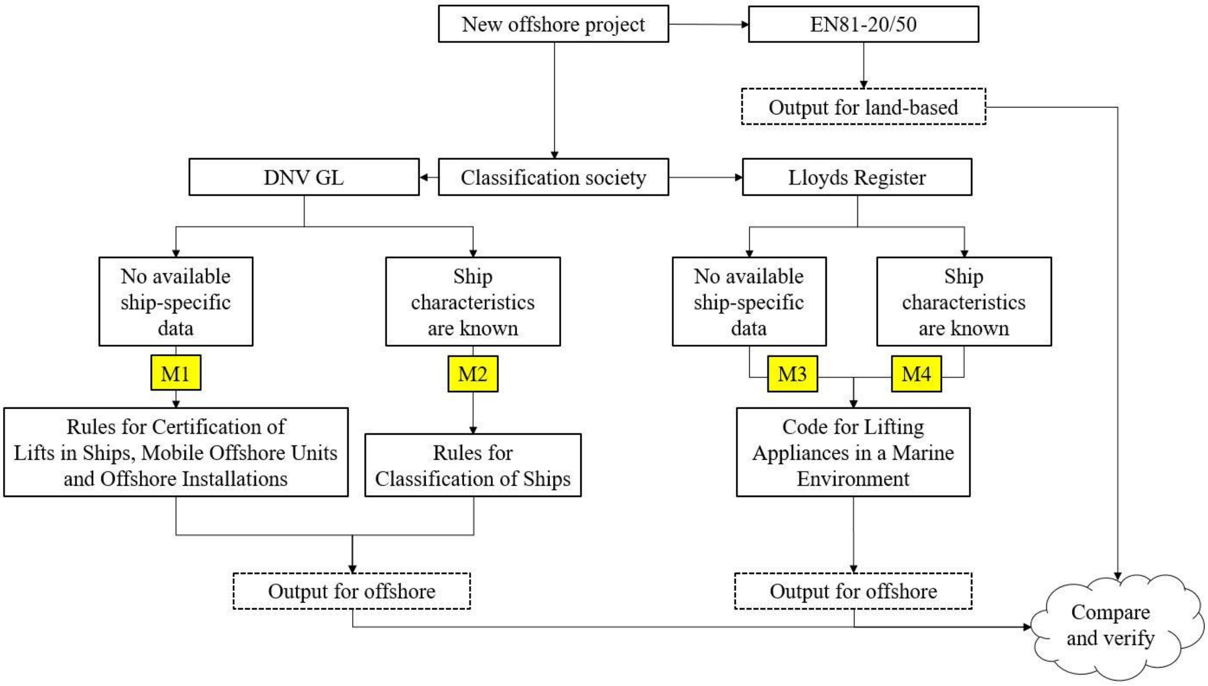

Figure 35. Procedure for calculating the guide rail dimensions.

Required information and input	***M1***	***M2***	***M3***	***M4***
Elevator weight, capacity and % counterweight	✓	✓	✓	✓
Orientation of guide rails	✓	✓	✓	✓
Vertical distance from BL to rotation center	✓	✓	✓	✓
Vertical distance from rotation center to COG	✓	✓	✓	✓
Longitudinal distance from AE to COG		✓		✓
Transverse distance from CL to COG		✓		✓
Ship data (L, B, C_B, etc.)		✓		✓

Table 10. Required information for the four methods.

5.3 Examples from existing projects

Table 11 presents two upcoming offshore projects; the expedition vessel Hurtigruten (Roald Amundsen) and a yacht from Ulstein (TBN, herby referred to as "Ulstein 307"). Hurtigruten is classed according to DNV GL, and the guide rail dimensions and bracket spacing are already decided on this project. Ulstein 307 is classed according to LR, but the dimensions have not been decided on this project yet.

Projects		***Hurtigruten***	***Ulstein 307***
Key figures	Rule length	133.5 m	78.9 m
	Moulded breadth	23.6 m	16.3 m
	Moulded depth	16 m	7.5 m
	Design draught	5.5 m	4.6 m
	Block coefficient	-	-
Evaluated elevator	Traction lift	Passenger lift	Passenger lift
	Capacity	1000 kg	900 kg
	Car size	1100 x 2100 mm	1300 x 1650 mm
	Travel height	23 m	9.3 m
	Distance in x-direction	55.8 m	53.4 m
	Distance in y-direction	2.5 m	0 m
	Distance in z-direction	32 m	21.3 m
Planned to be used	Guide rails	T127-2/B	T127-2/B
	Bracket spacing	1500 mm	1500 mm
Cost and weight	Guide rails with acc.	1701 EUR	1134 EUR
	Weight of guide rails	1015 kg	677 kg

Table 11. Offshore elevator projects.

5.4 Elevator car brackets

An assessment of two different brackets is presented in this section. The first design is a customized solution applicable for offshore steel trunks, while the second is designed for land-based concrete or glass trunks. It is delivered in standard dimensions, together with the rest of the elevator system from TKA’s factory in Neuhausen. For further presentation, they are hereby referred to as:

- Customized bracket solution
- Standard bracket solution

Most of the land-based concrete trunks consists of plane surfaces, which makes it easier to standardize a specific bracket due to the similarities of the trunks. A different design is needed for trunks installed in ships and other similar constructions, because of the bulkhead stiffeners on the inside and the transition between every deck. Two neighboring trunks with different layout are presented in Figure 36. The trunk to the left is built with stiffeners on the outside, while the trunk to the right with the stiffeners on the inside. This is the case for all neighboring trunks, and it is not possible to build both with stiffeners on the outside. It is however possible for single trunks, built next to a corridor for instance. TK usually designs for trunks with the stiffeners on the inside, since it is common for ship engineers to put the insulation between the stiffeners inside of the trunk, instead of the corridor. However, the former TK is involved in the planning phase, the more influence they will have. A plane trunk layout is always the preferable layout, both in terms of design and installation. One should therefore aim to be involved as early as possible, since this would save a lot of time and cost in the later stages.

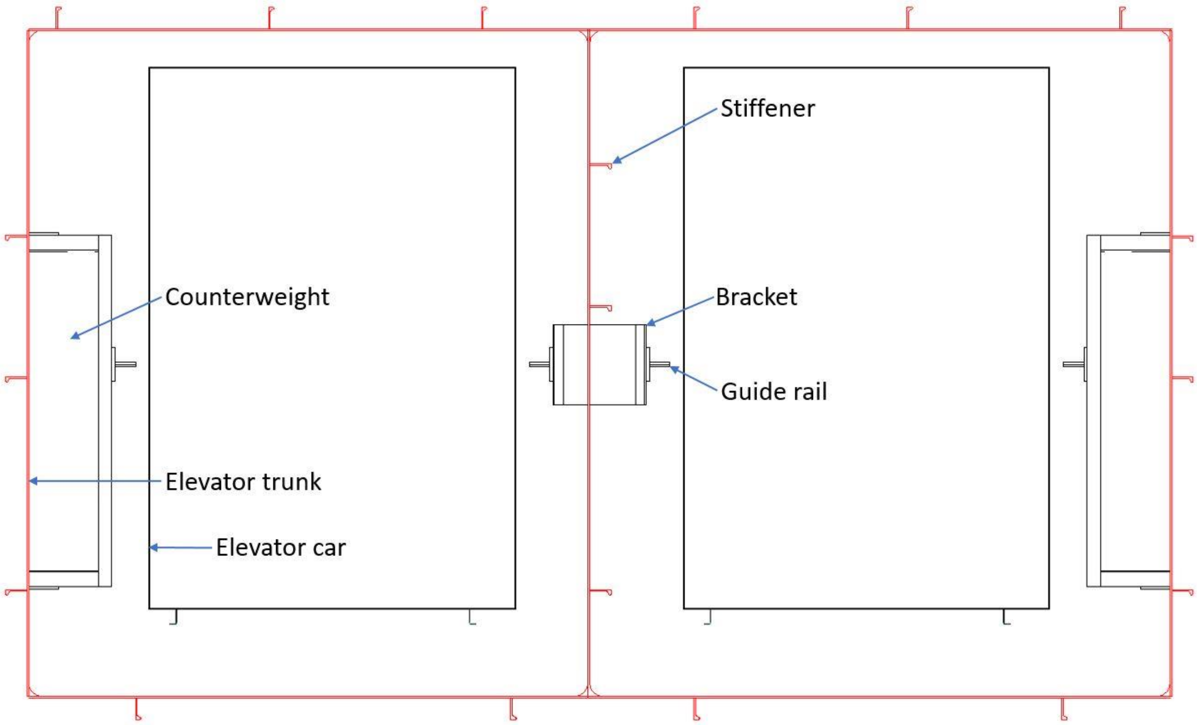

Figure 36. Two different elevator trunks.

Customized bracket solution

The customized solution, shown in Figure 37, is developed by TK and used in all their offshore trunks. It consists of two structural components with associated hex bolts, washers, nuts and rail clips to constrain the guide rail to the trunk. The U-channel is welded in the rear end, and the L-profile is then bolted on top of the channel. The dimensions of the U-channel vary depending on the trunk layout and the elevator weight. A cut out in the rear part of the channel is needed if the placement of the guide rails interferes with the stiffeners. Customization is also needed for the cut out, since the dimensions and the location of the stiffeners varies from ship to ship. This is shown with the alternative cut out in Figure 38.

The possibility for sideways adjustments is also an attractive feature for the brackets. There are two reasons for this. Firstly, because of the difference in guide rail quality. Guide rails in the upper price range are usually of high material quality with good tolerances, while guide rails in the lower price range are of minor quality with large misalignments between the ends. Secondly, because of the welding of the brackets in the installation phase. TK have experience large deviations between the placement in the drawings and the placement after the shipyard have welded them on. It is therefore desirable to have the possibility for corrective measures.

Furthermore, a bracket with the following specifications is chosen to evaluate the customized solution:

- Approximate 7 kg with accessories
- U-channel: 8 mm plate thickness, 300 x 210 x 60 mm (width x depth x height)
- L-profile: 6 mm plate thickness, 180 x 130 x 80 mm
- ± 70 mm sideways travel with respect to neutral position
- About 60 mm outwards travel with respect to neutral position
  (however, recommended to be installed as Figure 37)

Strengths	*Drawbacks*
▪ Structural capacity ▪ Few parts ▪ Applicable for all trunk layouts	▪ Installation time ▪ Possibility for large sideways adjustments ▪ Applicable for one guide rail dimension

Table 12. Customized bracket characteristics.

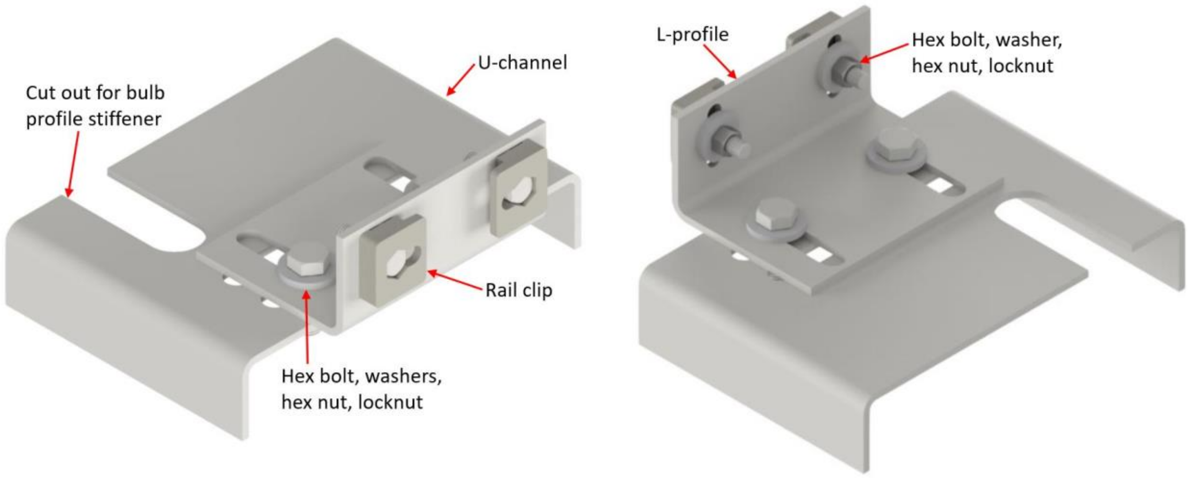

Figure 37. Customized bracket design.

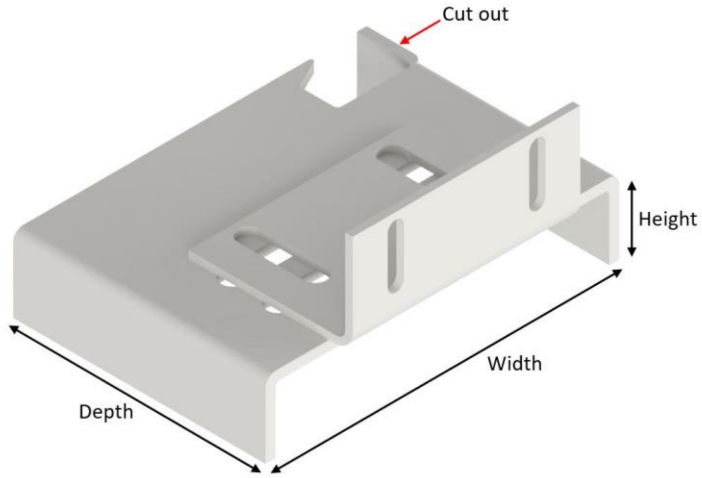

Figure 38. Alternative cut out for U-channel.

Standard bracket solution

TKA delivers a large variety of standardized brackets for cars. They come in different sizes and shapes, because of the weight and the distance from the car to the trunk. *AY adjustable bracket 80-230 (mat. no. 6011 0000743)* is the most common standardized bracket used by TK on their land-based elevators. The solution is originally designed for wall plug mounting or anchor rail mounting, which is the same as a Halfen channel. It consists of three structural components with associated bolts, washers, nuts and rail clips to constrain the guide rail. Two L-profiles of different sizes are constrained to a Halfen channel, where the largest profile is assembled on the Halfen channel due to the required strength. The Halfen channel is welded to the trunk along the upper and lower part of the channel, and the profiles are then bolted together to constrain the guide rail to the trunk. Halfen bolts are used to constrain the lower bracket, while hex bolts are used to constrain the upper bracket. The holes in the upper bracket are applicable for the following guide rail dimensions; T127-2/B, T125/B, T90/B, T70-1/A and T50/A. This solution requires either a plane surface or the possibility to install the brackets between two stiffeners, which again depends on the placement of the guide rails.

The *AY adjustable bracket 80-230* has the following specification:

- Approximate 6 kg with Halfen channel and accessories
- Upper L-profile: 4 mm plate thickness, 210 x 130 x 80 mm
- Lower L-profile: 6 mm plate thickness, 210 x 130 x 80 mm
- ± 195 mm sideways travel with respect to neutral position (due to Halfen channel)
- Approx. 50 mm outwards travel with respect to neutral position
  (however, recommended to be installed as Figure 40)

Strengths	***Drawbacks***
▪ Installation time ▪ Applicable for several guide rail dimensions ▪ Possibility for large sideways adjustments	▪ Less strength capacity ▪ More parts ▪ Dependent on trunk layout

Table 13. Standard bracket characteristics.

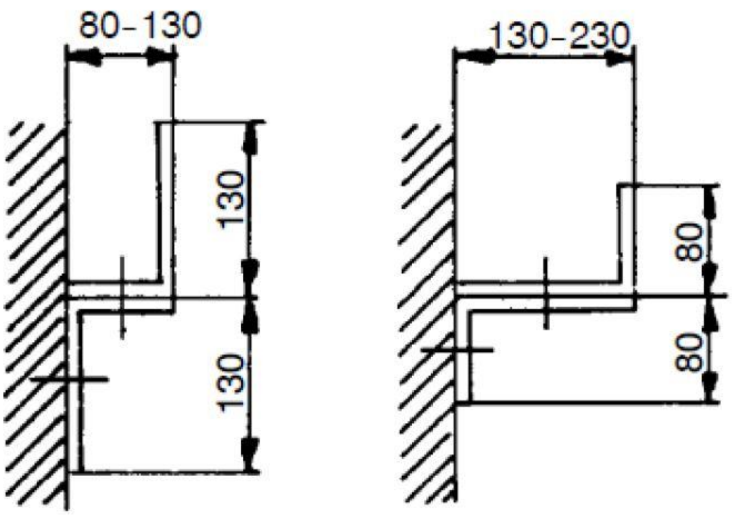

Figure 39. AY adjustable bracket 80-230 [internal document, TK].

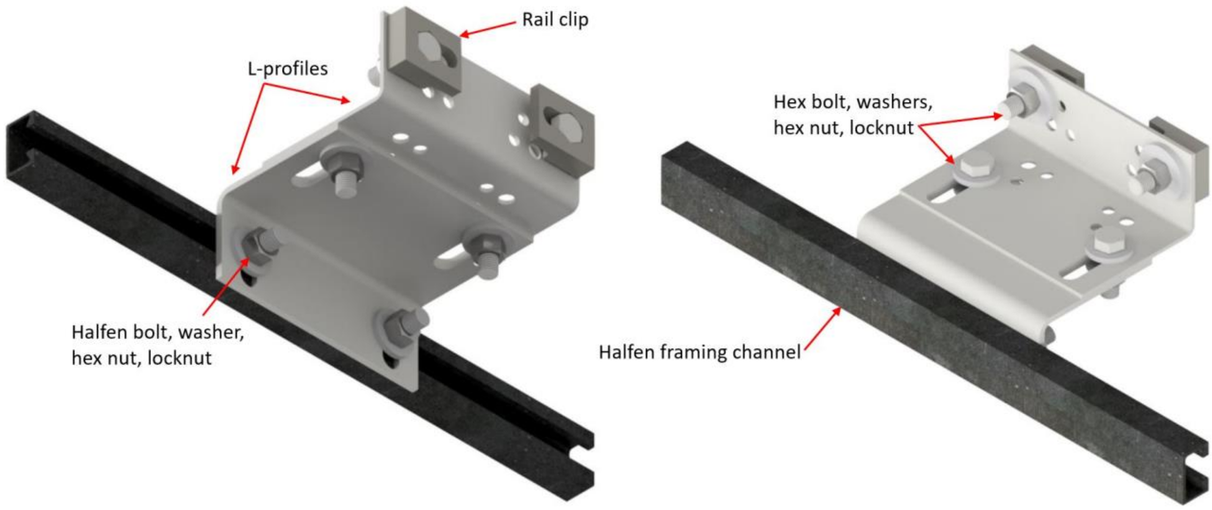

Figure 40. Standard bracket design.

5.5 Estimated cost of installation

Four different cases of installation are presented in Table 14. It is assumed the installation of car guide rails and brackets inside a trunk of 20 meters height. This height is taken as the approximate height out of five different expeditions vessels. Furthermore, it is assumed that the brackets are assembled directly on both sides of the trunk. As for the guide rails, T127 versus T90 is chosen for comparison, since this represents what TK uses now and what they could potentially be using in the future projects. The standard land-based and the customized offshore solution are chosen for comparing the difference in bracket installation. It is also assumed that the trunk is built with stiffeners on the outside, which makes it possible to use both solutions. The guide rails and brackets are then combined into four different cases, where base case 1 represents TK's current guide structure.

Base case	***1***	***2***	***3***	***4***
Approx. trunk height	20 m	20 m	20 m	20 m
Number of guide rails	8 pc	8 pc	8 pc	8 pc
Guide rail dimensions	T127-2/B	T127-2/B	T90/B	T90/B
Weight of guide rails	902 kg	902 kg	542 kg	542 kg
Bracket type	Customized	Standard	Customized	Standard
Bracket distance	1500 mm	1500 mm	1800 mm	1800 mm
Number of brackets	26 pc	26 pc	20 pc	20 pc
Weight of brackets	182 kg	156 kg	140 kg	120 kg
Installation				
Time required per trunk	50 h	35 h	44 h	30 h
Number of installers	2 pers.	2 pers.	2 pers.	2 pers.
Hourly rate per installer	75 EUR/h	75 EUR/h	75 EUR/h	75 EUR/h
Total installation cost	7500 EUR	5250 EUR	6600 EUR	4500 EUR

Table 14. Comparison in cost of installation [internal document, TK].

The required time, number of installers and hourly rate are based on both experience and documentation from TK. An installation team of two persons is usually what is required, and the hourly rate of 75 EUR/h (700 NOK/h) per installer is the fixed salary for each project. However, the number of hours required per trunk vary depending on the trunk layout, transportation and availability of components. This includes the transportation from the shipyard to the trunk, unpacking of the components and installation of the complete guide structure for the elevator car. There are no available data on this, but an estimate based on experience, assumptions and the combined knowledge from the engineers and the installers. TK's installation on earlier projects are used to estimate the hours for the T127 and T90 guide rails, as well as the customized brackets, while assumptions were needed for the standard bracket since this solution has never been used before.

5.6 FEM analysis of brackets

The FEM analysis involves four main steps; modelling the problem, selecting element type and size, apply loads and determine the boundary conditions. These steps must be defined as correctly as possible to achieve reliable results. Three different load cases are considered for each bracket. Load case 1 and 2 models the elevator in worst case at maximal travel height, with the force acting either directly on one bracket or between two brackets. Load case 3 models the elevator in exceptional condition when the safety gear is activated. The specifications in Table 15 are used to model and present the analysis in ANSYS.

Components	***Details***
Guide rail type	T90/B, including model of guide shoe
Length of guide rails	3100-4600 mm, depending on model
Guide brackets	Both solutions, but customized is presented in this section The analysis of the customized bracket is done with a plate thickness of 6 mm (originally 8 mm)
Distance between brackets	1500 mm
Safety gear	Instantaneous safety device, installed on both guide rails Captive roller type, k = 15

Table 15. Specifications for the FEM analysis.

Modelling the problem

The guide rails are usually welded to baseplates in the pit floor by the shipyard. This is done to ensure that most of the vertical force is absorbed in the hull structure of the ship, and not the brackets. The purpose of the brackets is therefore to withstand forces in transverse and longitudinal direction due to roll and pitch. However, each bracket is constrained by rail clips and will displace vertically before the sliding occurs. Sliding will take place between the surfaces of the bracket, guide rail and rail clips when the axial force is greater than the frictional force. This axial force is defined as the maximal force of one clamp pair prior to sliding, and it is equal to 5560 N for T90/B rail clips [internal document, TK]. It is therefore interesting to evaluate the stresses prior to sliding in safety gear operation as well.

The simplified FEM model for load case 1 is shown in Figure 41, which illustrates the elevator at the highest deck. It consists of a guide rail of about 3100 mm length, connected to three brackets with equal spacing of 1500 mm. The force is assumed to act directly on one bracket, which is distributed on either two or four guide shoes depending on the orientation of the bracket. The simplified FEM model for load case 2 is shown in Figure 42, which illustrates the elevator at a position 750 mm below top deck. It consists of a guide rail of about 4600 mm length, connected to four brackets with equal spacing of 1500 mm. The force is assumed to act between the two brackets, which divides the force, but also introduces a different bending moment compared to load case 1. The simplified FEM model for load case 3 is shown in Figure 43, which illustrates the elevator at the second lowest deck. It consists of a guide rail of about 4600 mm length, connected to three brackets with equal spacing of 1500 mm. The axial force from the safety gear is assumed to act on both faces on the shoe, due to the braking action of the device.

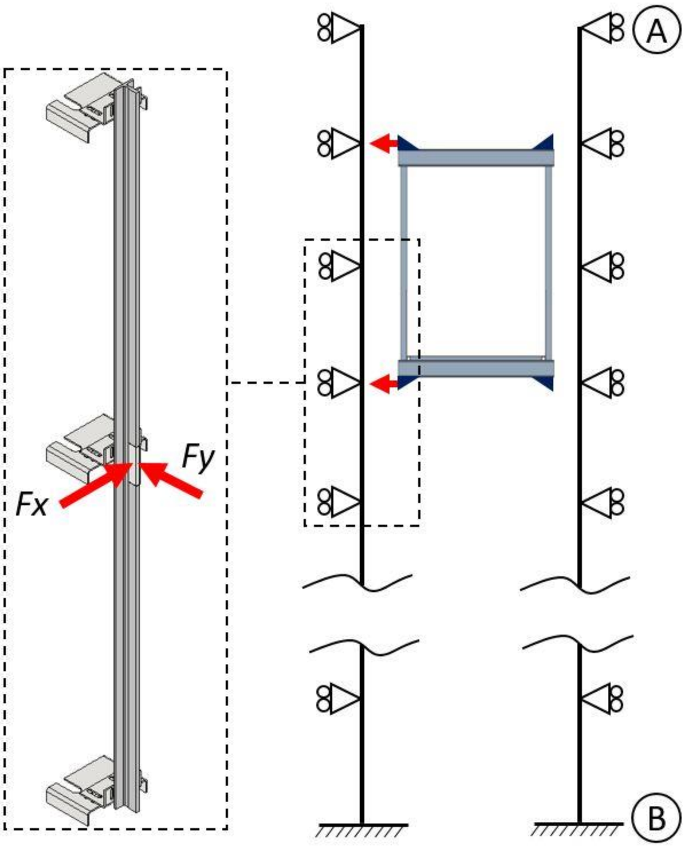

Figure 41. Force on one bracket.

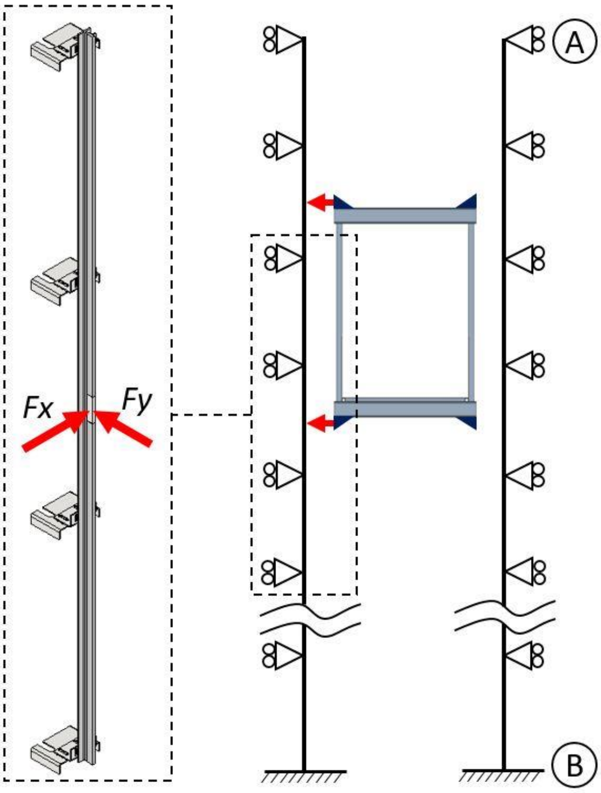

Figure 42. Force between two brackets.

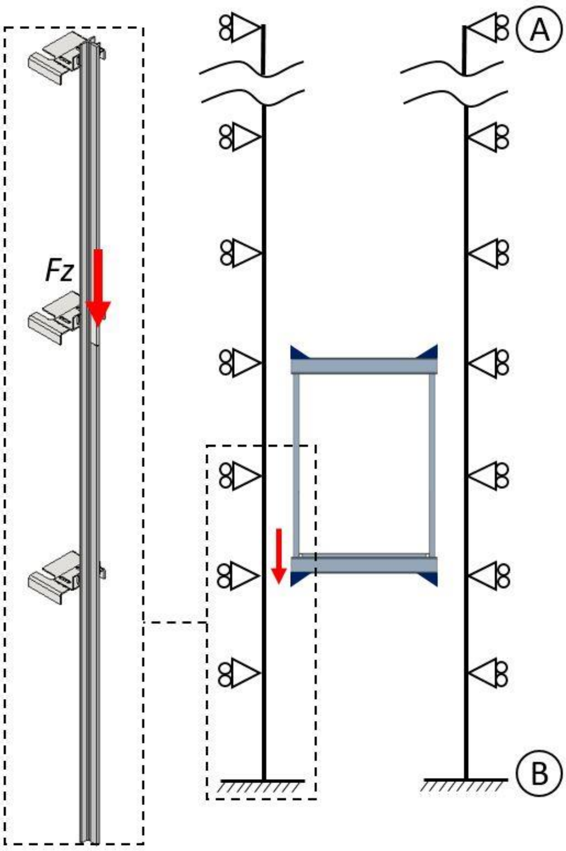

Figure 43. Safety gear operation.

All the brackets, the guide rail and the guide shoe (and rail clips for load case 3) are modelled using solid elements. The brackets in load case 1 and 2 are modelled without associated bolts and rail clips, since the purpose of these cases is to evaluate the strength capacity and not the contact stresses. It is more interesting to evaluate the capacity of the design rather than the connections between each component. It is therefore assumed that the contact surfaces between the L-profile and U-channel is *Bonded*. Bonded makes the region glued and does not allow for any sliding or separation between the faces. Rail clips are modelled in load case 3 to account for the ability to slide. It is therefore assumed that the contact surfaces between the L-profile, guide rail and rail clips are *Frictional*. Frictional allows for sliding and separation if the frictional force is exceeded [21]. This is model using a frictional coefficient of 0.2, which is one of the values used for clamped steel surfaces [22].

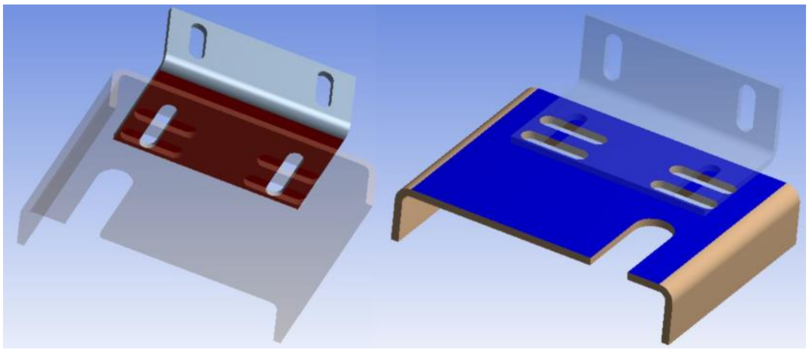

Figure 44. Bonded contact between L-profile and U-channel.

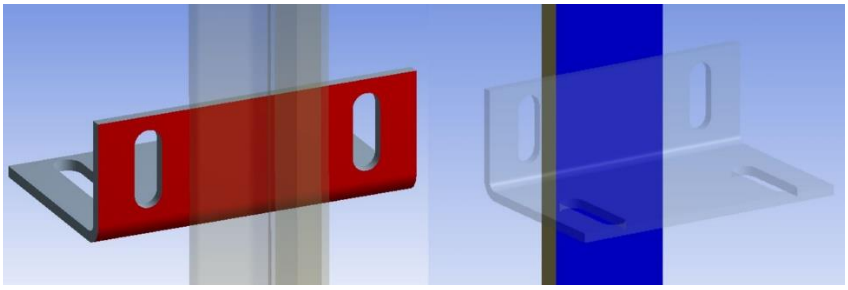

Figure 45. Bonded contact between L-profile and guide rail.

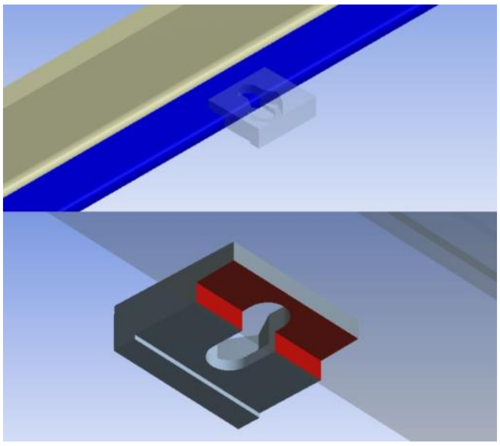

Figure 46. Frictional contact between rail clip, guide rail and L-profile.

Element mesh

Hex elements are chosen for all components because they are simple to modify and refine. It is easy to obtain a good *Element Quality*, with low *Aspect Ratio* and *Jacobian Ratio*, due to their geometry and ability to fill the volume of the solid element. The guide rail is meshed with a coarse mesh of 25 mm element size. The evaluated bracket(s) are meshed with an element size of 3 mm, and all other with an element size of 10 mm. This gives at least three elements in the thickness direction of the evaluated bracket. Hex elements are achieved by defining the solid elements to be mesh according to the *MultiZone* or the *Hex Dominant* method. The quality of the mesh is normally measured based on the element quality, the aspect ratio and the Jacobian ratio. The quality of each element is rated from 0 and 1, with 1 as the best element quality. For a good mesh, it is therefore important to increase the lowest value in the interval. As for the aspect and the Jacobin ratio, these values vary from negative values to very large positive values. For good results in these cases, one should avoid the negative values and reduce the large values as much as possible. It is referred to ANSYS Meshing User's Guide for further reading on the metrics [23]. Mesh convergence is also an important measure of element size. If the results for the stresses and displacements have converged, it is not necessary to refine the mesh anymore. This was tested with a reduction of 1 mm per simulation, and the results converged at an element size of 3 mm. The mesh procedure resulted in the following mesh metrics:

- Element Quality 0.6-1
- Aspect Ratio 1-2.9
- Jacobian Ratio 1-3

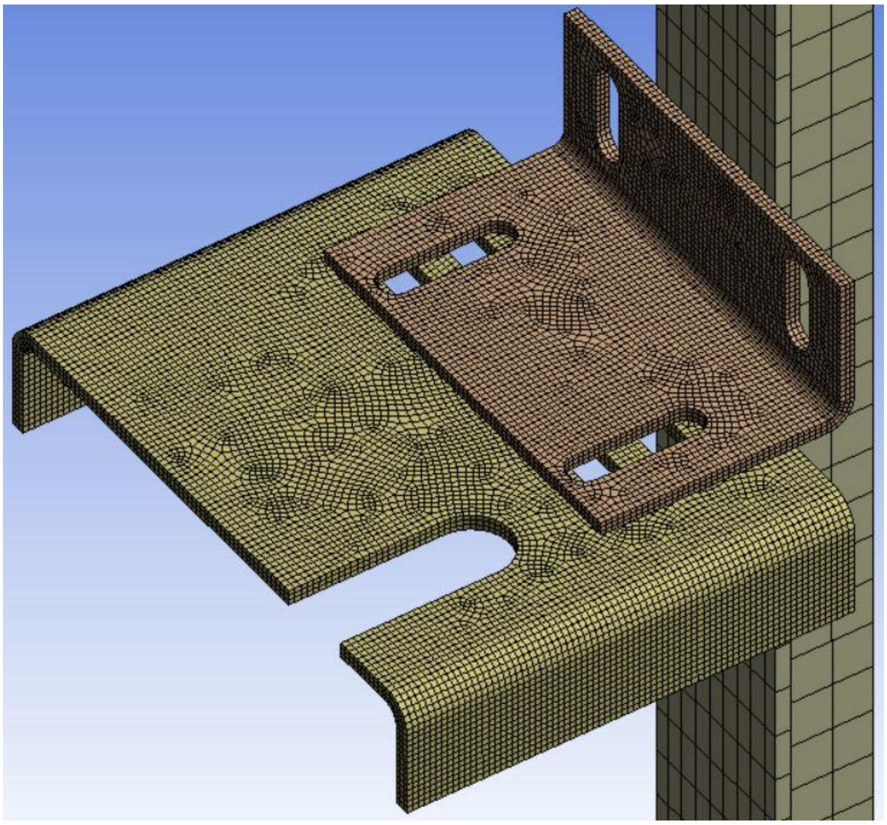

Figure 47. Mesh of bracket and guide rail.

Applied loads

DNV GL's rules for Classification of Ships is used to calculate the maximal force acting on the bracket(s) for load case 1 and 2. This is done by calculating the transverse and longitudinal forces at maximum elevator height for eight different ships, which is selected from the three ship classes defined in section 5.1. It was shown from the calculations that the transverse force was of highest magnitude. The worst case at 4000 kg elevator weight is therefore used to evaluate the capacity of the brackets. This force is used for evaluating the capacity in both transverse and longitudinal direction of the brackets, as shown in Figure 48.

As for the vertical force, it is only calculated using the force resulting from safety gear operation. The weight of the guide rails is not included due to the magnitude of the force. It is instead calculated using the same elevator weight and a safety gear coefficient of 15. As for the applied loads in ANSYS, *Force* is used to model the transverse force distributed on the faces of the guide shoe, while *Remote Force* is used to simulate the vertical force resulting from the safety device, as indicated on Figure 49.

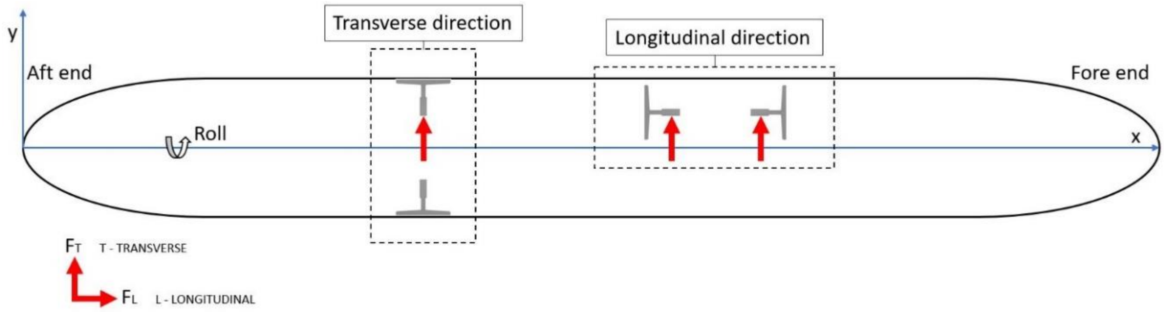

Figure 48. Transverse and longitudinal direction of brackets.

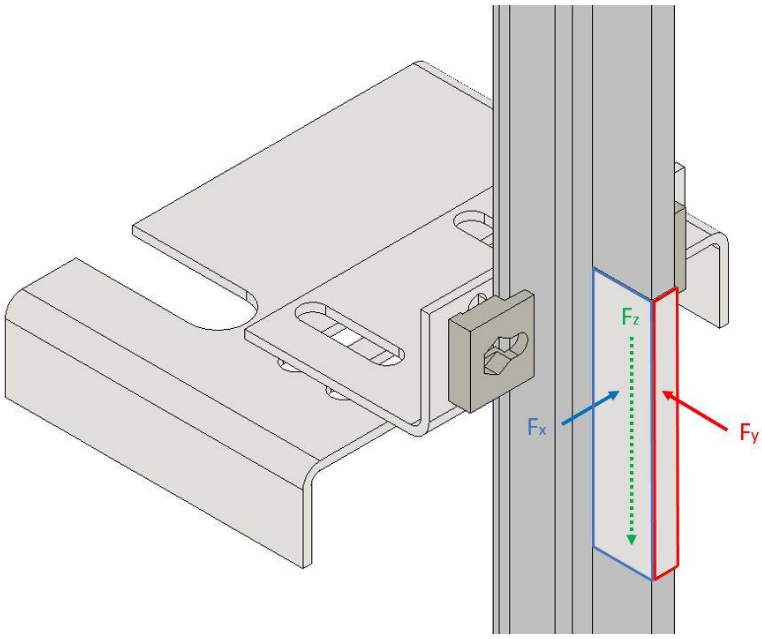

Figure 49. Applied loads.

Applied boundary conditions

The boundary condition *Fixed Support* is assigned to the brackets and the baseplate, as illustrated by the blue surfaces in Figure 50. A fixed constraint means that it is not allowed for any translational or rotational degree of freedom, which is a good assumption for welded surfaces. Notice that the bracket is also welded to one side of the stiffener, and not just in the rear end.

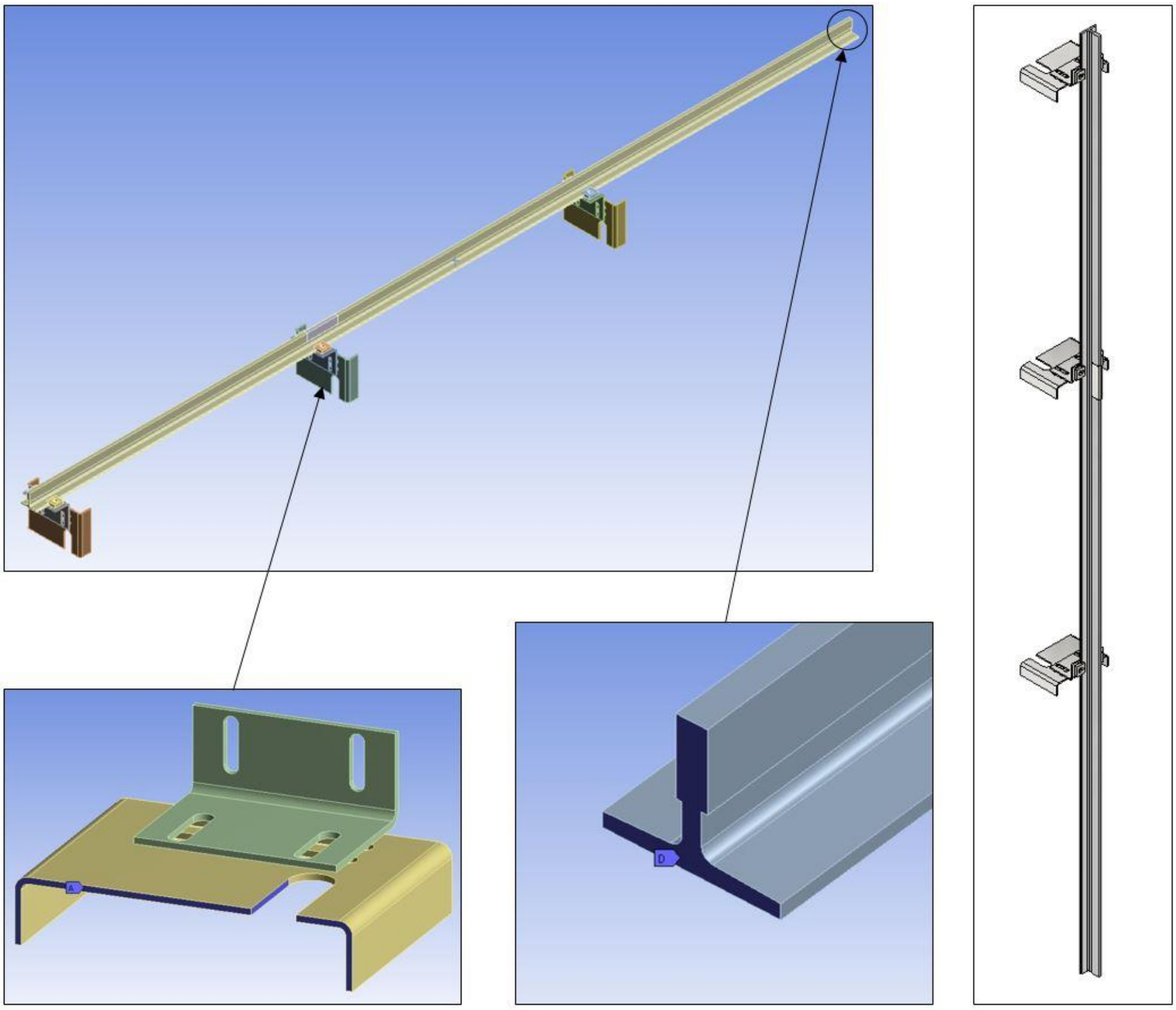

Figure 50. Boundary condition for brackets and baseplate.

6 Results

This chapter presents the guide rail calculations for a specific project, which includes the type of vessel and an example of a typical elevator weight. The potential cost savings of guide rail reduction and use of standard brackets are also presented. This involves one example of material reduction and one example of reduced installation time. A preview of the FEM results is also presented, which includes the worst case of both brackets. Lastly, the proposed design changes for the customized bracket are then presented.

6.1 Comparison in guide rail calculation methods

The expedition vessel Hurtigruten and a passenger elevator of 1000 kg capacity are used in the example to present the guide rail calculations.

Elevator specifications	
Capacity, rated load	1000 kg
Width and depth of the car	1400 x 1600 mm
Weight of car and car sling	1200 kg
Distance between brackets	1700 mm
Location, measured from AE, CL, BL [x y z]	[55.8 2.5 32] m

Table 16. Elevator specifications.

Calculation method	*Guide rails*	*Max. stress*	*Max. deflection*
Method 1, DNV GL	T90/B	σb_{roll} = 125.0 MPa	δ_{roll} = 2.8 mm
Method 2, DNV GL	T82/B	σb_{roll} = 117.6 MPa	δ_{roll} = 2.9 mm
Method 3, LR	T114/B	σb_{comb} = 105.4 MPa	δ_{roll} = 1.1 mm
Method 4, LR	T114/B	σb_{comb} = 105.7 MPa	δ_{roll} = 1.3 mm

Table 17. Calculated guide rail dimension for each method.

6.2 Savings on guide rail reduction

Table 18 presents the guide rail dimensions for two different offshore projects and the potential savings of material reduction. One passenger lift per ship is used as examples.

		Hurtigruten	***Ulstein 307***
Planned to be used	Guide rails	T127-2/B	T127-2/B
	Bracket spacing	1500 mm	1500 mm
Suggested dimensions	Guide rails	T90/B	T90/B
	Bracket spacing	2000 mm	2000 mm
Savings	Cost	-666 EUR	-444 EUR
	Weight	-406 kg	-271 kg

Table 18. Planned vs. suggested dimensions.

6.3 Installation of different guide rails and brackets

Table 19 presents four offshore projects with the number of ship orders and elevators planned to be installed in each ship. All projects are based on the same ship class with the assumption of similar trunks.

Projects	***Number of ships***	***Number of elevators***	***Total***
Hurtigruten	4	8	32
Hapag Lloyd	2	5	10
Color Line	1	9	9
Sunshine	1	6	6
Total number of trunks			*57*

Table 19. Offshore projects with similar installations.

Table 20 presents how long it would take and how much it would cost for each guide structure to be installed in 57 similar trunks. It does also present the potential cost savings depending on choice of guide structure. The result is based on the cases defined in section 5.5.

	Base case 1	***Base case 2***	***Base case 3***	***Base case 4***
Description of guide structure	Customized with T127-2/B	Standard with T127-2/B	Customized with T90/B	Standard with T90/B
Number of hours	2850 h	1995 h	2508 h	1710 h
Total cost	427 500 EUR	299 250 EUR	376 200 EUR	256 500 EUR
Cost savings		*- 30 %*	*- 12 %*	*- 40 %*

Table 20. Cost of different guide structure installations.

6.4 Mapping of load cases

Figure 51 shows a graphical representation of the forces as a function of the elevator weight. The solid lines represent the transverse force, while the dash lines represent the longitudinal force. Note to Figure 51; "C3" is a shortening for class 3, "C2" for class 2 and "C1" for class 1.

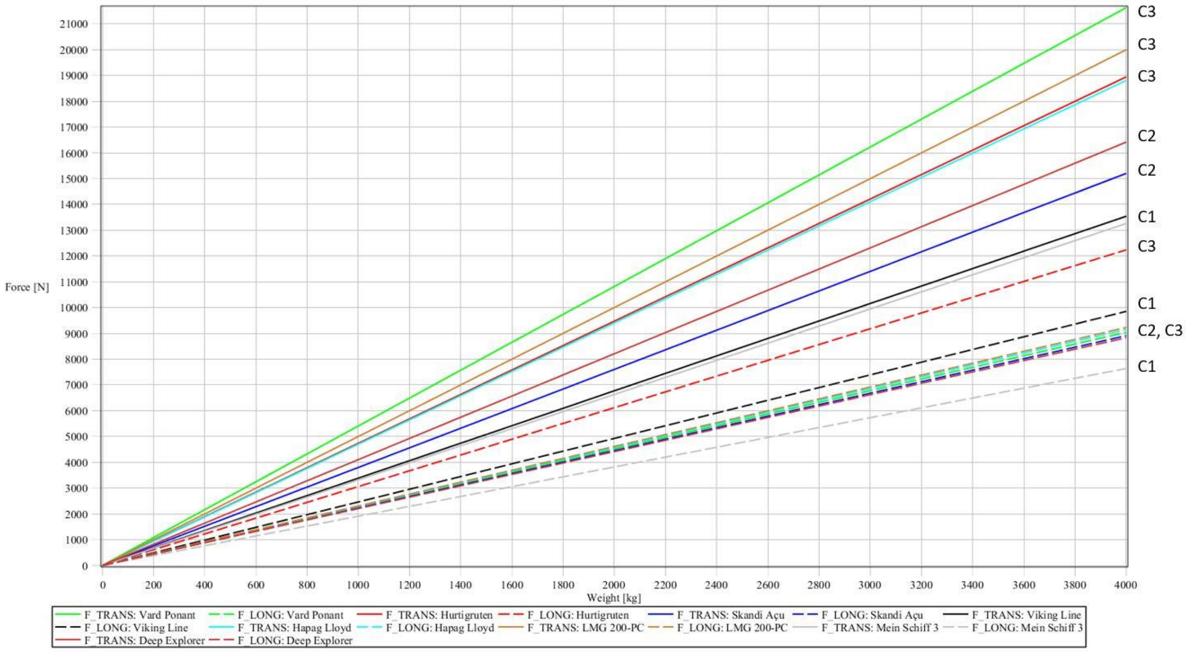

Figure 51. Calculation of ship classes.

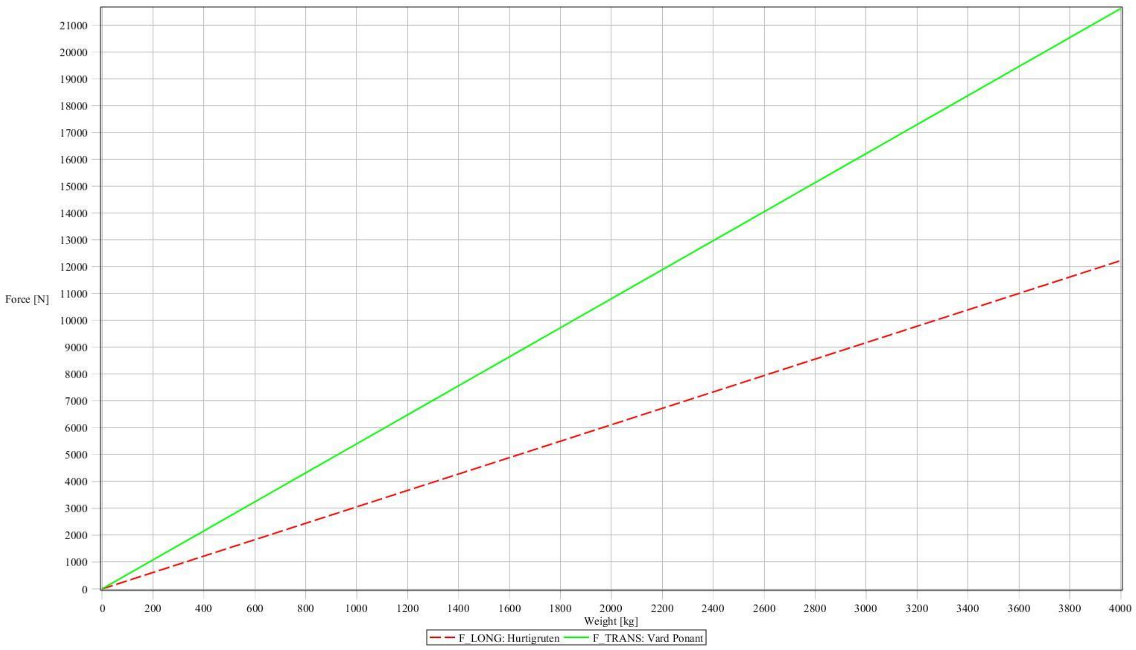

Figure 52. Worst case of transverse and longitudinal force.

6.5 FEM analysis of brackets

Figure 53 presents the stresses resulting from the force acting directly on one bracket, parallel to the trunk, while Figure 54 presents the stresses resulting from the force acting between two brackets, also parallel to the trunk.

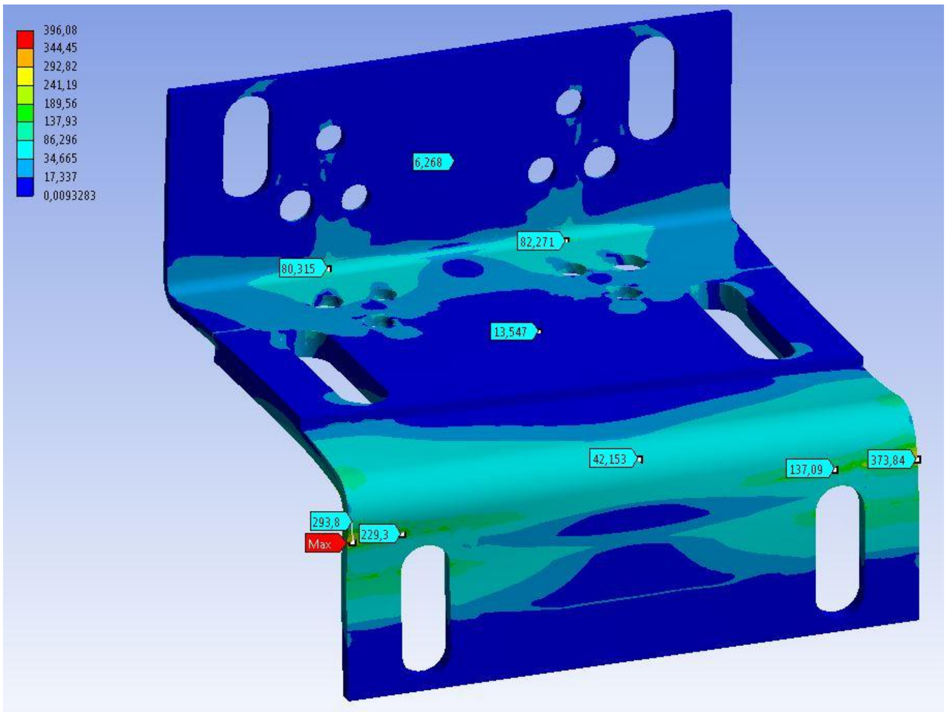

Figure 53. Standard bracket.

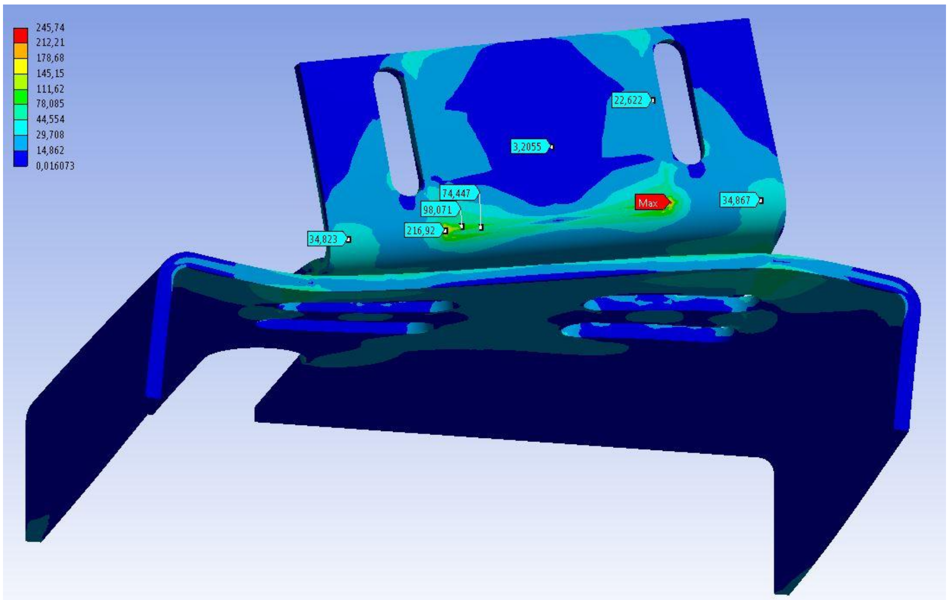

Figure 54. Customized bracket.

6.6 Proposed bracket design

The following design changes are proposed:

- U-channel: 6 mm plate thickness, 270 x 220 x 50 mm
- L-profile: 4 mm plate thickness, 210 x 130 x 80 mm
- Rectangular cut out of 107 x 25 mm, with 5 mm radius in edges
- Standard L-profile from TKA

This results in the following specifications:

- Approximate 6 kg with accessories
- ± 70 mm sideways travel with respect to neutral position
- Approx. 60 mm outwards travel with respect to neutral position
- Applicable for 5 guide rail dimensions

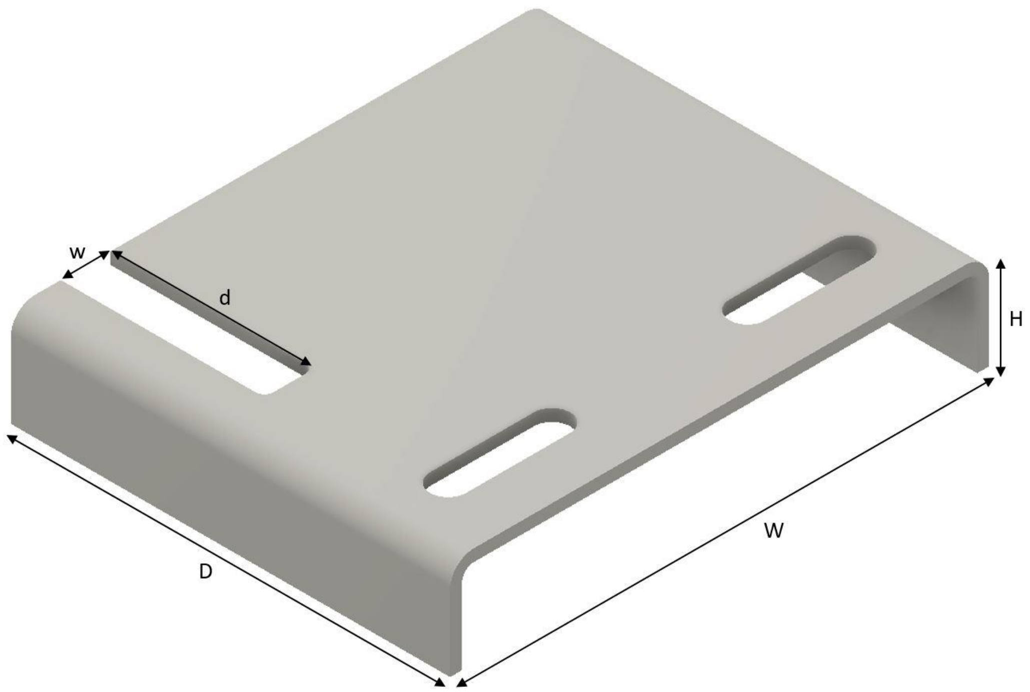

Figure 55. Proposed changes for U-channel.

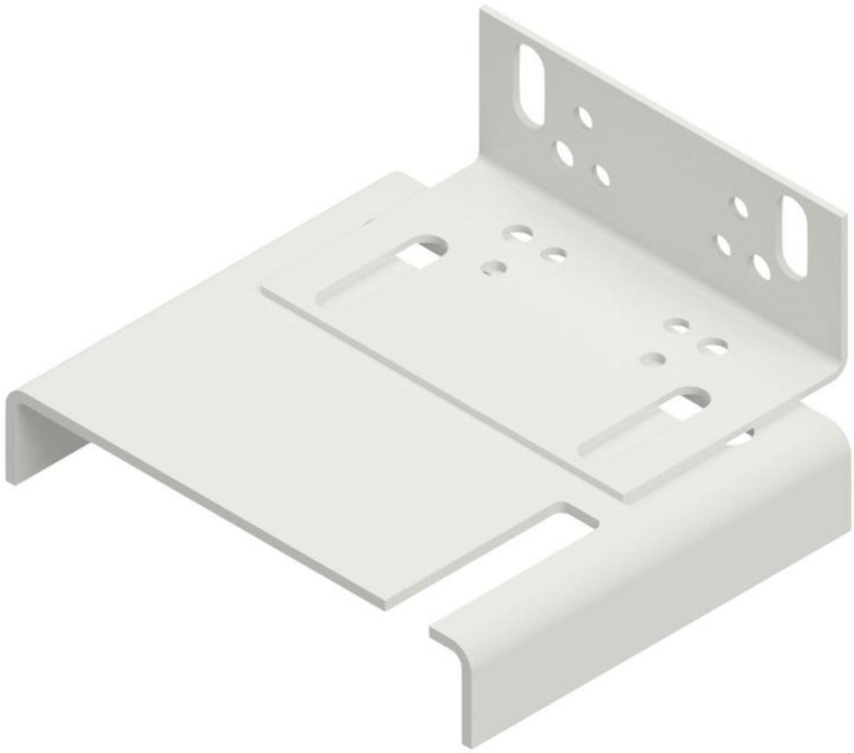

Figure 56. L-profile from standard bracket solution.

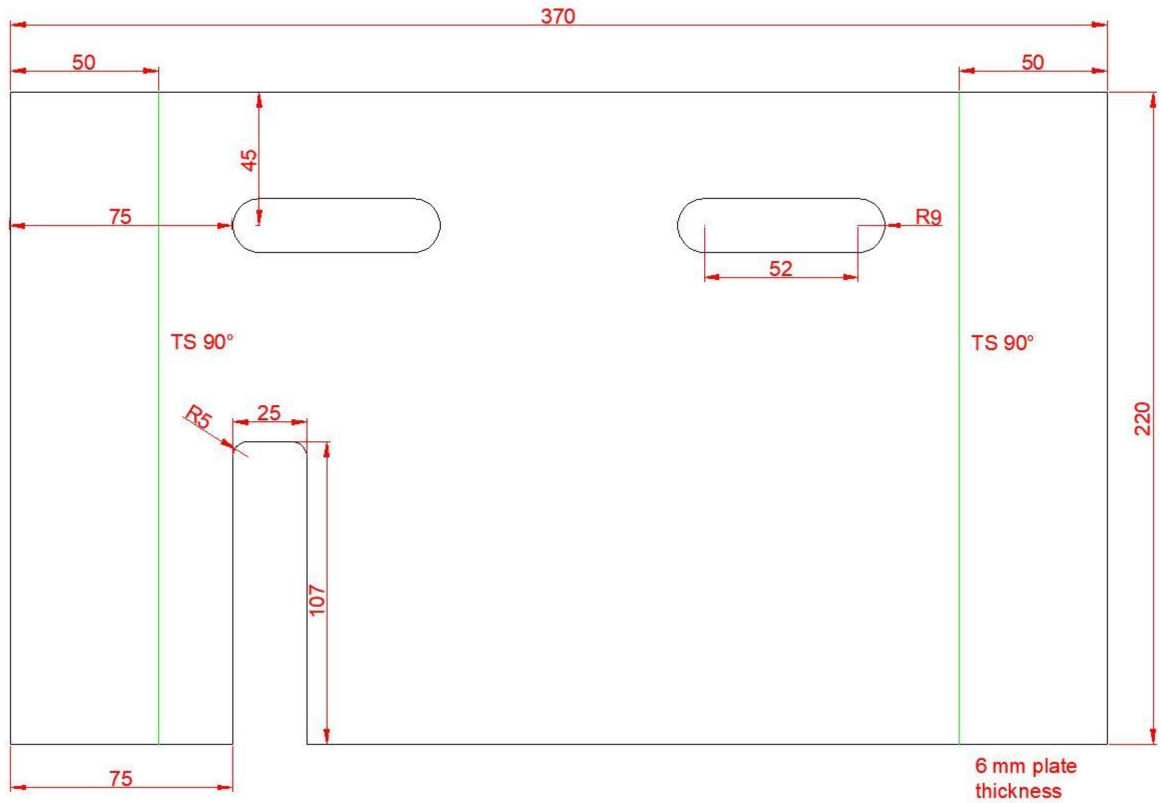

Figure 57. Dimensional drawing.

7 Discussion

This chapter discusses the main subjects of the thesis, which includes the four methods of calculating the guide rail dimensions and the spacing between the brackets, the ship classes with a mapping of the load cases, as well as the assessment of the two bracket solutions. A discussion of the potential cost savings is also given, in addition to the verification of the calculation tool and bracket assessment.

7.1 Guide rail calculation

The results from section 6.1 shows that the guide rail dimensions vary depending on the calculation method. Method 1 and 2 calculates the smallest dimensions, while method 3 and 4 calculates the largest dimensions. All four methods do however calculate smaller dimensions compared to what TK normally use. The difference between calculating according to DNV GL or LR is the combination of stresses and the restrictions on allowable deflection. DNV GL does not assume the stresses from roll and pitch to act simultaneously, something that is practiced by LR. LR allows however more deflection than DNV GL, but this is not the dimensional case.

Both Hurtigruten and Ulstein 307 are planned to be installed with T127-2/B guide rails, with a bracket spacing of 1500 mm, for one of their passenger lifts. Hurtigruten, which is classed according to DNV GL, could instead be installed with T90/B with a bracket spacing of 2000 mm. And Ulstein 307, which is classed according to LR, could also be installed with the same guide structure for the car. This is just an example from two trunks, but one could clearly see the potential of the methods.

Method 1

The result shows that method 1 is more conservative than method 2. It is however a quick and easy method for doing estimates, with not much of information needed to calculate the guide rails. The method is applicable for calculating the forces from operating, stowed and exceptional condition. However, the worst condition is clearly defined, and it is only necessary to calculate stowed condition if the aim is to quickly estimate the guide rail dimensions. DNV GL will still require the complete documentation, consisting of the guide rail stresses and deflections from operating, stowed and safety gear operation. Regarding the stowed condition, one must not include the rated load when calculating the guide rails with the weights from the passenger lift. Special considerations are needed for service lifts, and one must evaluate if exception should be made. It should be included in calculation of stowed condition if the elevator is meant for storing cargo or goods. A service lift in a luxury yacht is usually meant for the transportation of staff with some beverage, and no possibilities for storage, and should therefore be regarded as a passenger lift in the calculations. Furthermore, the method is known for the Lifting Appliances department of DNV GL. They are familiar with the procedures and theory of free harmonic vibrations, and the documentation would therefore be accepted given that the requirements are fulfilled.

Method 2

This method is probably the most accurate in representing the actual load case, due to the ship specific calculations. One could therefore consider it as the most optimized in terms of the guide rail calculations. The method is only applicable for calculating stowed condition, since it is based on extreme conditions at sea. It does however require additional information in the start-up phase of the project, which makes it more time consuming. This could on the other hand be solved with proper routines between the shipyard and TK. It is also a new method of calculation not presented to the Lifting Appliances department yet, but the method is created and used by the Ship Structures department, for examples in calculation of deck and machinery foundation strength. Therefore, it would be a contradiction to reject or not evaluate the method, before consulting to the Ship Structures department first.

Method 3 and 4

The results from method 3 and 4 are quite similar. They are in most cases more conservative than the methods based on DNV GL, but in some cases similar as well. One could say that the results are hard to anticipate, and it must be calculated for all conditions before a statement is made. The methods are based on the same equations for calculating the static and dynamic forces, but method 4 does also include the possibility to calculate other ship motions with ship specific input. The reason method 3 and 4 produces similar results is most likely because of the restrictions on roll and pitch angles in stowed condition, and that a minor difference in angle does not influence the calculation of forces that much.

7.2 Ship classes

The results from the mapping of load cases shows that the vessels in class 3 are subjecting the guide structure for the largest forces, both in transverse and longitudinal direction, while the ships in class 1 are subjecting the structure for the smallest forces. One could therefore assume that increasing ship motions are more critical than higher elevator travel.

The mapping of ship classes is based on DNV GL's Classification of Ships, which again builds on IACS Common Structural Rules for Bulk Carriers and Oil Tanker. These types of vessels are mainly meant for the transportation of bulk cargo and oil in the North Atlantic Ocean, which is known for its rough environmental conditions. This is comparable to the service and supply vessels, which operates in rough conditions in the North Sea, the Norwegian Sea and the Barents Sea. The two other ship classes consist however of expedition vessels and cruise liners that are mainly meant for operating along different coasts, but also crossing open sea for new adventures. And if bad weather is expected, the captain will plan for alternative routing or an extra day at the port. In other words, they are subjected to better conditions than the service and supply vessels. This issue is not considered in the thesis, and all ship classes are instead calculated with the same conditions.

A standardization of load cases is not as practical and easy as first assumed. TK deals with many different ship types, and it is hard to divide them into three classes with similarities in both ship dimensions, motions and elevator travel height. Based on the result, which showed too much variation of forces in their respective class, it is not recommended to use a standardization for this purpose. There was no common thread that indicated what forces to expect in each class. Instead, a more customized and accurate method is suggested. The shipyard will always give their potential suppliers the information about the main particulars or the key figures of the ship. Detailed documentation of the forces acting on the guide structure could therefore be calculated with the use of the particulars and position of the elevator. The block coefficient, and in some cases the depth from the main deck, is the only additional information that TK needs to request from the shipyard.

7.3 FEM analysis

A structural FEM analysis is used to evaluate the strength capacity of both brackets, with loads of 4000 kg applied in three different directions. Most of the results seems reliable in terms of how the components deform. The exception is the result from the load being applied in y-direction on one bracket, especially for the customized solution. Most of the deformation is obtained in the L-profile, since this component yields before the U-channel. This resulted in a necking in the area near the outer angle. One could question if this is the actual deformation when exposing the profile to this kind of load case. An alternative mesh was modelled to see if this would give a different result, but the results were quite similar. The necking resulted in a stress concentration with medium to high stresses. The high stresses are however distributed on a very small area, typical local peak stresses. It is therefore not critical for the rest of the structure, and evaluated as acceptable. Furthermore, both brackets were exposed to five different load cases. The worst case for the customized solution was the model with the force acting between the two brackets, applied in x-direction, while the worst case for the standard solution was the model with the force acting on one bracket, applied in x-direction.

7.4 Customized bracket solution

The customized brackets are designed to be applicable for trunks with stiffeners on the inside, and to have sufficient strength capacity with the use of high safety factors. The results from the FEM analysis shows that the plate thickness of the U-channel can be reduced to 6 mm, and still have acceptable safety factors. A minor reduction is not recommended due to the welding of the bracket, since good weldability is achieved with equal plate thickness between the bracket, bulkhead and stiffener. Furthermore, the bracket would be applicable for all bulb profile stiffeners up to 100 x 8 mm with a rectangular cut out of 107 x 25 mm, and manufactured with a radius of 5 mm in each corner. Larger dimensions are usually not used for the ships that TK are working on, but one should keep in mind that this could happen. No recommendations are given regarding the depth, but specified to 220 mm on the drawing. One should however adjust the depth for each project, which is equal to the distance from the guide rail to the trunk. The width is reduced by 30 mm without losing the possibility for sideways adjustments. A reduction of the plate thickness and the width would not save much weight, but make it easier to handle in the installation phase. This is however not included in the estimation of installation cost. The holes are reduced from four to two, since the bracket is properly constrained with only two M16 hex bolts and associated parts. Four holes are seldom used in the installation phase as well, based on the information from TK. This is also verified with calculation of the bolts strength capacity.

7.5 Standard bracket solution

The standard brackets have not been used for any offshore projects yet, mainly because of its drawback on stiffeners, but also because of its assumed strength capacity. It has not been considered as a robust solution, capable of supporting the forces resulting from the ship motions. Based on the FEM analysis however, it is shown that the design has surprisingly good strength properties. The strength capacity is not as good as the customized solution, but it can be used for loads up to 4000 kg. The highest stresses are obtained in the intersection between the L-profile and the Halfen channel. It is not recommended to use it for larger weights, as this seems to be the limit of allowable yielding.

7.6 Cost savings

Reducing the guide rail dimensions from T127 to T90 would save approximately 300-400 kg and 450-650 EUR per trunk, based on the two examples of expedition vessels. It would not save as much weight compared to other components on the ship, but be helpful in terms of the transportation from the shipyard to the trunk. As for the brackets, it is shown that replacing the customized bracket with the standard bracket would not save much weight, given that the plate thickness of the customized solution is reduced to 6 mm. This is because they would both end up on the approximately same weight. As for the cost of installation, the result shows that it is not the reduction of guide rail dimensions that affects the installation time, but rather what bracket they use for the installation. For the example used in this thesis, it is shown that a reduction of 30 % in cost of installation could be achieved just by using the standard brackets instead of the customized brackets, while only a reduction of 12 % could be achieved if only the guide rail dimensions were reduced. It is also shown that a reduction of 40 % could be achieved if the T127 guide rails were replaced with the T90 guide rails, and the customized brackets replaced with the standard brackets. The result of cost savings shows therefore that the installation time is more influencing on the cost than the price of the material. It could also be said that optimizing the brackets would save more cost than optimizing the guide rails.

7.7 Verification of calculation tool and bracket assessment

Method 1 of calculating the guide rails has been presented to the department in Ålesund, and to their colleagues in Neuhausen. Personal correspondence has also been received from DNV GL, stating that the calculation will be accepted by their section given that the calculation of stresses in both conditions are presented. The method is already in use by the engineers at TK. They have at several projects sent corrective measures to TKA regarding the reduction of guide rail dimensions, but still waiting for their updated offer. In addition, no formal documentation has been received from TKA, stating that their Research and Development department is using the calculation tool. As for the brackets, TK are planning to use the standard brackets instead of the customized brackets for their upcoming projects, for instance on their ferries and expedition vessels. And according to TK's latest update, it is stated that the expedition vessel Vard Ponant will most likely be installed with T90/B guide rails and standard brackets, which will save both weight and cost of material and installation.

8 Conclusion

In this thesis, four different methods for calculating the forces acting on the guide rails and brackets have been developed. The methods fulfill all the existing requirements for elevator guide structure on ships, and results in dimensions with high safety factors. Most important, all the methods calculate more optimized guide rail dimensions compared to what TK uses today. Initially, it is recommended to use method 1 in the calculation of guide rail strength, given that the ship is classed according to DNV GL. This is because they are familiar with the procedures, which also leads to reduced time for evaluation. There are a lot of upcoming projects for TK with decision to make on guide rail dimensions, and as the method is “approved”, it is therefore recommended to use it. In a longer perspective, it is suggested to use method 2 since this is probably the most accurate in calculating the load case, and it gives therefore the most optimized solutions. As for LR, they both give similar results and it is up to the engineer to select which method he/she prefers.

A mapping of load cases based on different ship types is presented in this thesis. The mapping was done to evaluate the need for standardized load cases. The ships were divided into three classes based on type, similarities in ship dimensions, ship motions and elevator travel height. It is not recommended to use a standardization for the calculation of the guide rail strength. Instead, it is recommended to calculate customized load cases for each project. Method 2 can be used to find accurate load cases, based on ship-specific input, only by requesting some additional information from the shipyard.

An assessment of two bracket solutions are presented in this thesis. The customized bracket is applicable for all kind of trunk layouts and designed with large strength capacity. The results from the FEM analysis shows that it is not necessary to manufacture the bracket with the current dimensions, especially the plate thickness, as it is overdimensioned. It is therefore recommended to reduce the thickness of the plate and the width of the U-channel. Furthermore, it is recommended to use the L-profile from the standard bracket solution. This would reduce the weight and make the solution applicable for several guide rail dimensions. As for the standard bracket, it is verified that the solution is applicable for TK’s maritime and offshore applications. Based on the FEM analysis, the result shows that it could be used for loads up to 4000 kg. It is recommended to use this solution if the trunk is built with stiffeners on the outside, or if it is possible to place the guide rails in the middle of two stiffeners. TK should also be involved at an earlier stage of the project and make demands on how the trunk should be built.

The optimized guide structure, consisting of T90 guide rails and standard brackets, would save most of the cost, especially for the installation of the guide rails and brackets. As for cost of material, it is shown that the savings are obtained when the guide rail dimensions are reduced, and not by changing the brackets. This applies also for the reduction of material weight. The benefit of the standardized bracket is firstly obtained in the reduction of installation cost.

Lastly, the fact that Vard Ponant will most likely be installed with T90 guide rails and standard brackets is a verification of the work done in the thesis.

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10 Appendices

Appendix A	Guide rail calculations, method 1
Appendix B	Guide rail calculations, method 2
Appendix C	Guide rail calculations, method 3 and 4
Appendix D	Standard bracket solution
Appendix E	Customized bracket solution
Appendix F	Tentative title and abstract for an article

APPENDIX A

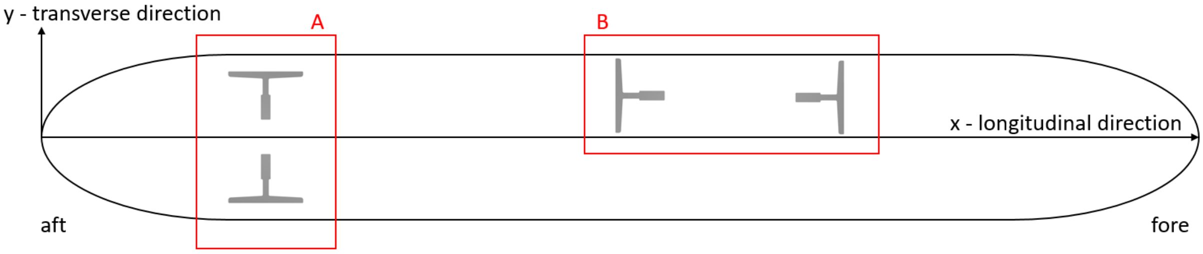
Guide rail calculations, method 1

based on

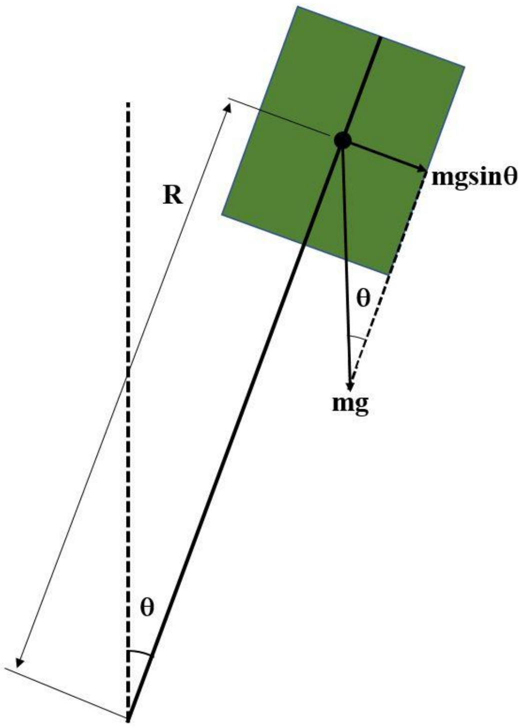
DNV's Rules for Certification of Lifts in Ships,
Mobile Offshore Units and Offshore Installations

1 Input

(1) Select guide rail		**T127-2/B**	
(2) Select elevator type		**Traction**	
(3) Select orientation of guide rail (see illustration)		**A**	
(4) Select car or counterweight		**Car**	
(5) Select condition		**Stowed**	
(6) Select car size (or insert customized values)	A	**1400x1600**	mm^2
(7) Select % counterweight	q	**0,5**	



(8) Specify distance between brackets	l_k	**1900**	mm
(9) Specify travelled cable weight suspended from car (for direct acting lifts only)	W_{rope}	**500**	kg



(10) Specify distance from rotation axis to mass center	R	25,25	m

The rotation axis R may be taken at a height z above the baseline:

$$z = \text{the smaller of } \frac{D}{4} + \frac{T_{LC}}{2} \text{ and } \frac{D}{2}$$

z vertical coordinate of the ship's rotation center

D moulded depth of ship

T_{LC} midship draught at loading condition

2 Output

(11) Weight of car	W_{car}	**1200**	kg
(12) Rated load	W_{rated}	**1000**	kg
(13) Counterweight	W_{cwt}	**1700**	kg
(14) Design weight	W_{design}	**1200**	kg
(15) Roll angle	θ	**22,5**	deg
(16) Roll period	T_{θ}	**10**	s
(17) Pitch angle	ϕ	**7,5**	deg
(18) Pitch period	T_{ϕ}	**7**	s
(19) Longitudinal acceleration	a_x	**2,7**	m/s^2
(20) Transverse acceleration	a_y	**3,9**	m/s^2

3 Result

(21) Transverse force	F_{trans}	**9202**	N
(22) Longitudinal force	F_{long}	**7700**	N
(23) Buckling force	$F_{buckling}$	**22 000**	N
(24) Bending moment from roll	Mb_{roll}	**2 185 570**	Nmm
(25) Bending moment from pitch	Mb_{pitch}	**914 434**	Nmm
(26) Bending stress from roll	σb_{roll}	**70,1**	MPa
(27) Bending stress from pitch	σb_{pitch}	**25,3**	MPa
(28) Deflection from roll	δ_{roll}	**1,6**	mm
(29) Deflection from pitch	δ_{pitch}	**0,6**	mm
(30) Buckling stress	σ_k	**12,4**	MPa
(31) Safety factor for roll	SF_R	**3,0**	
(32) Safety factor for pitch	SF_P	**8,4**	
(33) Safety factor for buckling	SF_B	**17,0**	

4 Technical background

	σ_y	I_x	I_y	W_x	W_y	A	σ_k
T70/A	225	409500	188600	9169	5389	940	140
T70-70-9/A	225	528100	246200	10790	7020	1125	140
T75/A	225	402900	264700	9286	7060	1091	140
T80-80-9/A	225	802000	388300	14210	9700	1356	140
T82/A	225	493100	301700	10270	7358	1091	140
T89/A	225	598300	524100	14350	11780	1577	140
T90/A	225	1020000	524800	20860	11660	1725	140
T125-L1/A	225	564600	1078800	10970	11260	1439	140
T75/B	265	402900	264700	9286	7060	1091	210
T78/B	265	299200	263900	7564	6766	985	210
T82/B	265	493100	301700	10270	7358	1091	210
T89/B	265	598300	524100	14350	11780	1577	210
T90/B	265	1020000	524800	20860	11660	1725	210
T114/B	265	1793000	1086000	29700	19050	2089	210
T125/B	265	1510000	1591000	26160	25460	2282	210
T127-1/B	265	1879000	1499000	30650	23610	2274	210
T127-2/B	265	2017000	2299000	31170	36200	2872	210
	MPa	mm^4	mm^4	mm^3	mm^3	mm^2	MPa

Size	Area	Capacity	Weight	CWT
1100x1600	1,76	1000	1200	1700
1400x1600	2,24	1000	1200	1700
1100x2100	2,31	1000	1200	1700
1525x1525	2,33	1000	1400	1900
1400x2000	2,80	1200	1300	1900
1370x2030	2,78	1200	1600	2200
1600x2100	3,36	1500	1380	2130
1800x2100	3,78	1800	1450	2350
1800x2300	4,14	2000	1500	2500
Customized	-	*1200*	*1600*	*2200*
	m^2	kg	kg	kg

5 Calculation procedure

Assume free harmonic vibration (example with force resulting from roll motion):

$\theta := 22.5\text{deg}$

$T\theta := 10\text{s}$

$\varphi := 7.5\text{deg}$

$T\varphi := 7\text{s}$

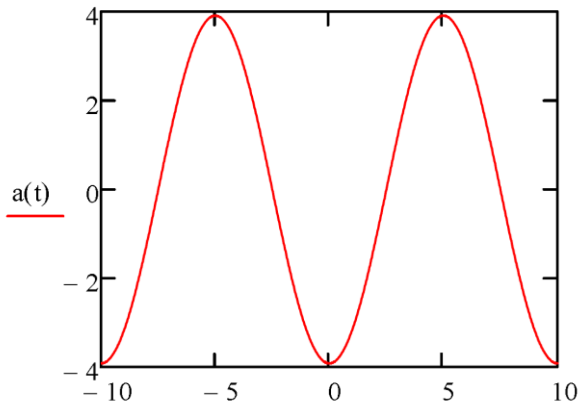
$B := R \cdot \theta$

$\omega := \dfrac{1}{T\theta}$

$x(t) := B \cdot \cos(2\pi\,\omega \cdot t)$

$v(t) := \dfrac{d}{dt}x(t)$

$a(t) := \dfrac{d}{dt}v(t)$



$n := 2$

$M := 1200\text{kg}$

$\text{Force} := \dfrac{1}{n} M \cdot \left(a\left(\dfrac{T\theta}{2}\right) + g \cdot \sin(\theta) \right)$

Buckling stress:

Radius of gyration	**i**	27	mm
Coefficient of slenderness	**λ**	72	
Buckling factor	**ω**	1,62	
Buckling stress	**σ_c**	12,4	MPa

6 References

Requirements for ship motions and calculations for buckling stress are from:

[1] DNV GL Rules for Certification of Lifts in Ships, Mobile Offshore Units and Offshore Installations (2008)

Specifications for the guide rails are from:

[2] Product catalog, Savera Elevator System Solutions, Savera Group

Bending moments, bending stresses and deflections are calculated based on elementary theory of simply supported beams.

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APPENDIX B

Guide rail calculations, method 2

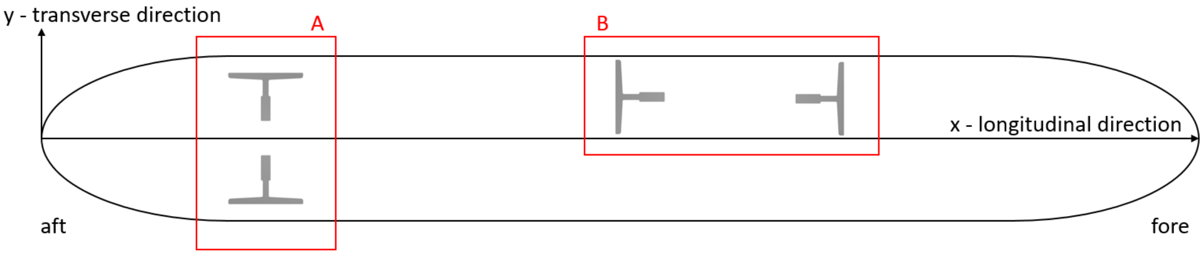
based on

DNV's Rules for Classification of Ships

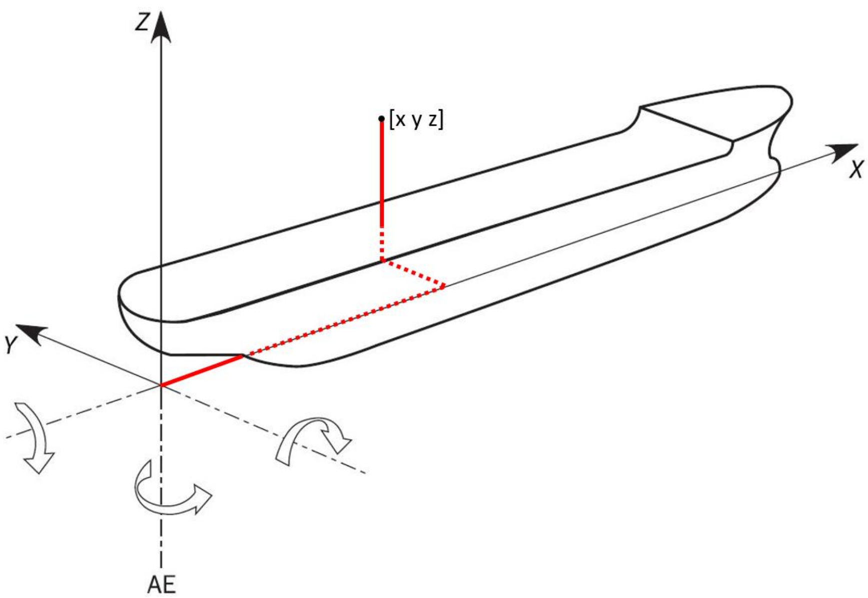
1 Input

1.1 Elevator specifications

(1) Select guide rail		T90/B	
(2) Select elevator type		Traction	
(3) Select orientation of guide rail (see illustration)		A	
(4) Select car or counterweight		Car	
(5) Select car size (or insert customized values)	A	1400x1600	mm^2
(6) Select % counterweight	q	0,5	

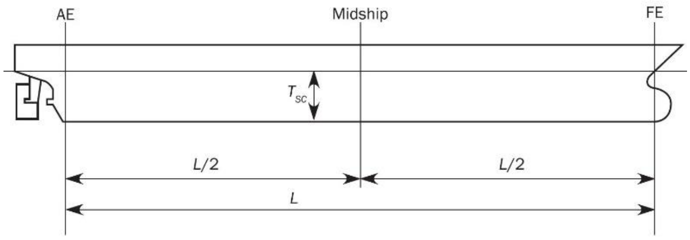


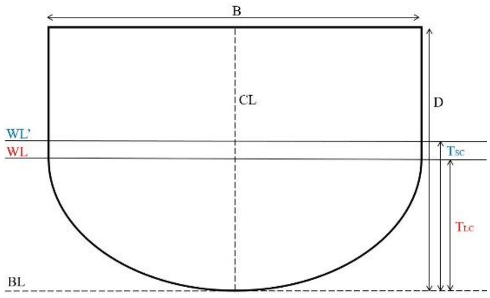
(7) Specify distance between brackets	l_k	1900	mm
(8) Specify travelled cable weight suspended from car (for direct acting lifts only)	W_{rope}	500	kg



(9) Specify x-coordinate of car/cwt from aft end	x	55,8	m
(10) Specify y-coordinate of car/cwt from centerline	y	2,5	m
(11) Specify z-coordinate of car/cwt from baseline	z	32	m

1.2 Ship dimensions and characteristics





(12) Specify length between p.p. – rule length	L_{pp}	133,5	m
(13) Specify moulded breadth	B	23,6	m
(14) Specify moulded depth	D	16	m
(15) Specify scantling draught	T_{SC}	5,5	m
(16) Specify midship draught at loading condition	T_{LC}	5,5	m
(17) Specify block coefficient at draught T_{SC}	C_B	-	

2 Output

(18) Weight of car	W_{car}	**1200**	kg
(19) Rated load	W_{rated}	**1000**	kg
(20) Counterweight	W_{cwt}	**1700**	kg
(21) Design weight	W_{design}	**1200**	kg

(22) Roll angle	θ	**28,7**	deg
(23) Roll period	T_θ	**16,5**	s

(24) Pitch angle	ϕ	**15,7**	deg
(25) Pitch period	T_ϕ	**10,1**	s

(26) Longitudinal acceleration	a_x	**4,6**	m/s^2
(27) Transverse acceleration	a_y	**7,1**	m/s^2
(28) Vertical acceleration	a_z	**5,0**	m/s^2

3 Results

(29) Transverse force	F_{trans}	**5684**	N
(30) Longitudinal force	F_{long}	**3671**	N
(31) Buckling force	$F_{buckling}$	**22 000**	N
(32) Bending moment from roll	Mb_{roll}	**1 349 905**	Nmm
(33) Bending moment from pitch	Mb_{pitch}	**435 990**	Nmm
(34) Bending stress from roll	σb_{roll}	**64,7**	MPa
(35) Bending stress from pitch	σb_{pitch}	**37,4**	MPa
(36) Deflection from roll	δ_{roll}	**1,9**	mm
(37) Deflection from pitch	δ_{pitch}	**1,2**	mm
(38) Buckling stress	σ_k	**22,3**	MPa
(39) Safety factor for roll	SF_R	**3,3**	
(40) Safety factor for pitch	SF_P	**5,7**	
(41) Safety factor for buckling	SF_B	**9,4**	

4 Technical background

	σ_y	I_x	I_y	W_x	W_y	A	σ_k
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T127-2/B	265	2017000	2299000	31170	36200	2872	210
	MPa	mm^4	mm^4	mm^3	mm^3	mm^2	MPa

Size	Area	Capacity	Weight	CWT
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1100x2100	2,31	1000	1200	1700
1525x1525	2,33	1000	1400	1900
1400x2000	2,80	1200	1300	1900
1370x2030	2,78	1200	1600	2200
1600x2100	3,36	1500	1380	2130
1800x2100	3,78	1800	1450	2350
1800x2300	4,14	2000	1500	2500
Customized	-	*1200*	*1600*	*2200*
	m^2	kg	kg	kg

5 Calculation procedure

Calculation of ship motions, resulting forces and buckling stress:

Acceleration parameter	a_0	0,54	m/s²
Vertical coordinate of the ship's rotation center	**R**	6,75	m
Roll radius of gyration	**kr**	9,20	m
Metacentric height	**GM**	1,65	m
Ratio between T_{LC} and T_{SC}	f_T	1,00	
Heading correction factor	f_β	1	
Coefficient for strength assessment	f_{ps}	1	
Reduction factor related to service restrictions	f_r	1	
Bilge keel factor	f_{BK}	1	
Correction factor	f_p	1	
Pitch factor	λ_ϕ	160,2	
Roll angle	θ	28,7	deg
Roll period	T_θ	16,5	s
Pitch period	ϕ	15,7	deg
Pitch angle	T_ϕ	10,1	s
Surge acceleration	a_{surge}	1,74	m/s²
Sway acceleration	a_{sway}	2,71	m/s²
Speed (L < 100)	v_1	0	kt
Speed (100 ≤ L < 150)	v_2	3,35	kt
Speed (L ≥ 150)	v_3	5	kt
Selected speed based on length of ship	$v_{selected}$	3,35	kt
Heave acceleration (L < 100)	a_{heave}	5,00	m/s²
Heave acceleration (100 ≤ L < 150)	a_{heave}	4,98	m/s²
Heave acceleration (L ≥ 150)	a_{heave}	5,16	m/s²
Selected heave acceleration based on length of ship	$a_{heaveselect}$	4,98	m/s²
Roll acceleration	a_{roll}	0,07	m/s²

Appendix B: Guide rail calculations

Rules for Classification of Ships

Pitch acceleration (L < 100)	a_{pitch}	0,124	m/s²
Pitch acceleration (100 ≤ L < 150)	a_{pitch}	0,119	m/s²
Pitch acceleration (L ≥ 150)	a_{pitch}	0,120	m/s²
Selected pitch acceleration based on length of ship	$a_{pitchselect}$	0,119	m/s²
Correction factor based on length of ship			
(L < 150)	f_L	1,39	
(100 ≤ L < 150)	f_L	0,86	
(L ≥ 150)	f_L	0,60	
Selected correction factor based on length of ship	$f_{Lselected}$	0,86	
Correction factor based on speed	f_v	0,33	
Pitch acceleration	$a_{pitch-x}$	3,00	m/s²
Envelope longitudinal acceleration (z < 0,875 T_{LC})	a_{x-env}	3,81	m/s²
Envelope longitudinal acceleration (0,875 T_{LC} ≤ z < 1,75 T_{LC})	a_{x-env}	5,28	m/s²
Envelope longitudinal acceleration (z ≥ 1,75 T_{LC})	a_{x-env}	4,57	m/s²
Selected longitudinal acceleration	$a_{x-selected}$	4,57	m/s²
Transverse acceleration	a_{roll-y}	1,83	m/s²
Envelope transverse acceleration	a_{y-env}	7,07	m/s²
Pitch acceleration in z-direction	$a_{pitch-z}$	0,02	m/s²
Roll acceleration in z-direction	a_{roll-z}	0,18	m/s²
Envelope vertical acceleration	a_{z-env}	4,98	m/s²
Vertical force alone	F_V	14760	N
Vertical force in combination with transverse force	F_{VC}	11772	N
Transverse force in combination with vertical force	F_{TC}	5684	N
Vertical force in combination with longitudinal force	F_{VC}	14760	N
Longitudinal force in combination with vertical force	F_{LC}	3671	N
Radius of gyration	i	24	mm
Coefficient of slenderness	λ	78	
Buckling factor	ω	1,75	
Buckling stress	σ_k	22,3	MPa

6 References

Ship motions and accelerations are calculated from:

[1] DNV GL Rules for Classification: Ships (Jan 2017)
Part 3 Hull Chapter 4 Load

Transverse, longitudinal and vertical forces are calculated from:

[2] DNV GL Rules for Classification: Ships (Jan 2016)
Part 1 Chapter 1 Section 4 Design Loads C500

Buckling stress is calculated from:

[3] DNV GL Rules for Certification of Lifts in Ships, Mobile Offshore Units and Offshore Installations (2008)

Specifications for the guide rails are from:

[4] Product catalog, Savera Elevator System Solutions, Savera Group

Bending moments, bending stresses and deflections are calculated based on elementary theory of simply supported beams.

APPENDIX C

Guide rail calculations, method 3 and 4

based on

Lloyd’s Register’s Code for Lifting Appliances
in a Marine Environment

1 Input

1.1 Introduction

If no actual ship-specific data is available, the following ship motions are used:

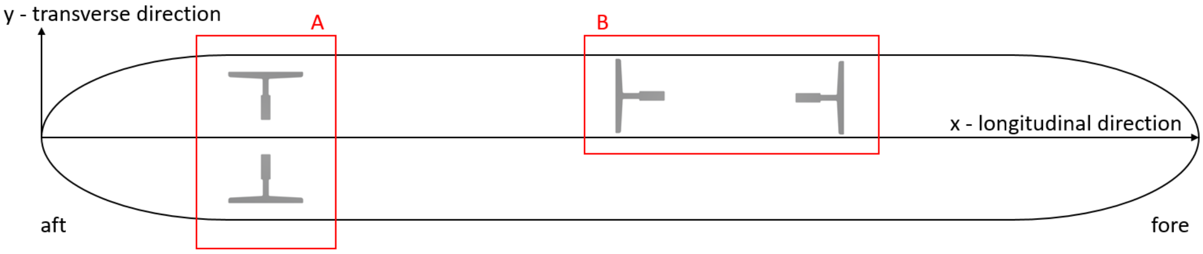
Operating/exceptional condition	Roll: ± 10°, with 10-second period. Pitch: ± 7°, with 5-second period.
Stowed condition	Roll: ± 22,5°, with 10-second period. Pitch: ± 7,5°, with 7-second period.

If ship characteristics are known, the angles and periods may be calculated using:

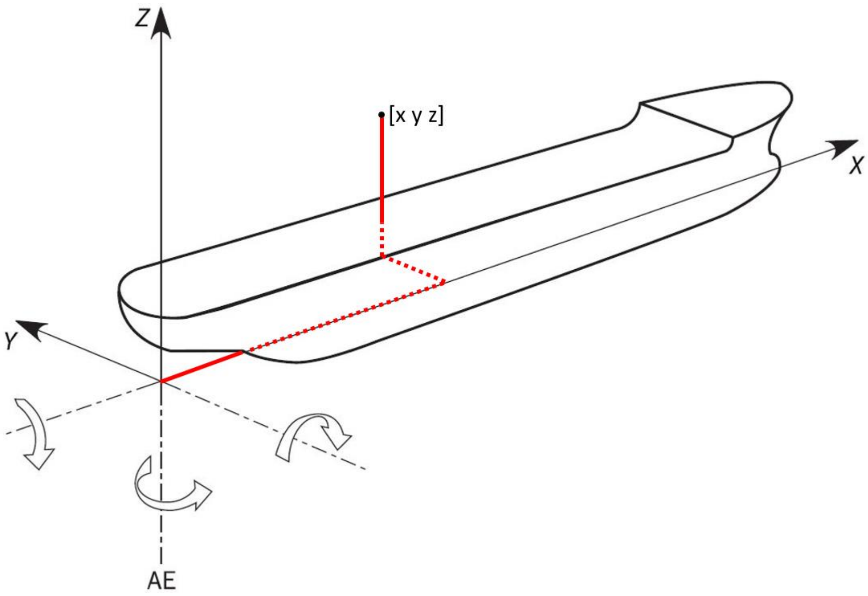
Chapter 4, 2.11.4 *Forces due to ship motion*
Table 4.2.2 *Ship motions*

1.2 Elevator specifications

(1) Select guide rail		T114/B	
(2) Select orientation of guide rail (see illustration)		A	
(3) Select car or counterweight		Car	
(4) Select condition		Exceptional	
(5) Select car size (or insert customized values)	A	1400x1600	mm^2
(6) Select % counterweight	q	0,5	

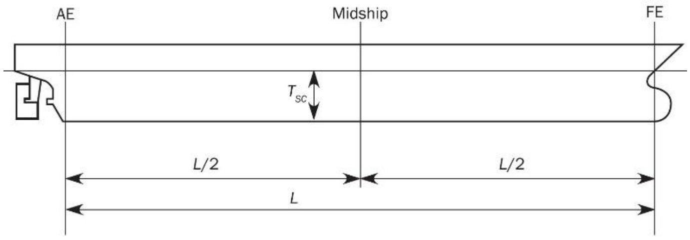


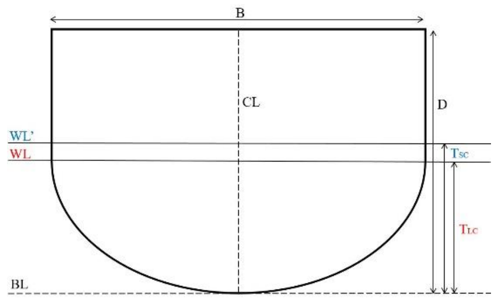
(7) Specify distance between brackets	l_k	1900	mm



(8) Specify x-coordinate of car/cwt from aft end	x	55,8	m
(9) Specify y-coordinate of car/cwt from centerline	y	2,5	m
(10) Specify z-coordinate of car/cwt from baseline	z	32	m

1.3 Ship dimensions and characteristics





(11) Specify length between p.p. – rule length	L_{pp}	133,5	m
(12) Specify moulded breadth	B	23,6	m
(13) Specify moulded depth	D	16	m
(14) Specify midship draught at loading condition	T_{LC}	5,5	m

2 Output

(15) Weight of car	W_{car}	**1200**	kg
(16) Rated load	W_{rated}	**1000**	kg
(17) Counterweight	W_{cwt}	**1700**	kg
(18) Design weight	W_{design}	**2200**	kg

(19) Roll angle	θ	**10,0**	deg
(20) Roll period	T_θ	**10,0**	s

(21) Pitch angle	ϕ	**7,5**	deg
(22) Pitch period	T_ϕ	**7,0**	s

3 Result

(23) Transverse force	F_{trans}	**3748**	N
(24) Longitudinal force	F_{long}	**2817**	N
(25) Vertical force	F_{vert}	**63 972**	N

(26) Bending moment from roll	Mb_{roll}	**890 073**	Nmm
(27) Bending moment from pitch	Mb_{pitch}	**334 521**	Nmm

(28) Bending stress from roll	σb_{roll}	**30,0**	MPa
(29) Bending stress from pitch	σb_{pitch}	**17,6**	MPa
(30) Combined bending stress	σb	**47,5**	MPa

(31) Compression stress	σc	**30,6**	*MPa*

(32) Deflection from roll	δ_{roll}	**0,7**	mm
(33) Deflection from pitch	δ_{pitch}	**0,4**	mm

(34) Safety criteria, buckling + bending	SC	**0,40**	≤	**0,85**
(35) Safety factor, bending	SF	~~1,00~~	≥	**1,00**

4 Technical background

	σ_y	I_x	I_y	W_x	W_y	A	σ_k
T70/A	225	409500	188600	9169	5389	940	140
T70-70-9/A	225	528100	246200	10790	7020	1125	140
T75/A	225	402900	264700	9286	7060	1091	140
T80-80-9/A	225	802000	388300	14210	9700	1356	140
T82/A	225	493100	301700	10270	7358	1091	140
T89/A	225	598300	524100	14350	11780	1577	140
T90/A	225	1020000	524800	20860	11660	1725	140
T125-L1/A	225	564600	1078800	10970	11260	1439	140
T75/B	265	402900	264700	9286	7060	1091	210
T78/B	265	299200	263900	7564	6766	985	210
T82/B	265	493100	301700	10270	7358	1091	210
T89/B	265	598300	524100	14350	11780	1577	210
T90/B	265	1020000	524800	20860	11660	1725	210
T114/B	265	1793000	1086000	29700	19050	2089	210
T125/B	265	1510000	1591000	26160	25460	2282	210
T127-1/B	265	1879000	1499000	30650	23610	2274	210
T127-2/B	265	2017000	2299000	31170	36200	2872	210
	MPa	mm^4	mm^4	mm^3	mm^3	mm^2	MPa

Size	Area	Capacity	Weight	CWT
1100x1600	1,76	1000	1200	1700
1400x1600	2,24	1000	1200	1700
1100x2100	2,31	1000	1200	1700
1525x1525	2,33	1000	1400	1900
1400x2000	2,80	1200	1300	1900
1370x2030	2,78	1200	1600	2200
1600x2100	3,36	1500	1380	2130
1800x2100	3,78	1800	1450	2350
1800x2300	4,14	2000	1500	2500
Customized	-	*1200*	*1600*	*2200*
	m^2	kg	kg	kg

5 Calculation procedure

Transverse metacentric height	**GM**	1,65	m
Moulded breadth of ship	**B**	23,6	m
Vertical coordinate of ship's rotation center	**z**	6,8	m

Roll factor	θ	0,4	
Roll angle (NB! $22 \leq \phi \leq 30$ in stowed condition)	ϕ	26,2	deg
Roll period	T_r	12,9	s
Pitch angle (NB! $\psi \leq 8$)	ψ	7,7	deg
Pitch period	T_p	5,8	s
Heave amplitude	**h**	1,7	m
Heave period	T_h	5,8	s

Source	F_{vert} **(normal)**	F_{trans} **(parallel)**	F_{long} **(parallel)**	
Static roll	10 561	5200	0	N
Static pitch	11 666	0	1575	N
Static combined	11 106	3757	1120	N
Dynamic roll	328	3313	0	N
Dynamic pitch	-814	0	4811	N
Dynamic heave roll	2112	1040	0	N
Dynamic heave pitch	2333	0	315	N

Roll motion only	F_T	9552	N
Pitch motion only	F_L	6701	N
Combined motion	F_N	14 273	N
Static roll motion only	FS_T	5200	N
Static pitch motion only	FS_L	1575	N
Static combined motion	FS_N	11 106	N

Buckling stress:

Radius of gyration	**r**	29,3	mm
Robertson's constant	**a**	5,5	
End constraint condition	**K**	1,0	
Slenderness ratio	**s**	65	
Buckling constant	**n**	0,3	
Critical stress	σ_e	483,4	MPa
Critical compression stress	σ_c	158,7	MPa

6 References

Ship motions, accelerations and forces are calculated from:

[1] Lloyd's Register's Code for Lifting Appliances in a Marine Environment (July 2016)

Specifications for the guide rails are from:

[2] Product catalog, Savera Elevator System Solutions, Savera Group

Bending moments, bending stresses and deflections are calculated based on elementary theory of simply supported beams.

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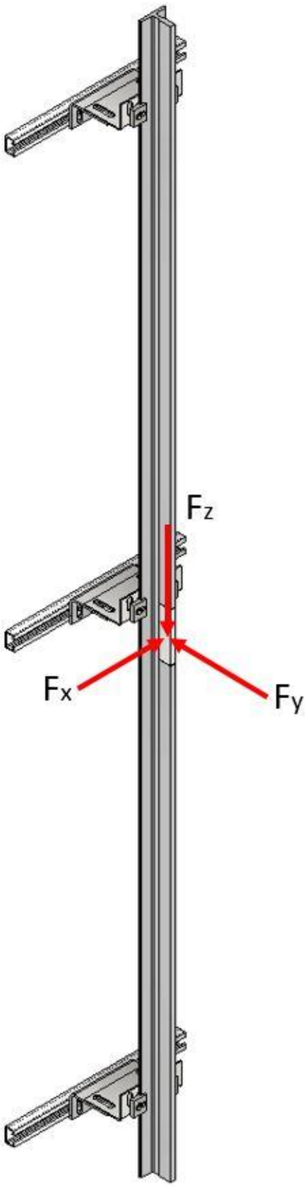
APPENDIX D

Standard bracket solution

Structural analysis in ANSYS

1 Introduction

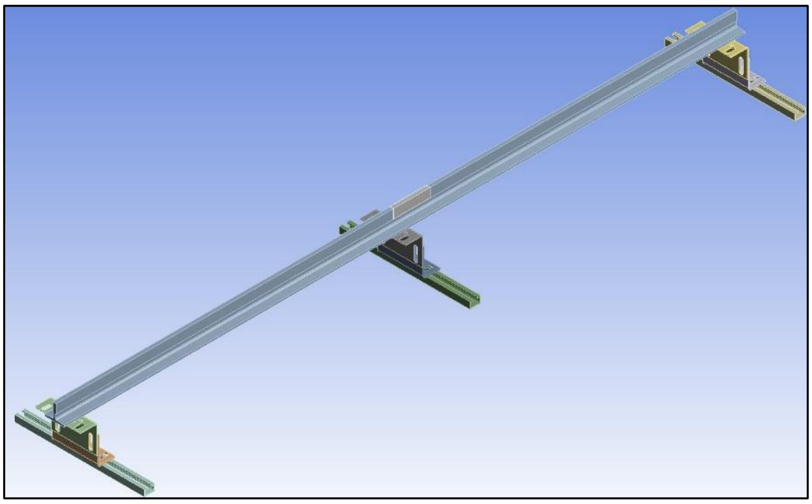
1.1 Applied loads



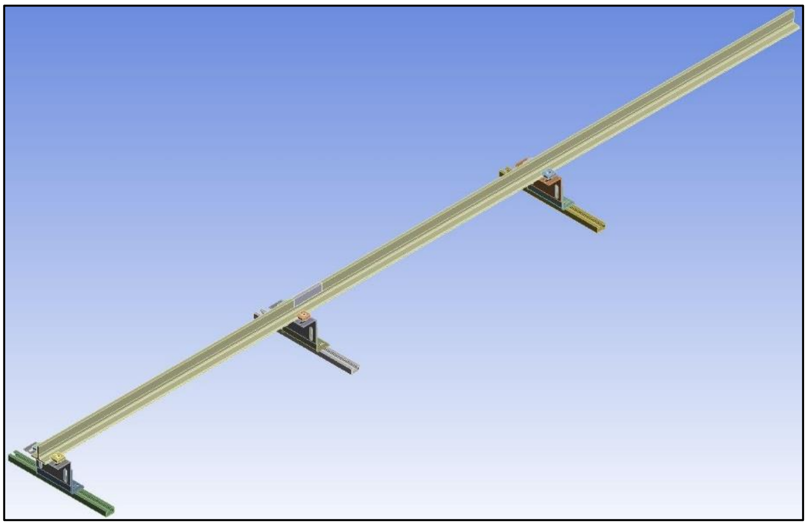
Direction	***Force***	***Unit***
Normal to trunk, x-direction	$F_x = 11$	kN
Parallel to trunk, y-direction	$F_y = 5{,}5$	kN
Parallel to trunk, z-direction	$F_z = 60$	kN

1.2 FEM models

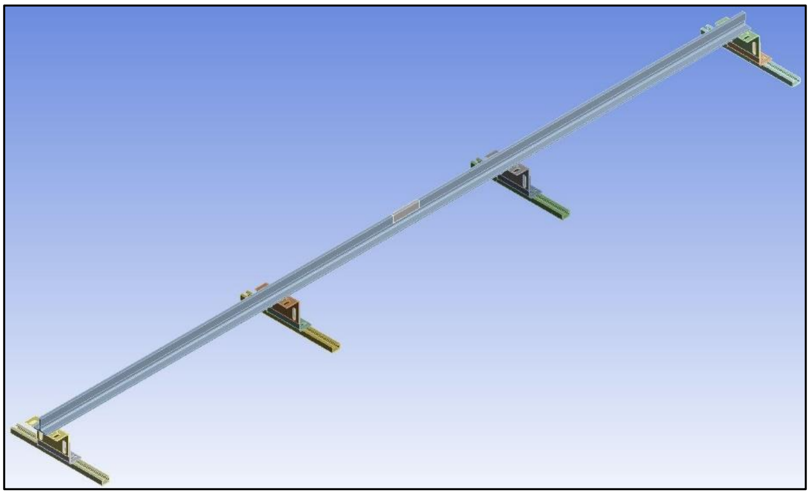
Load case 1 Force acting on one bracket, x-direction and y-direction



Load case 2 Force acting on one bracket caused by safety gear, z-direction

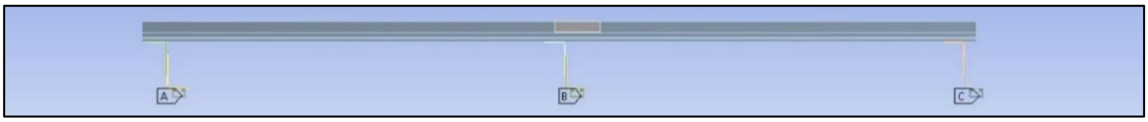


Load case 3 Force acting between two brackets, x-direction and y-direction



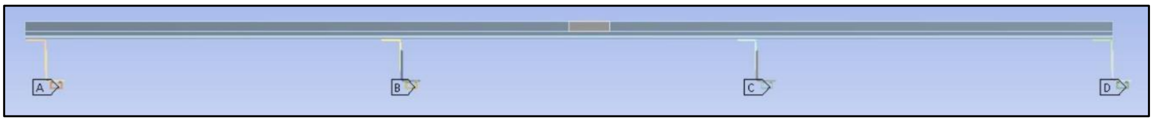
1.3 Boundary conditions

Load case 1

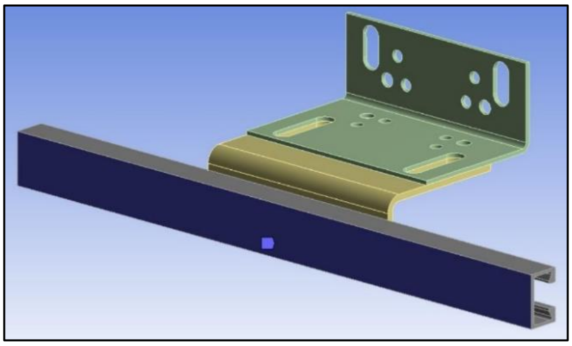


Load case 2

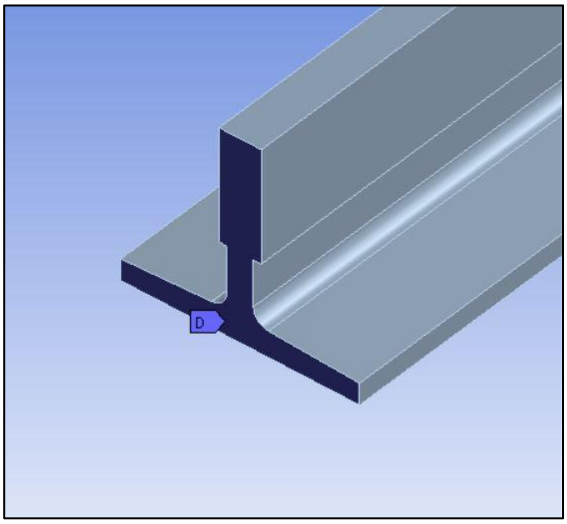
Load case 3



Brackets

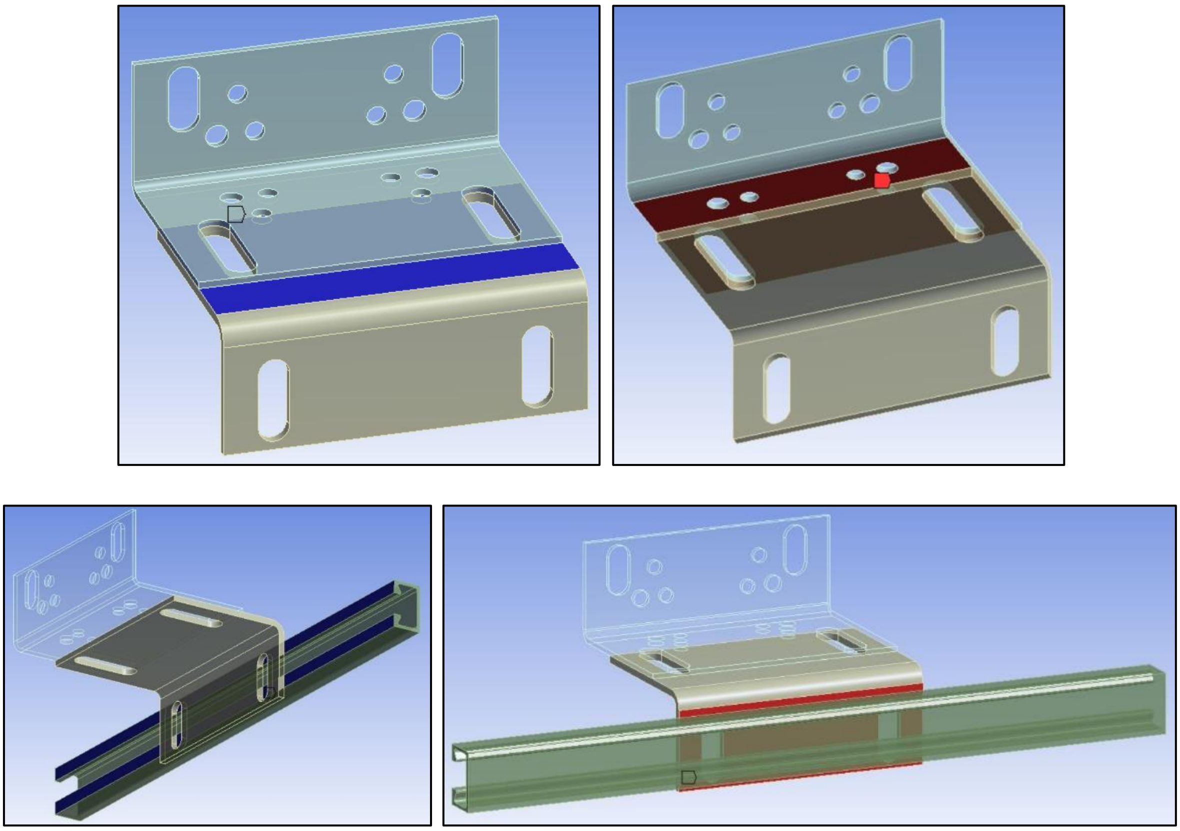


Baseplate (for load case 2)

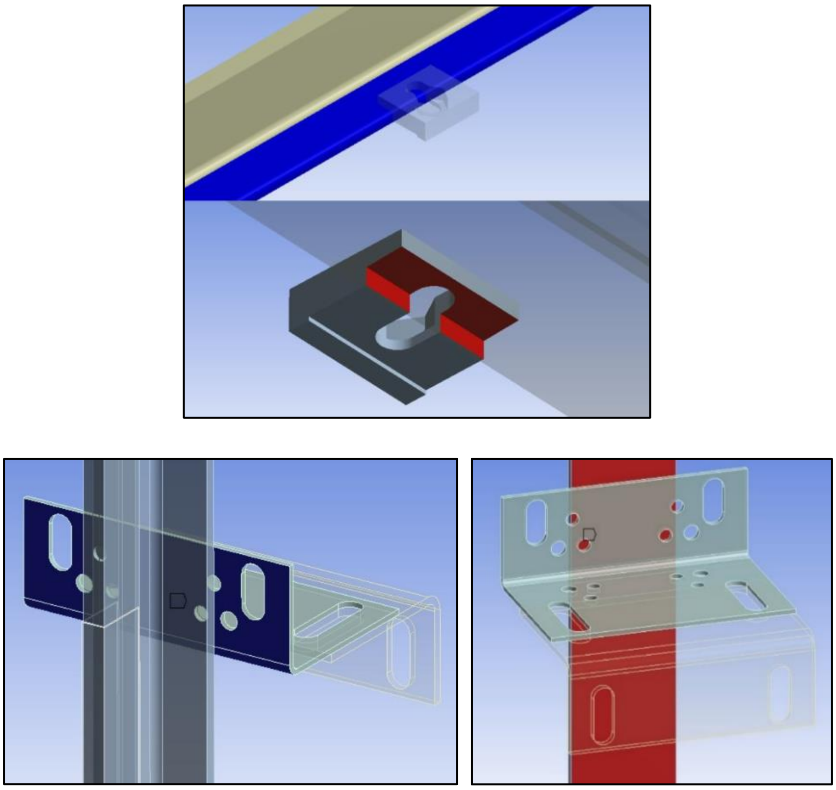


1.4 Contacts

Bonded contact for all load cases

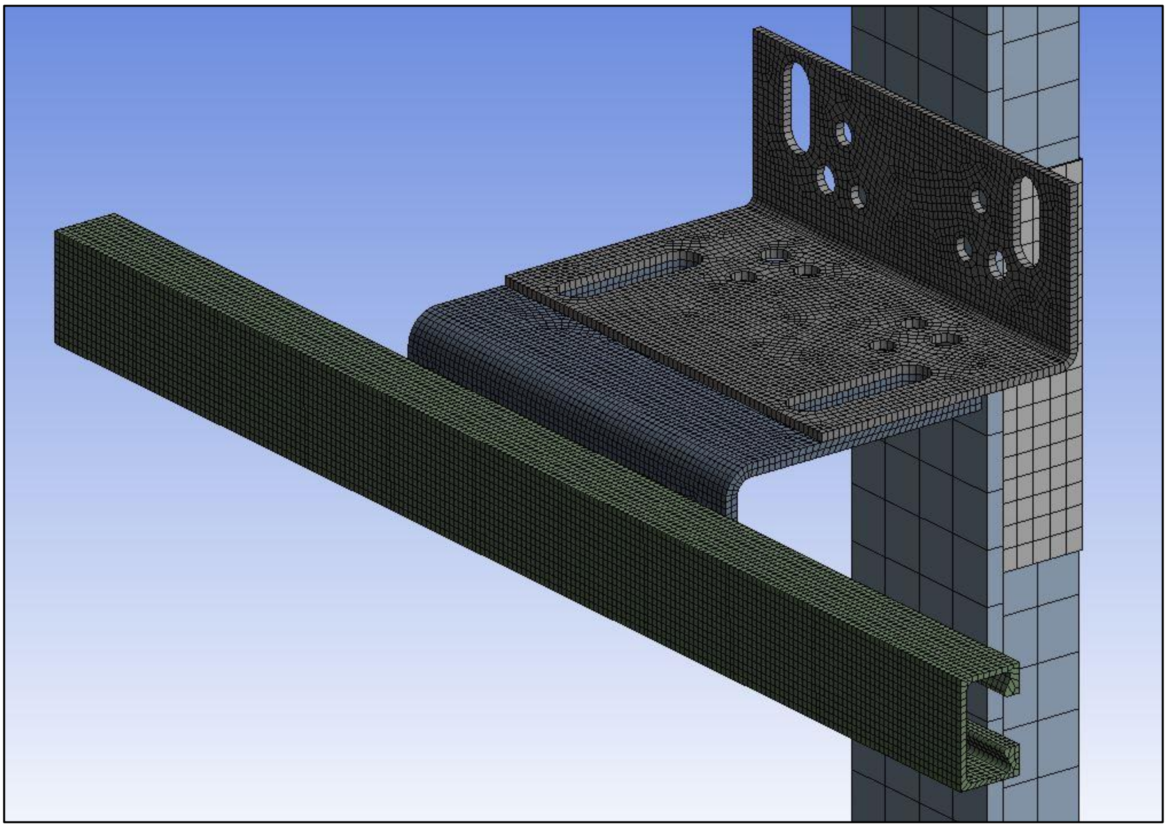


Frictional contact for the following parts in load case 2

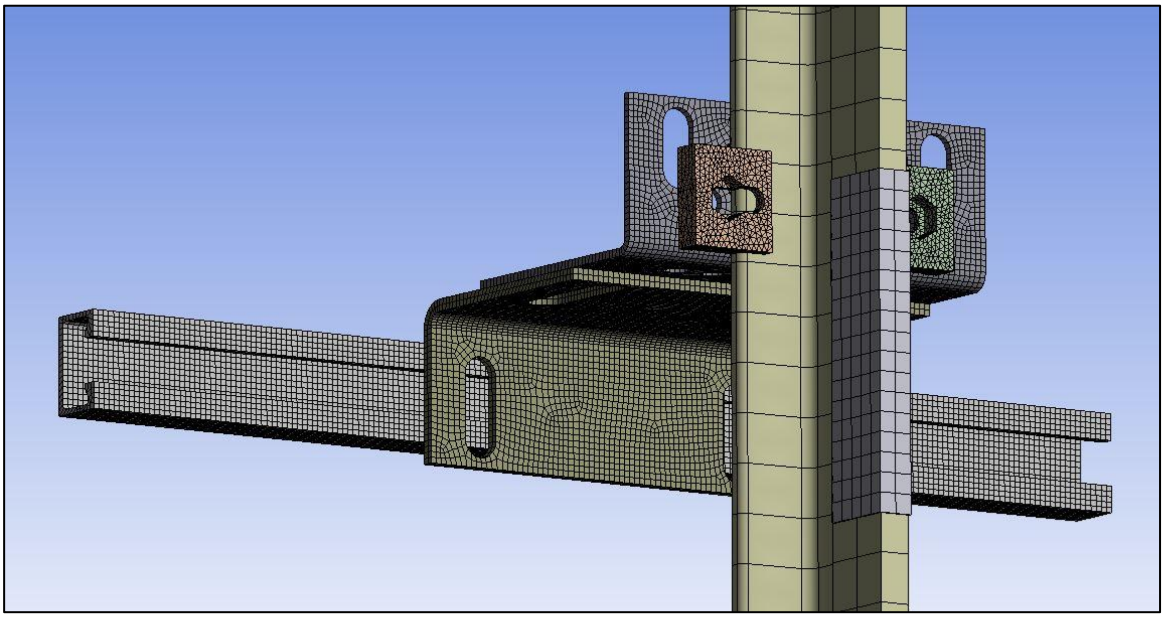


1.5 Mesh

Hex meshed bracket (L-profiles, Halfen channel), guide rail and guide shoe



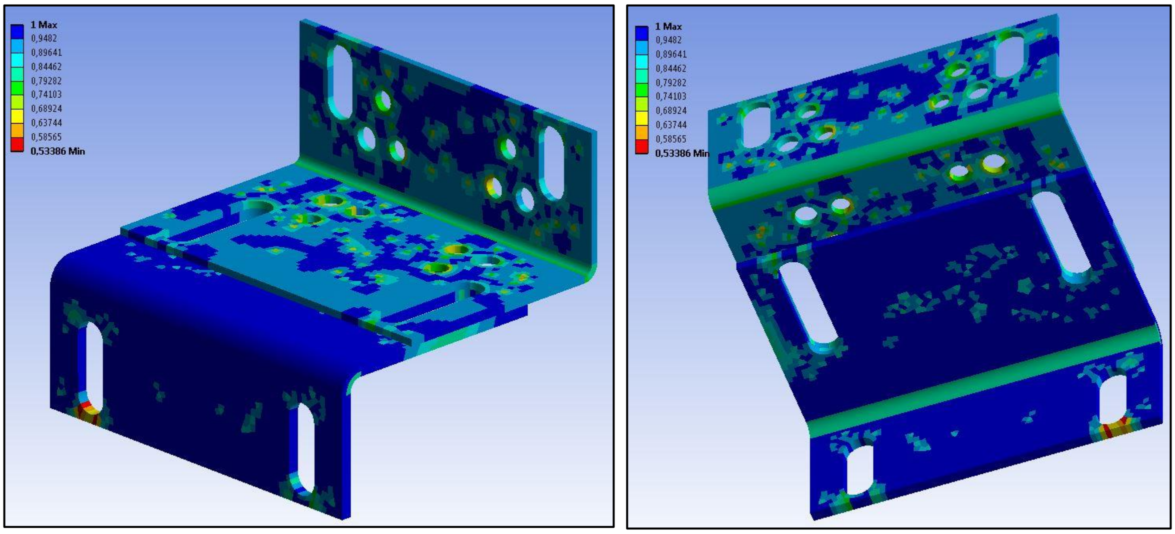
Tetra meshed rail clips



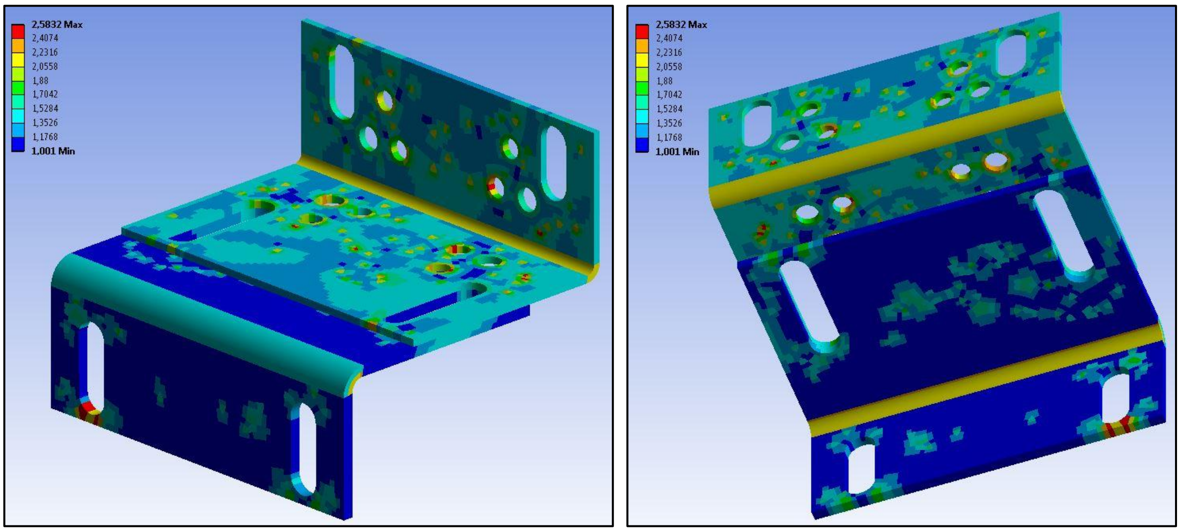
Model	***Nodes***	***Elements***
Load case 1	207 269	33 415
Load case 2	240 717	51 219
Load case 3	362 867	57 749

Components	***Mesh type and method***	***Element size***
Evaluated bracket(s)	Hex mesh, MultiZone	3 mm
Other brackets	Hex mesh, MultiZone	10 mm
Guide shoe	Hex mesh, Hex Dominant	10 mm
Guide rail	Hex mesh, Hex Dominant	25 mm
Rail clips (for load case 2)	Tetra mesh	3 mm

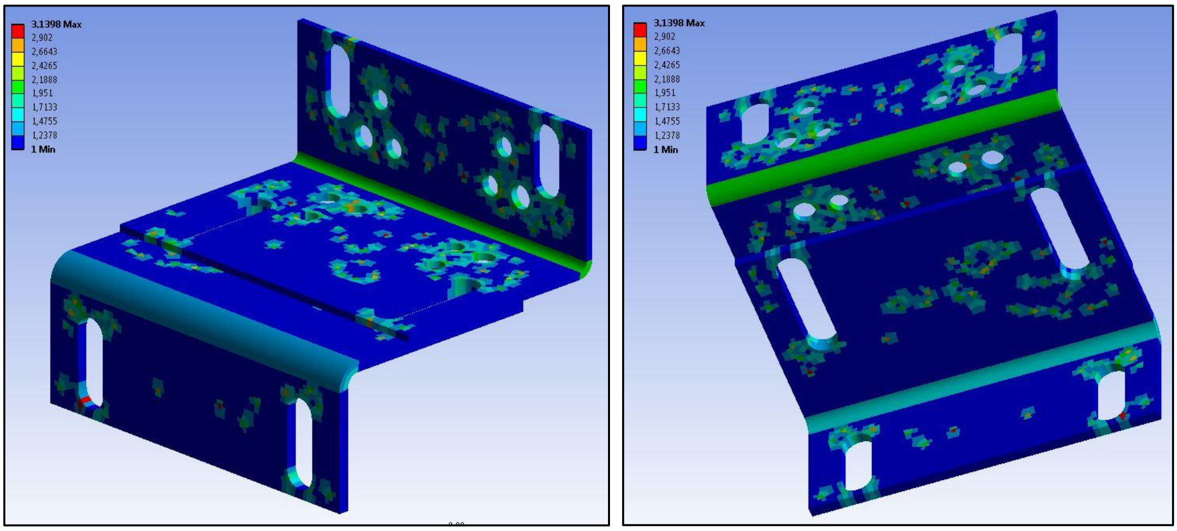
Element quality, bracket



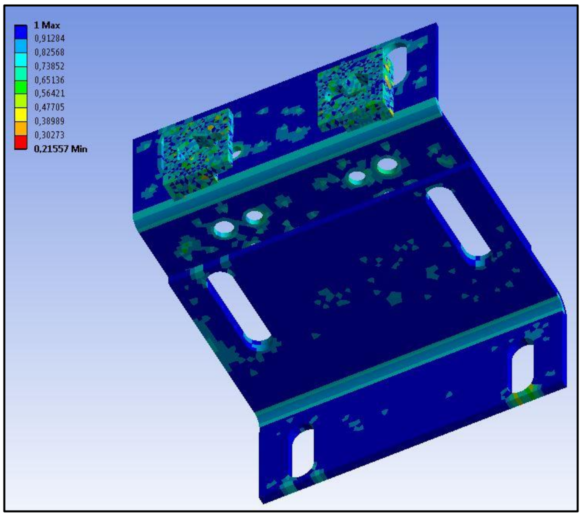
Aspect ratio, bracket

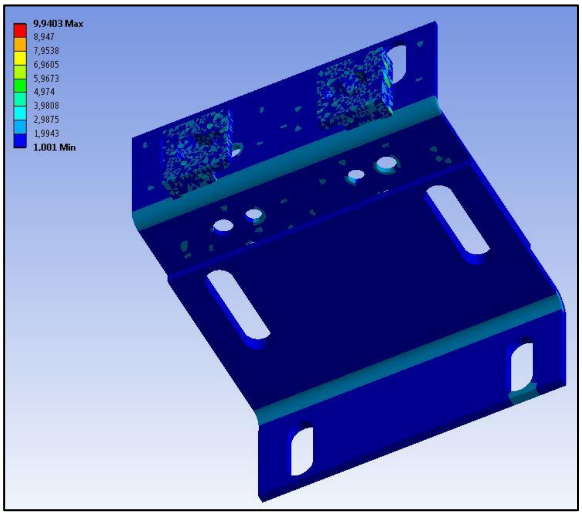


Jacobian ratio, bracket

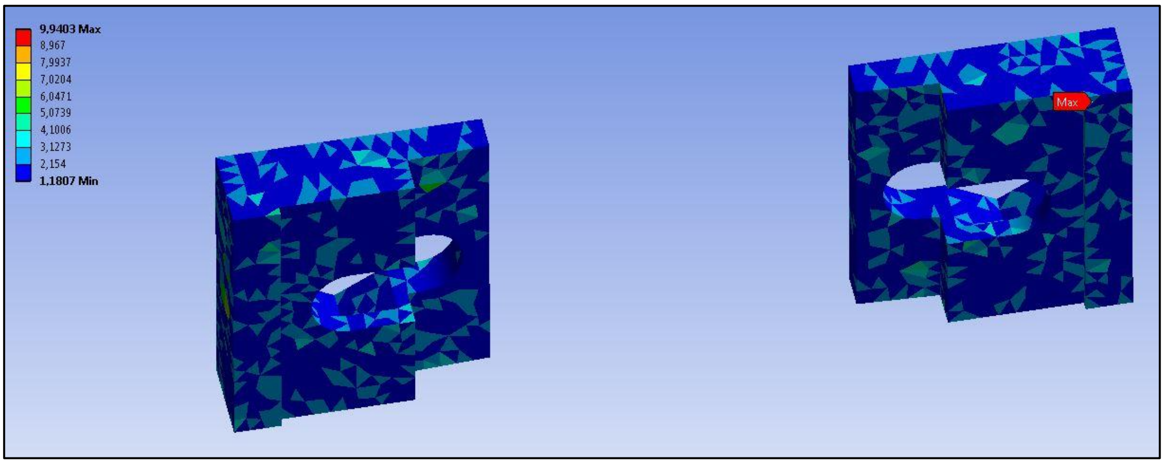


Element quality, rail clips

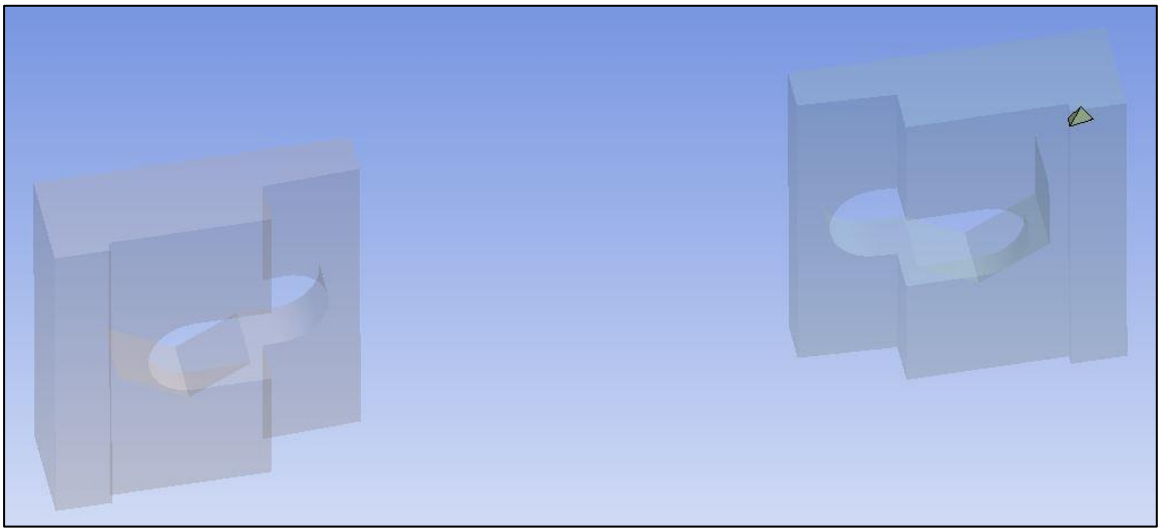




Aspect ratio, rail clips



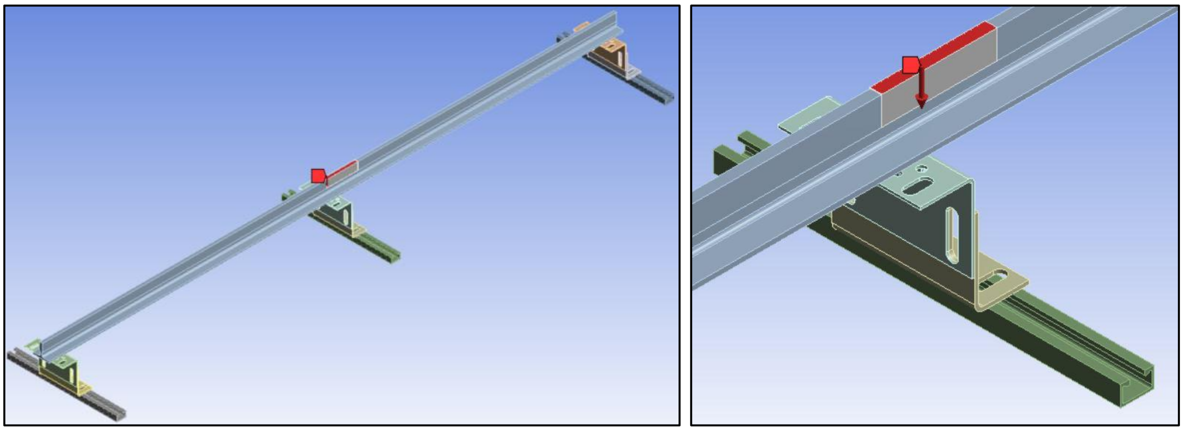
Element with max. aspect ratio



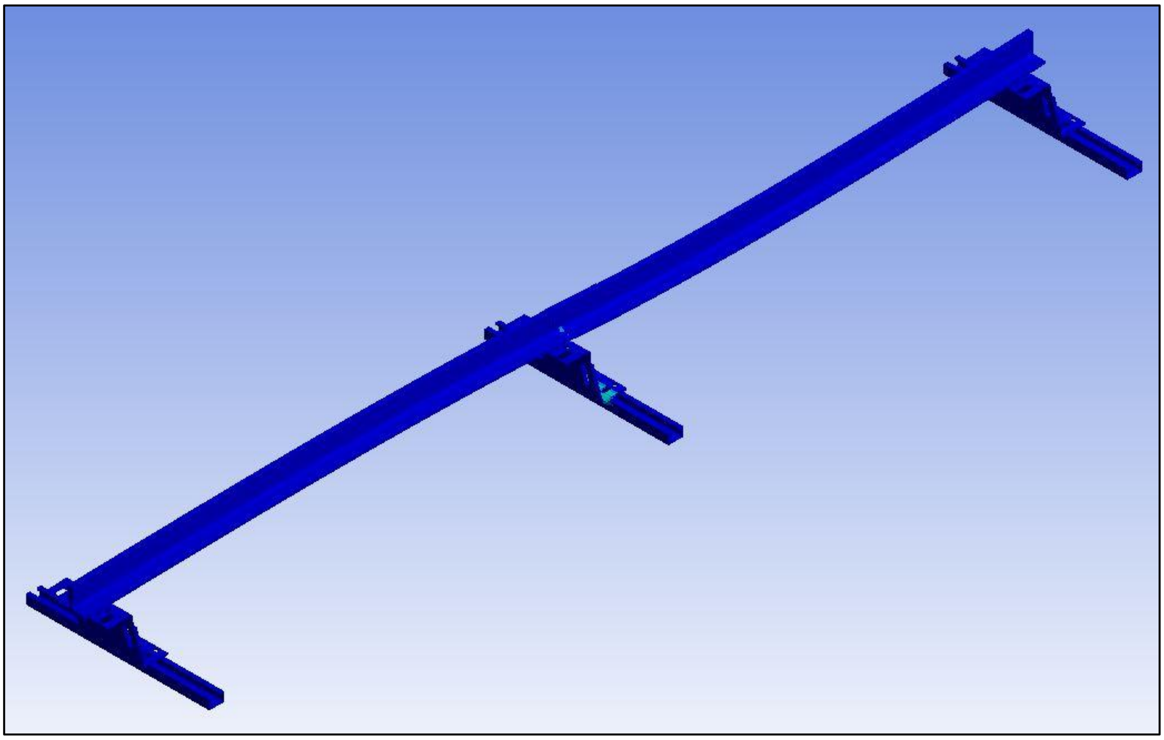
2 Forces acting on one bracket

2.1 Force in y-direction

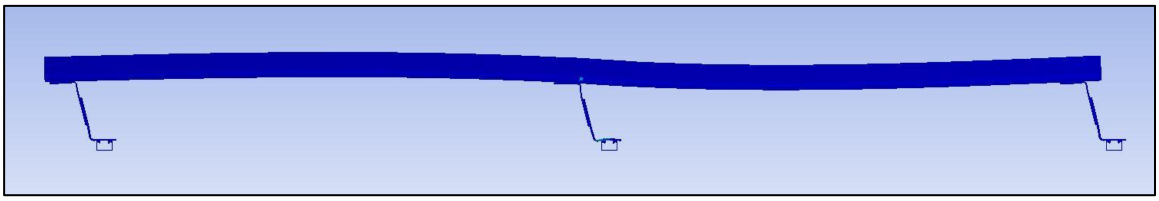
Applied force, F_y = 11 kN



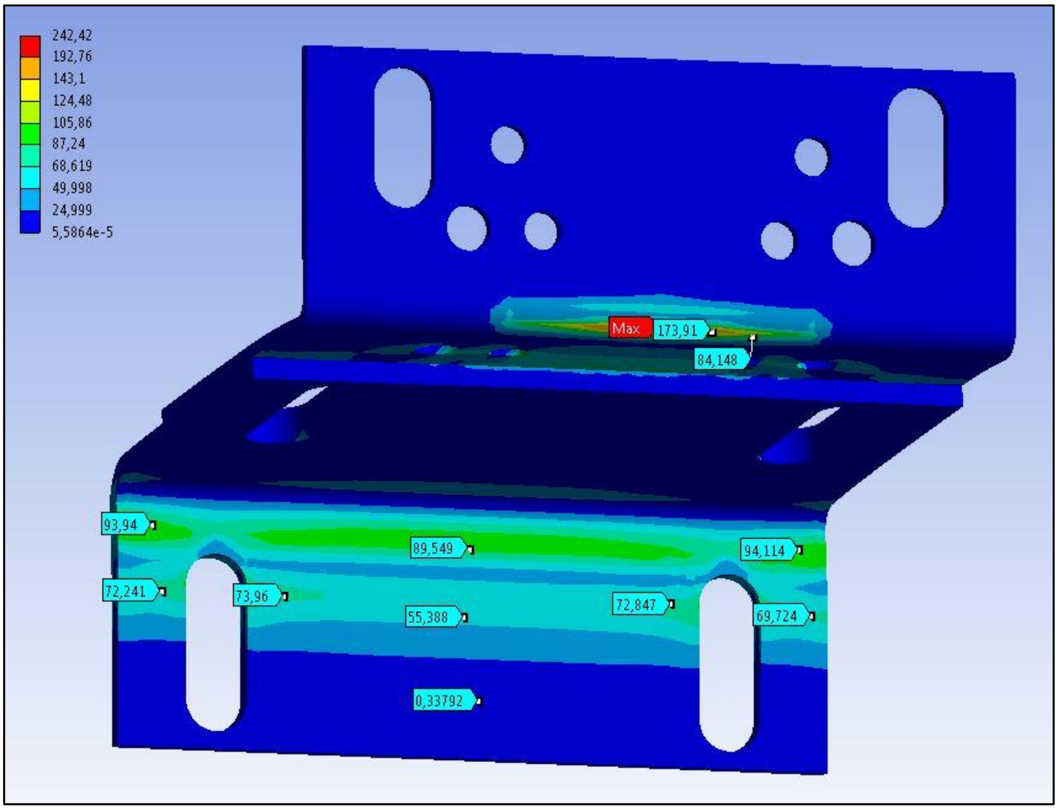
Deformation, isometric view



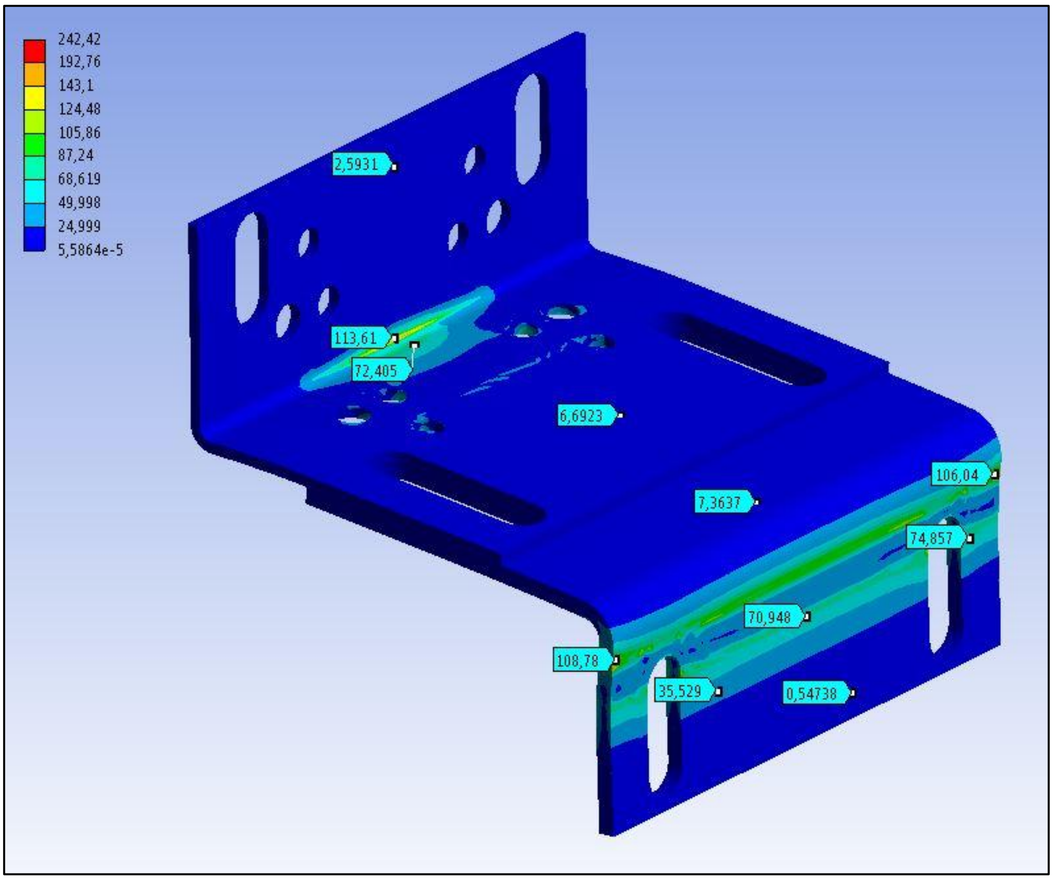
Deformation, side view



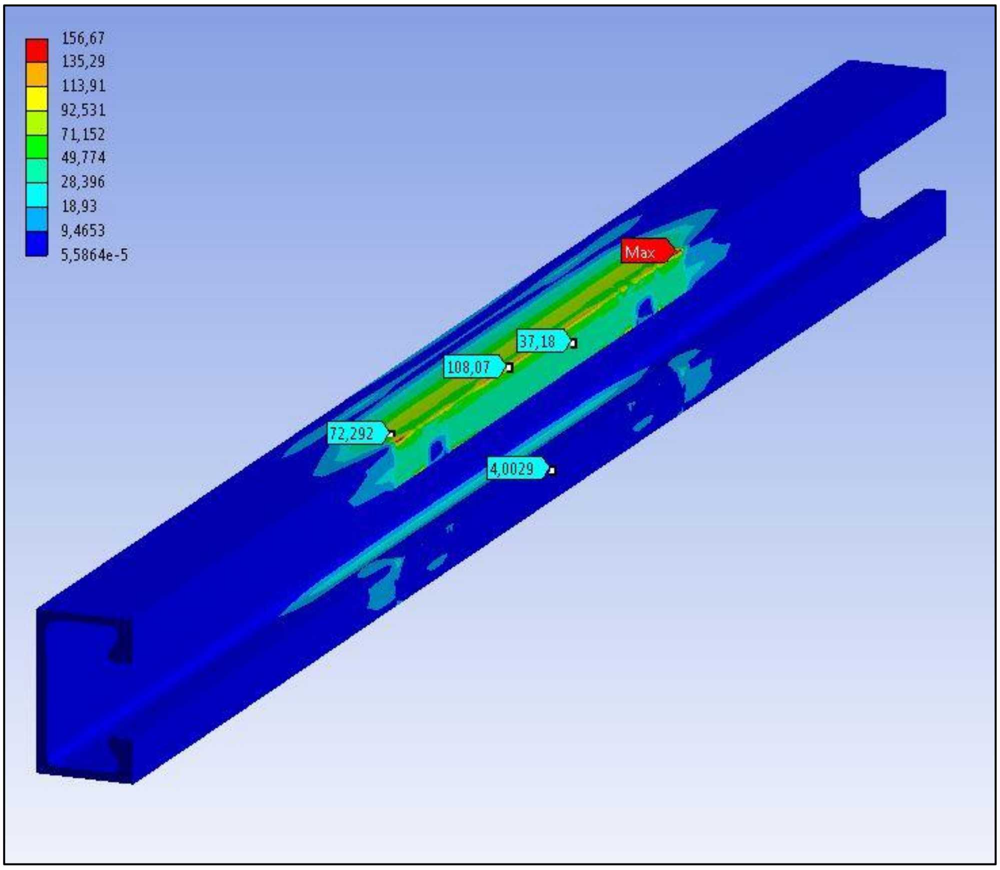
Equivalent stress in MPa, bracket



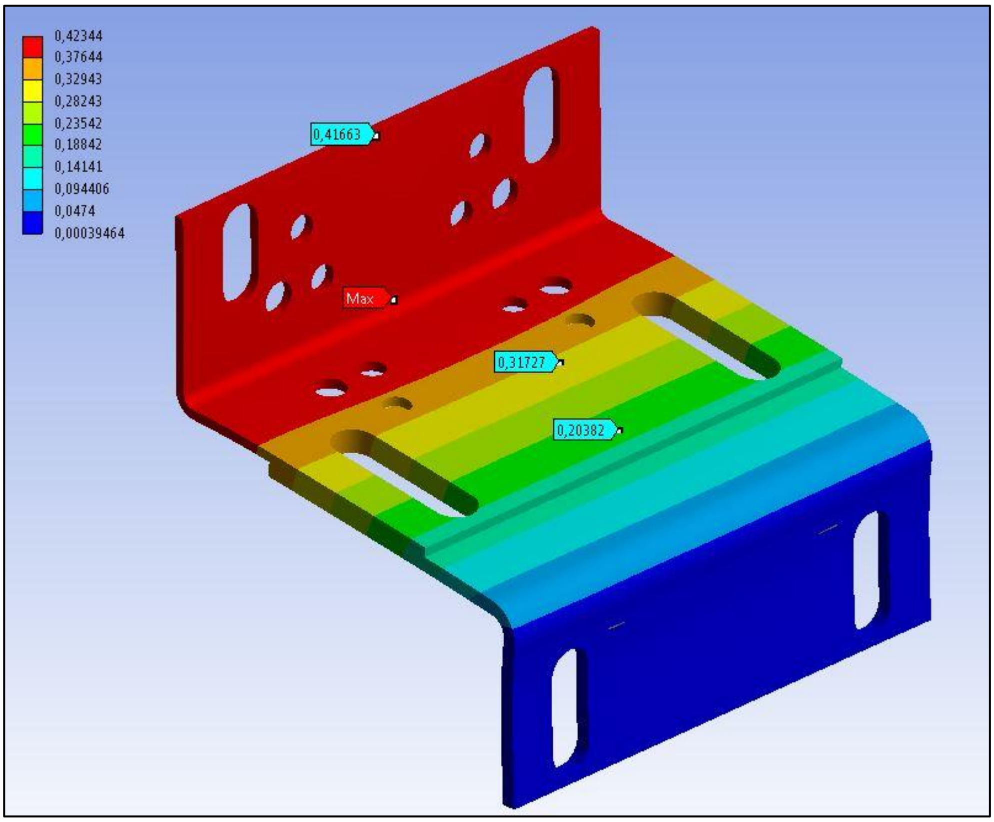
Equivalent stress in MPa, alternative view



Equivalent stress in MPa, Halfen channel

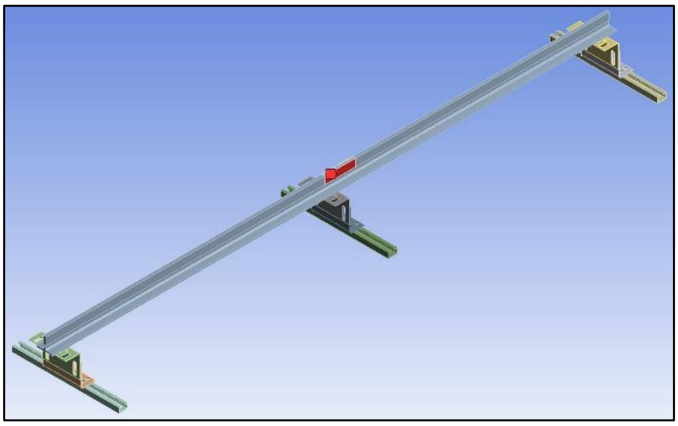


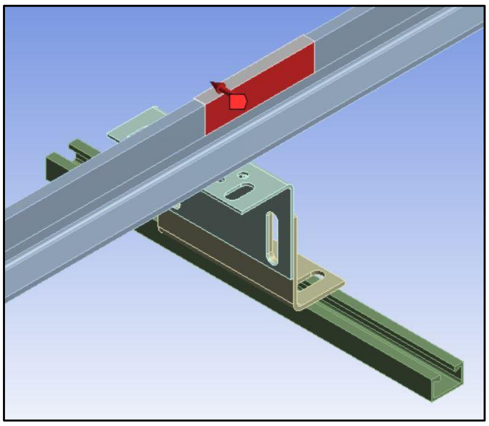
Deformation in mm, bracket



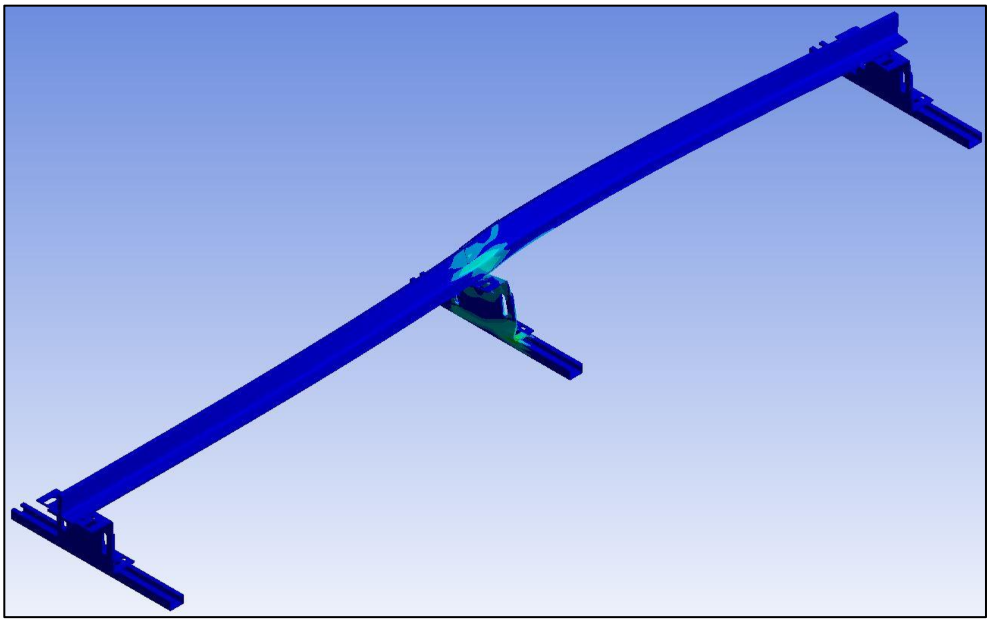
2.2 Force in x-direction

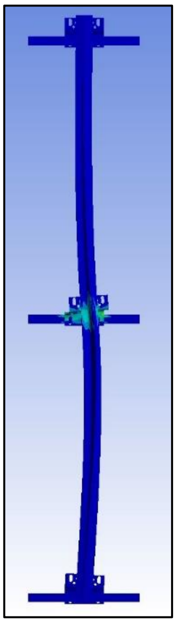
Applied force, F_x = 5,5 kN



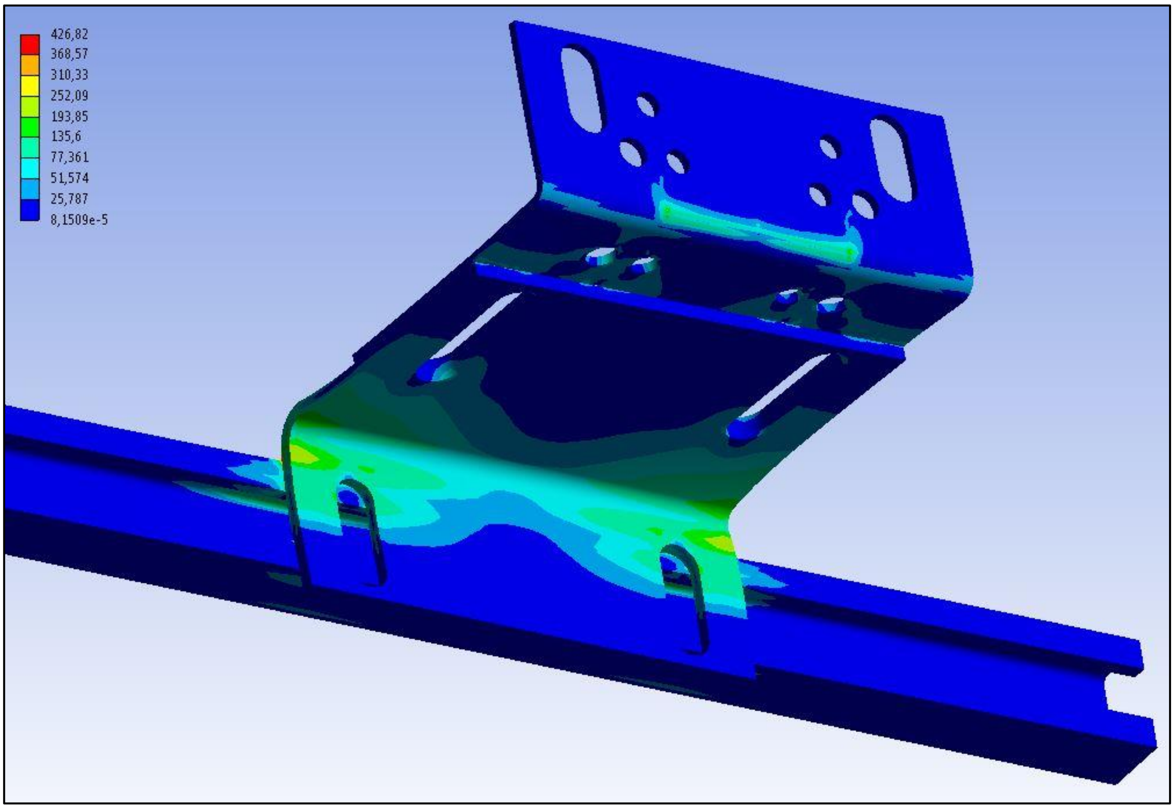


Deformation, isometric and front view

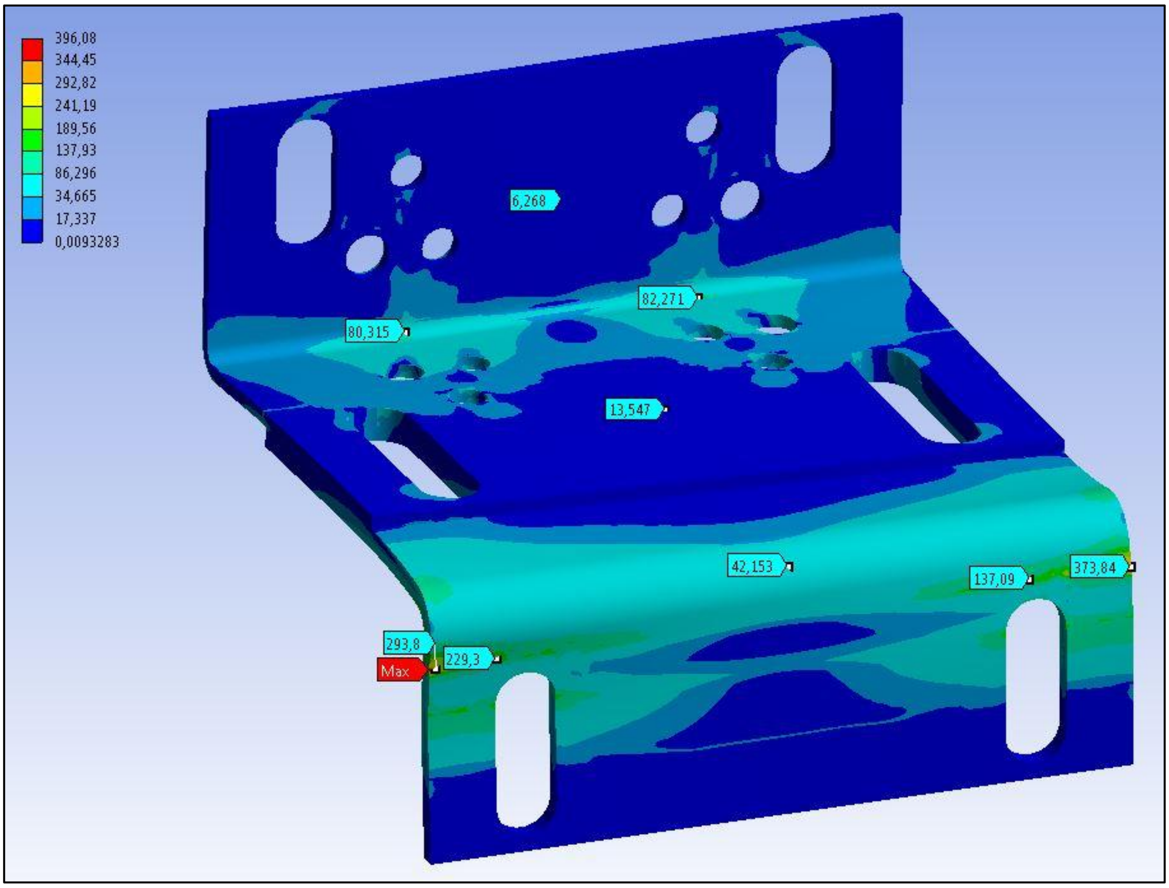




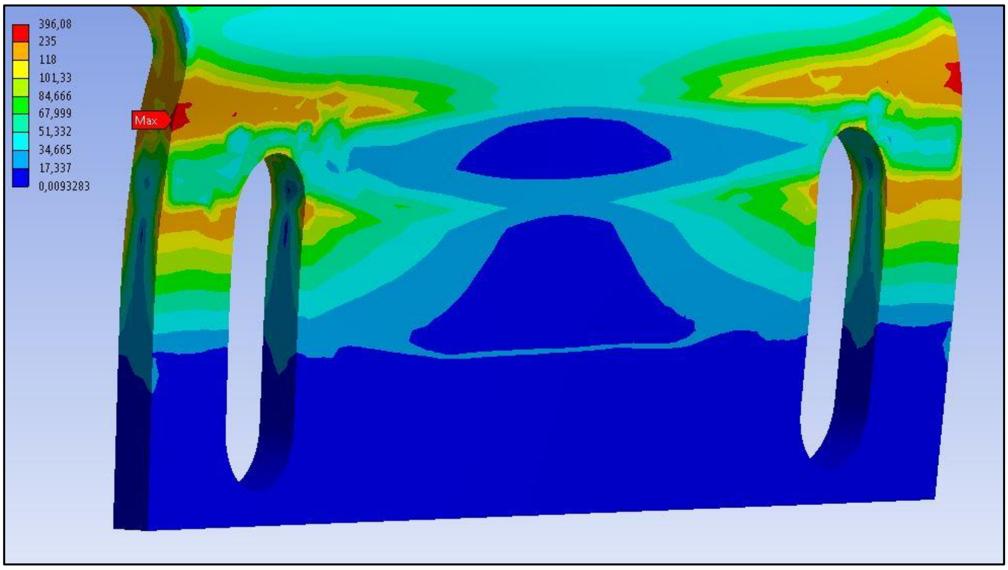
Equivalent stress in MPa, bracket



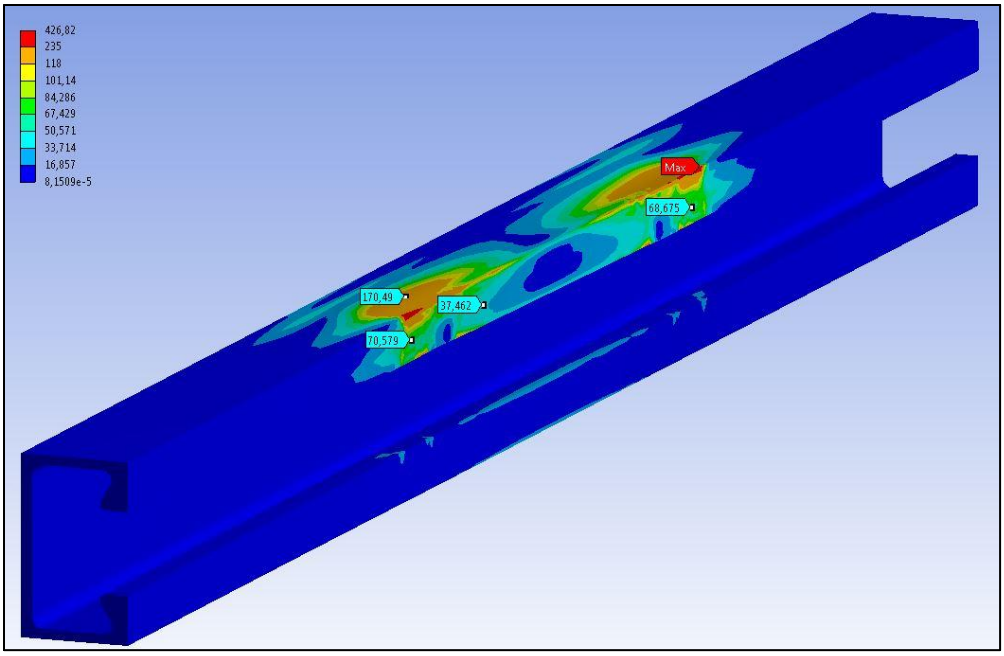
Equivalent stress in MPa, without Halfen channel



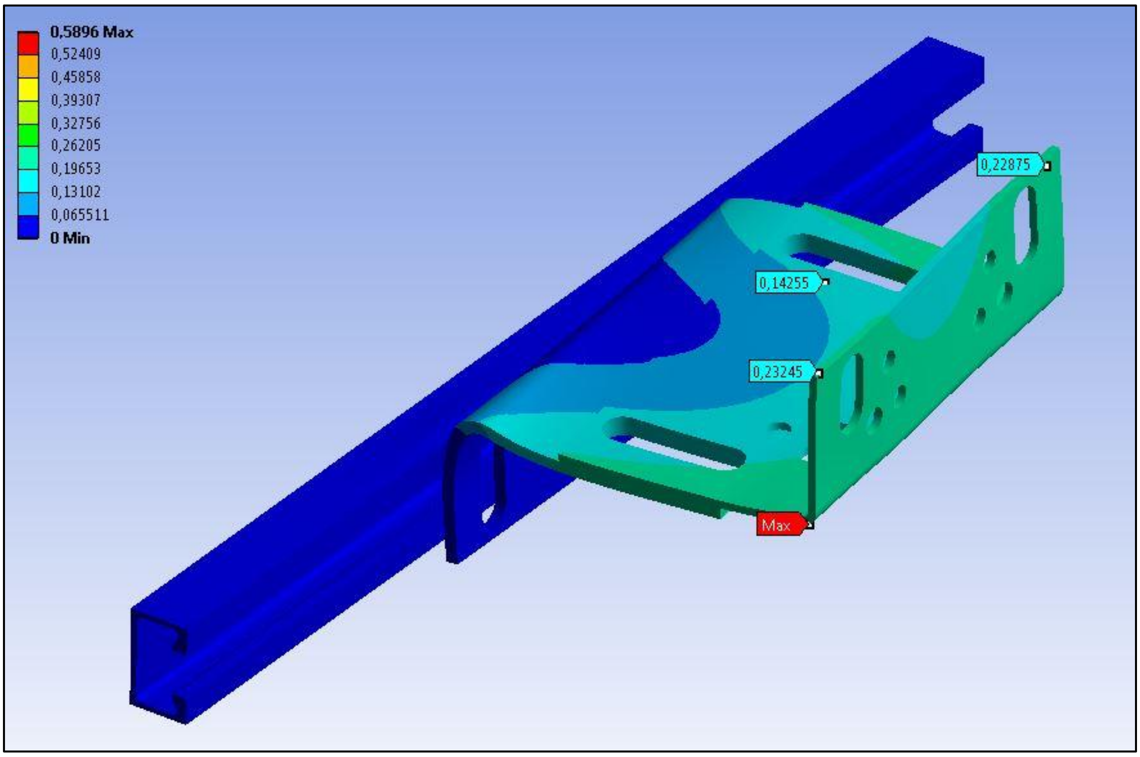
Equivalent stress in MPa, local analysis



Equivalent stress in MPa, Halfen channel

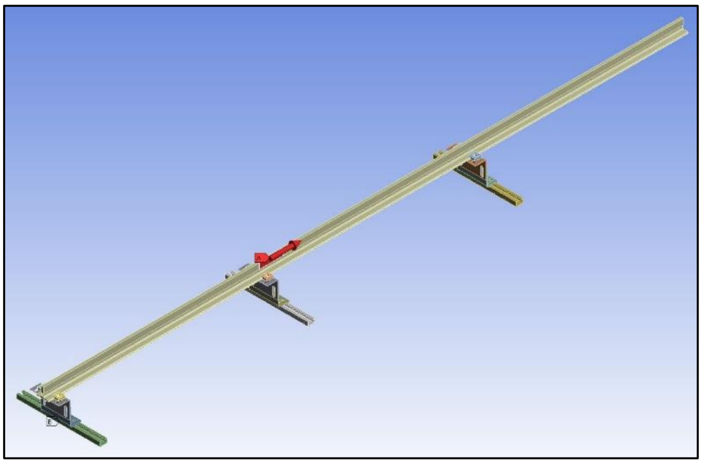


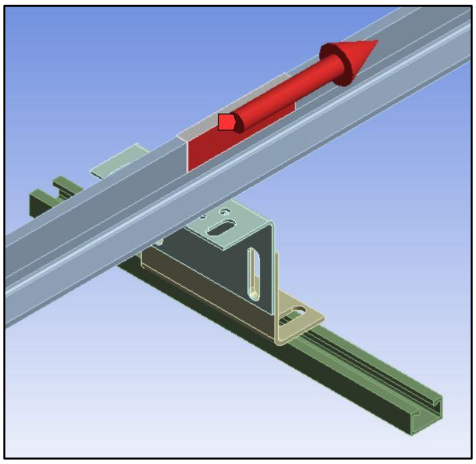
Displacement in mm, bracket



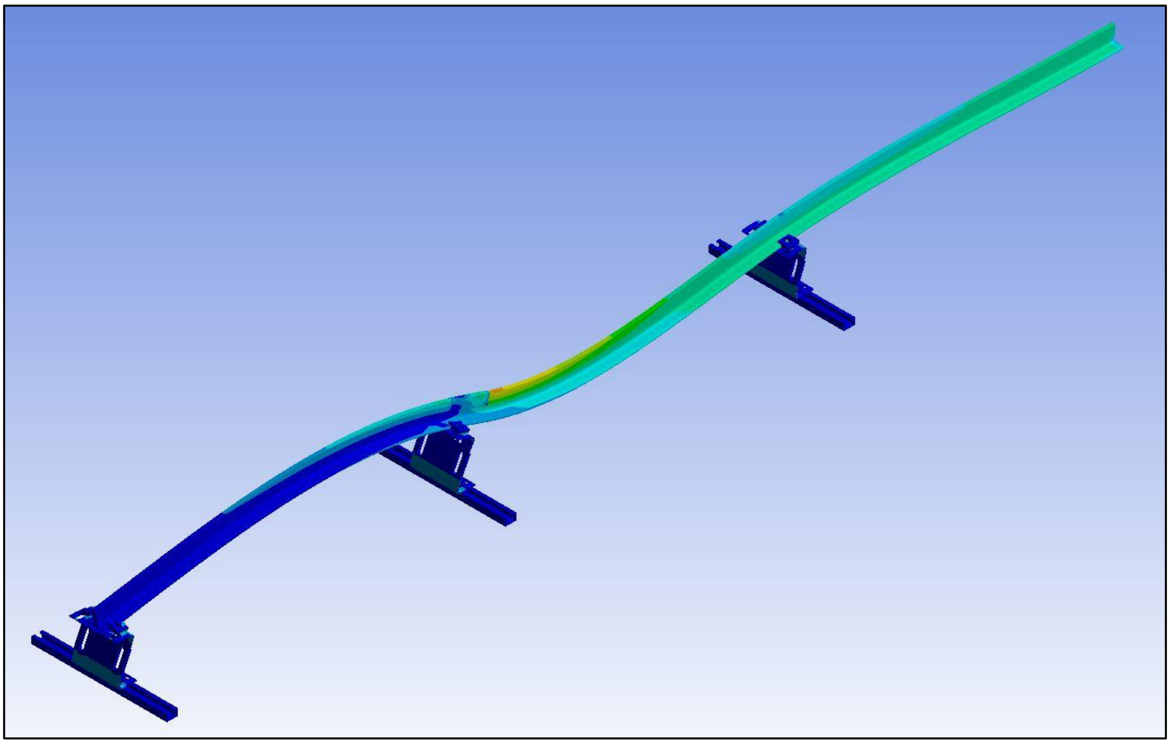
2.3 Force in z-direction

Applied force, F_z = 60 kN

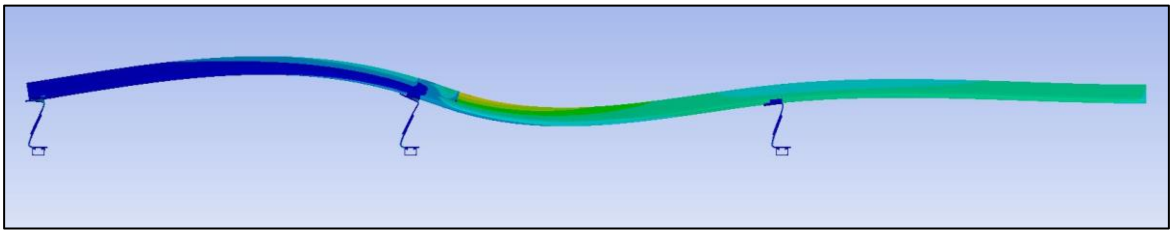




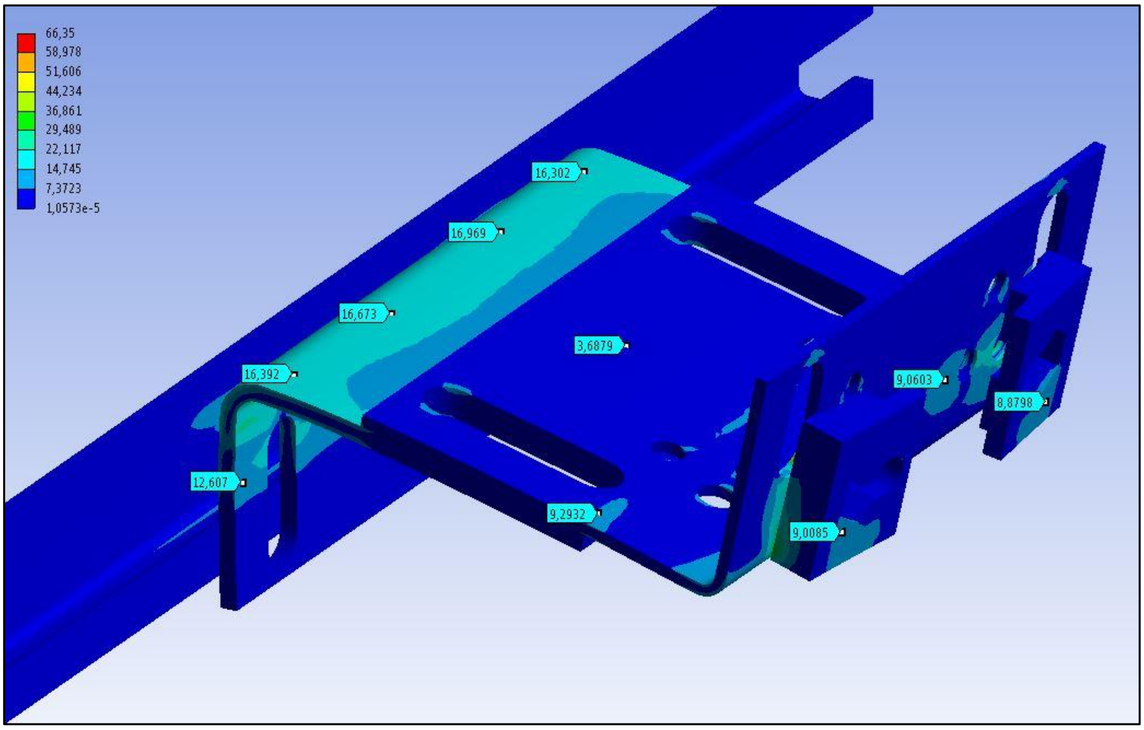
Deformation, isometric view



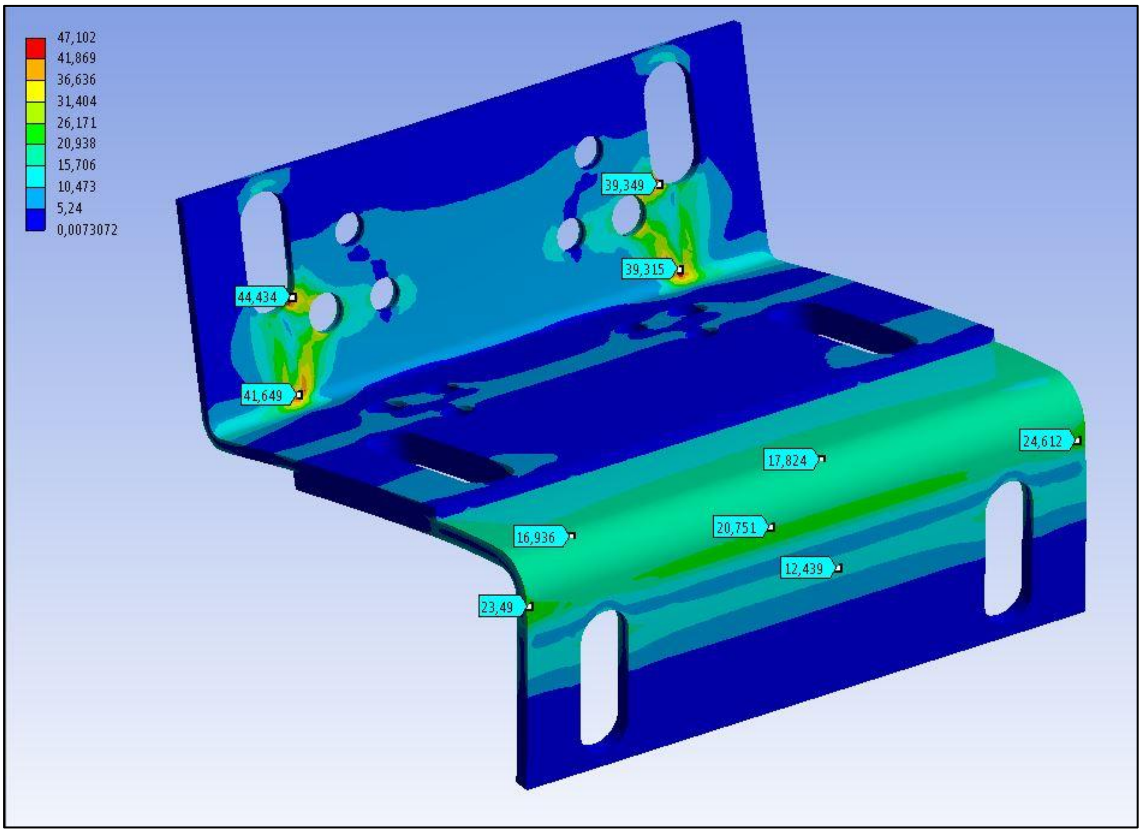
Deformation, side view



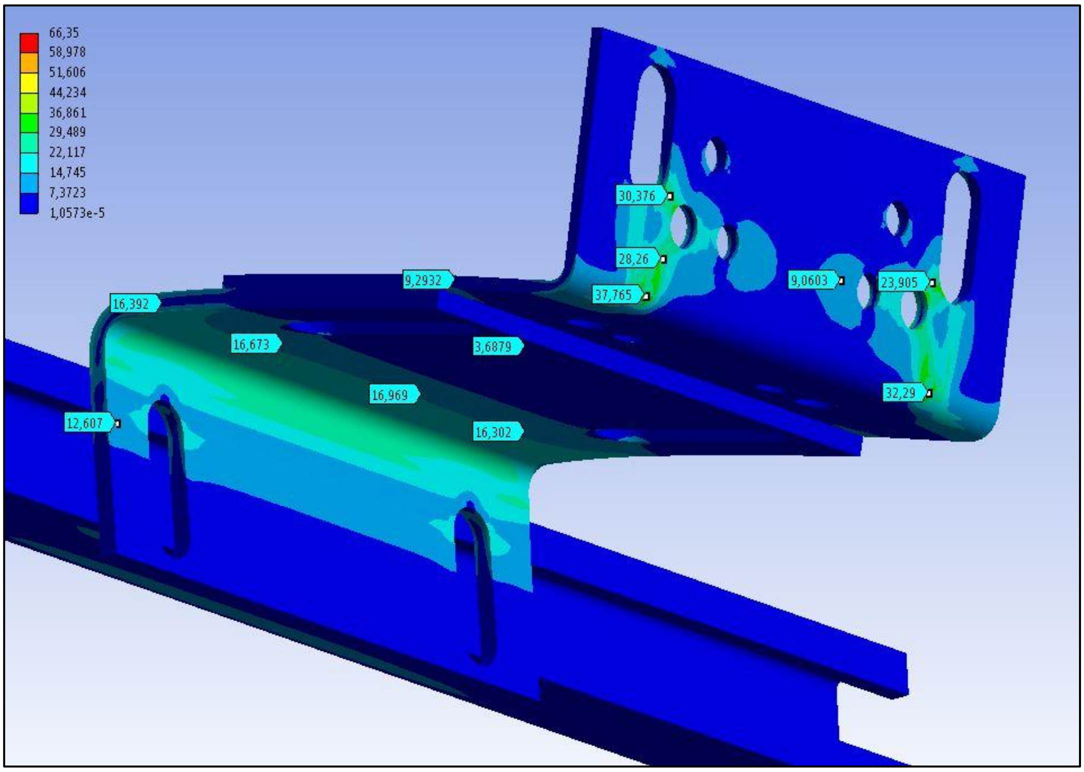
Equivalent stress in MPa, bracket



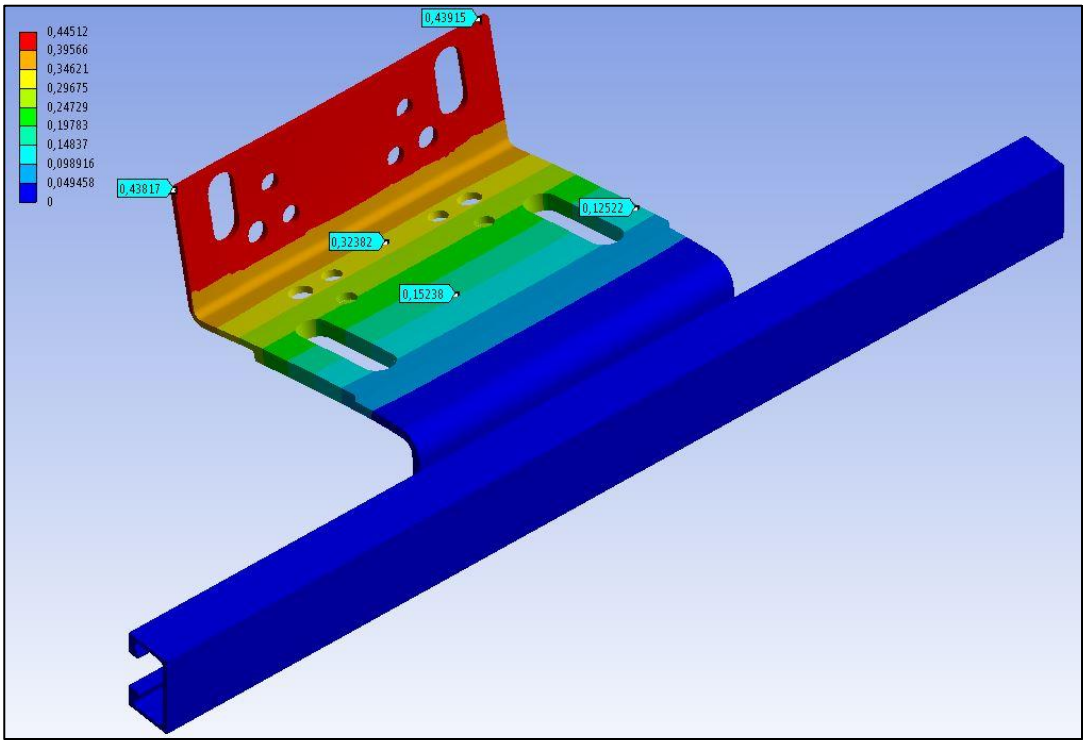
Equivalent stress in MPa, without Halfen channel



Equivalent stress in MPa, alternative view



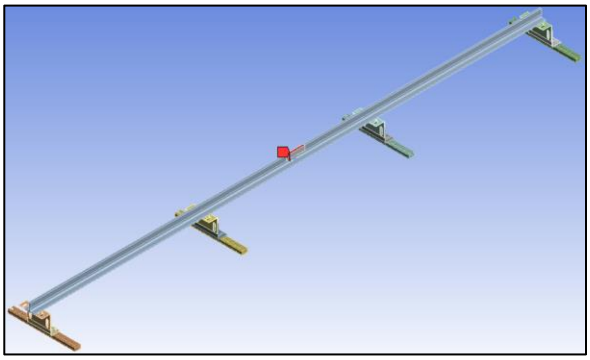
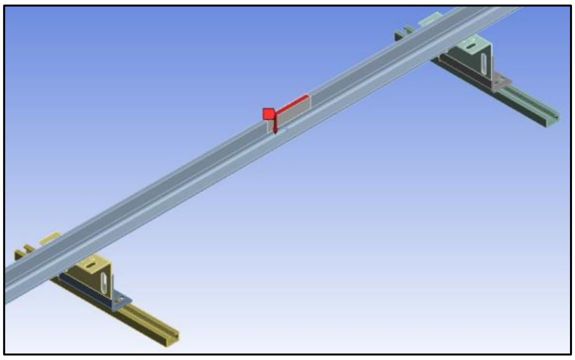
Displacement in mm, bracket



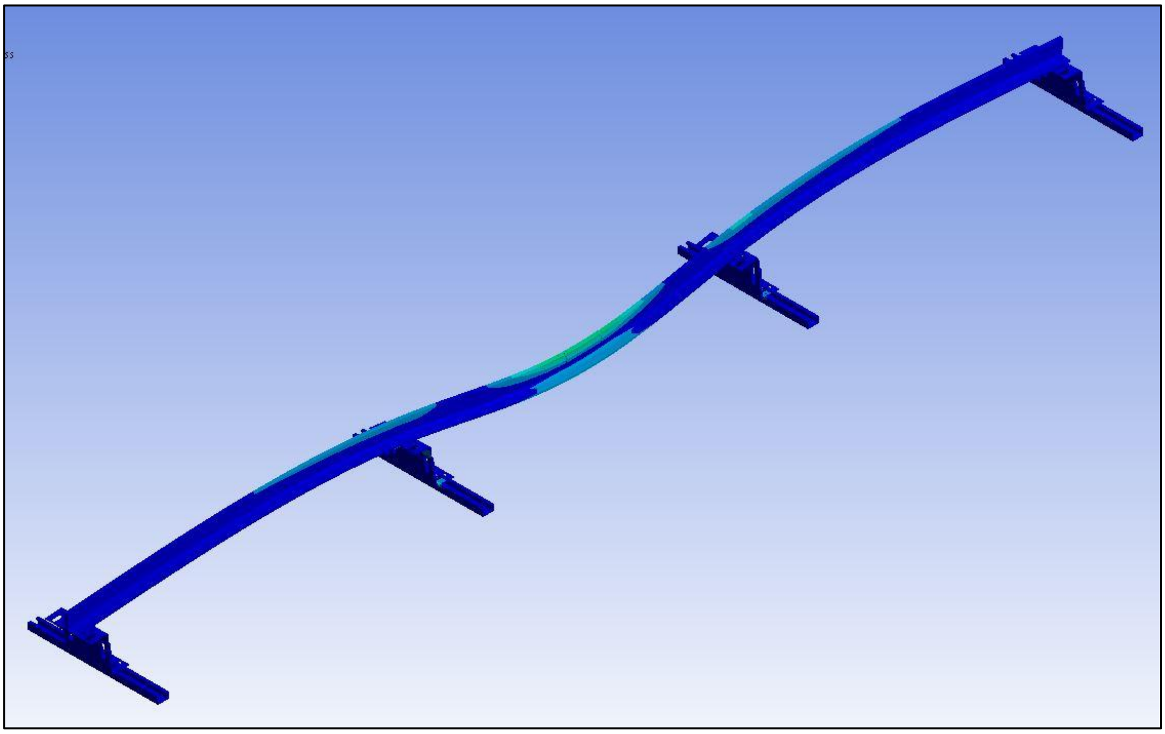
3 Forces acting between two brackets

3.1 Force in y-direction

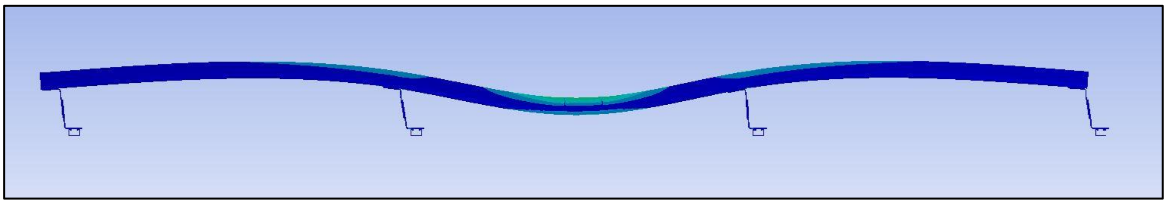
Applied force, F_y = 11 kN

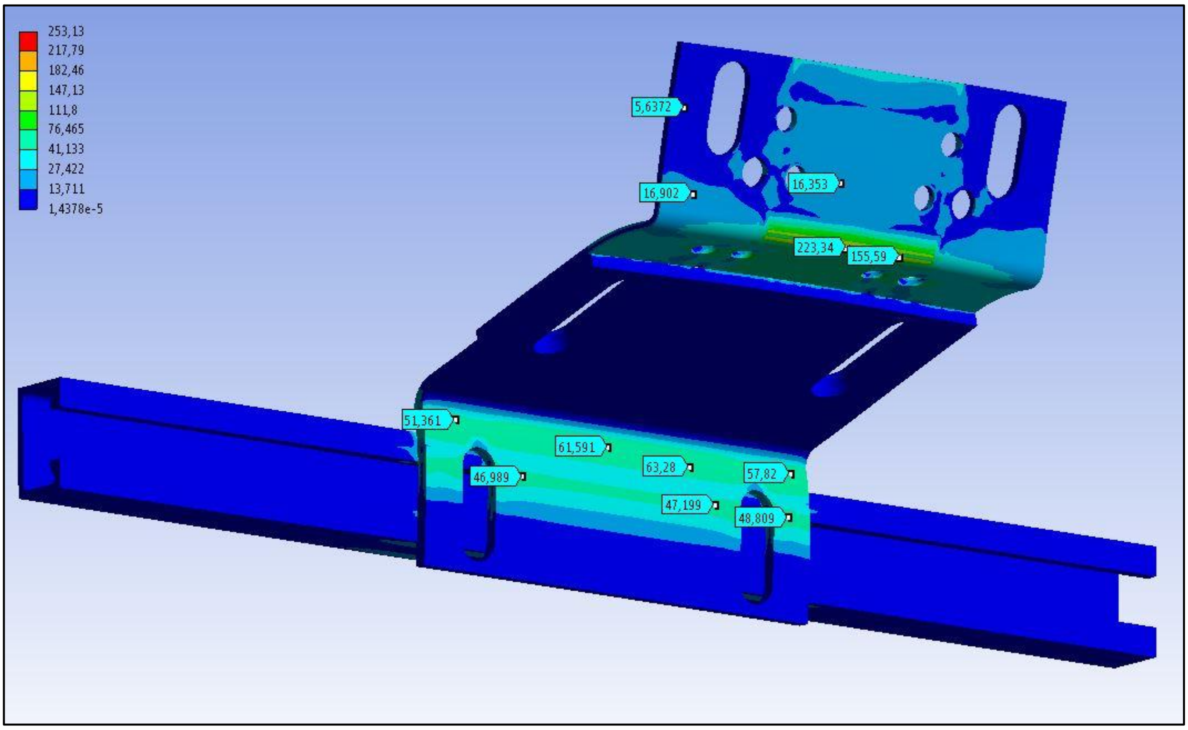
Deformation, isometric view



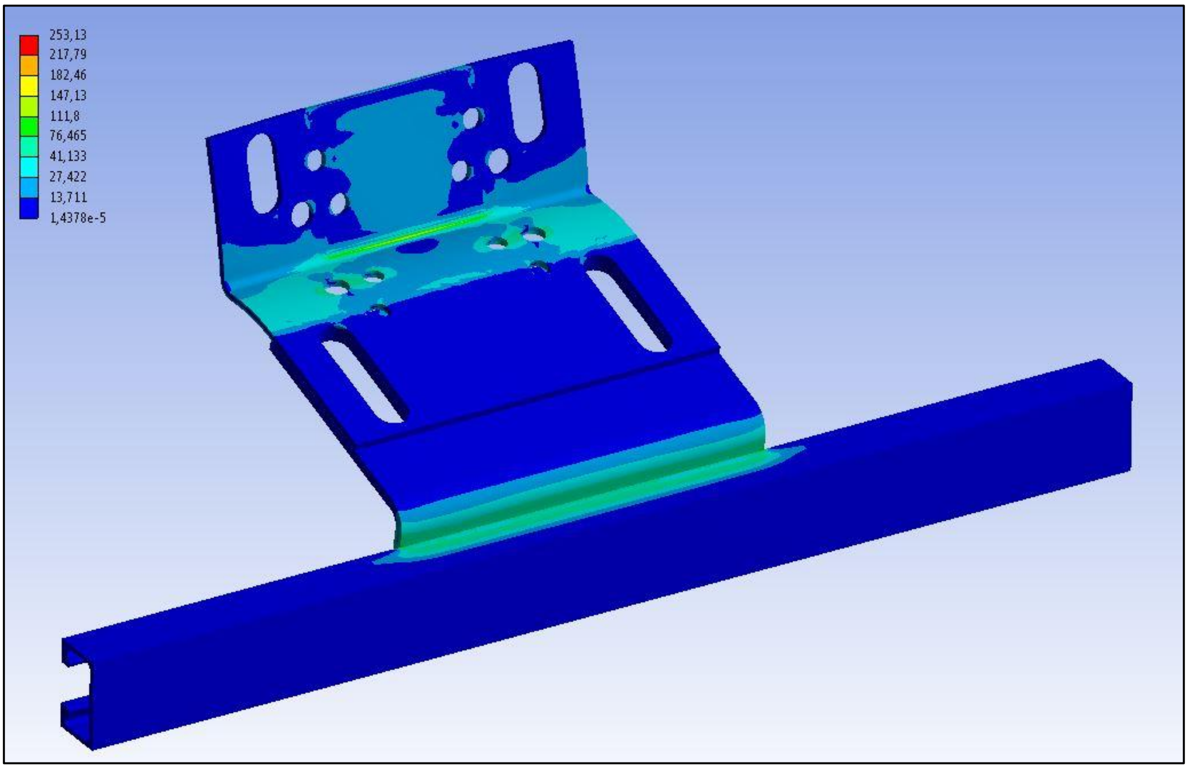
Deformation, side view



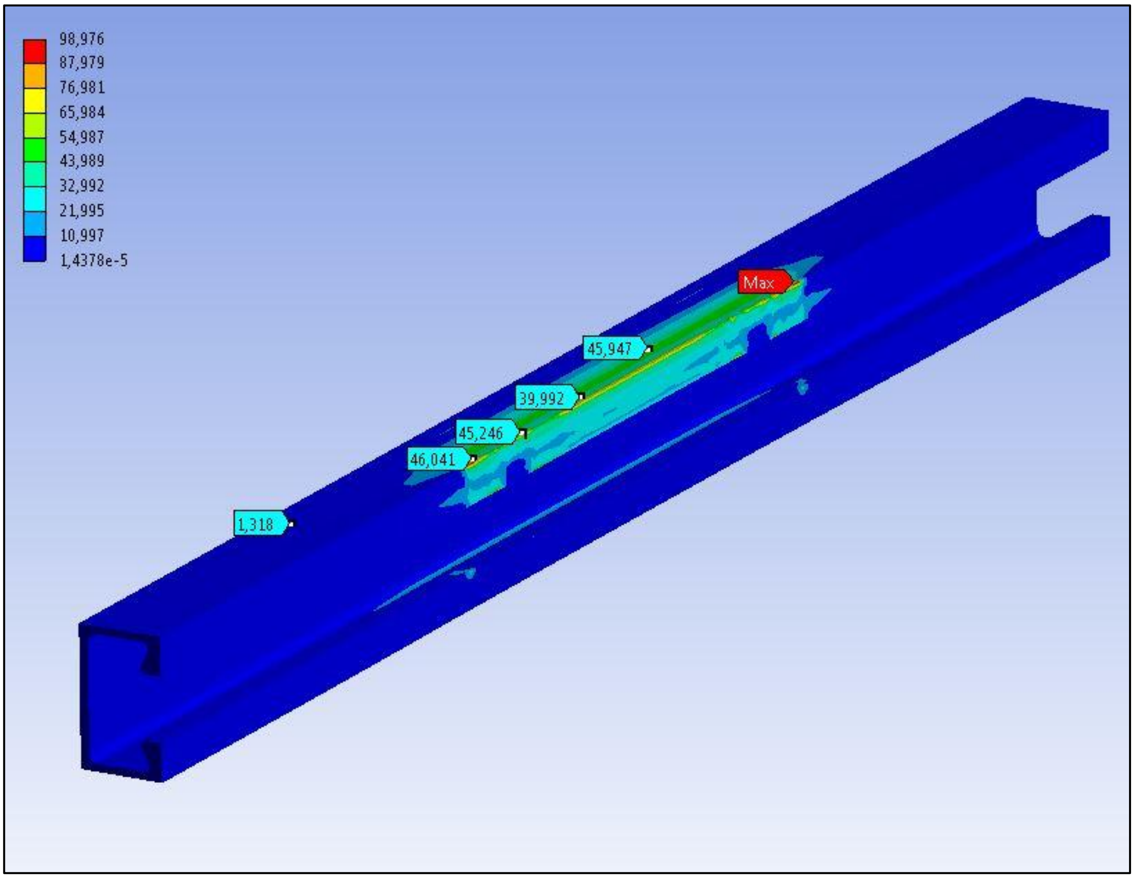
Equivalent stress in MPa, bracket



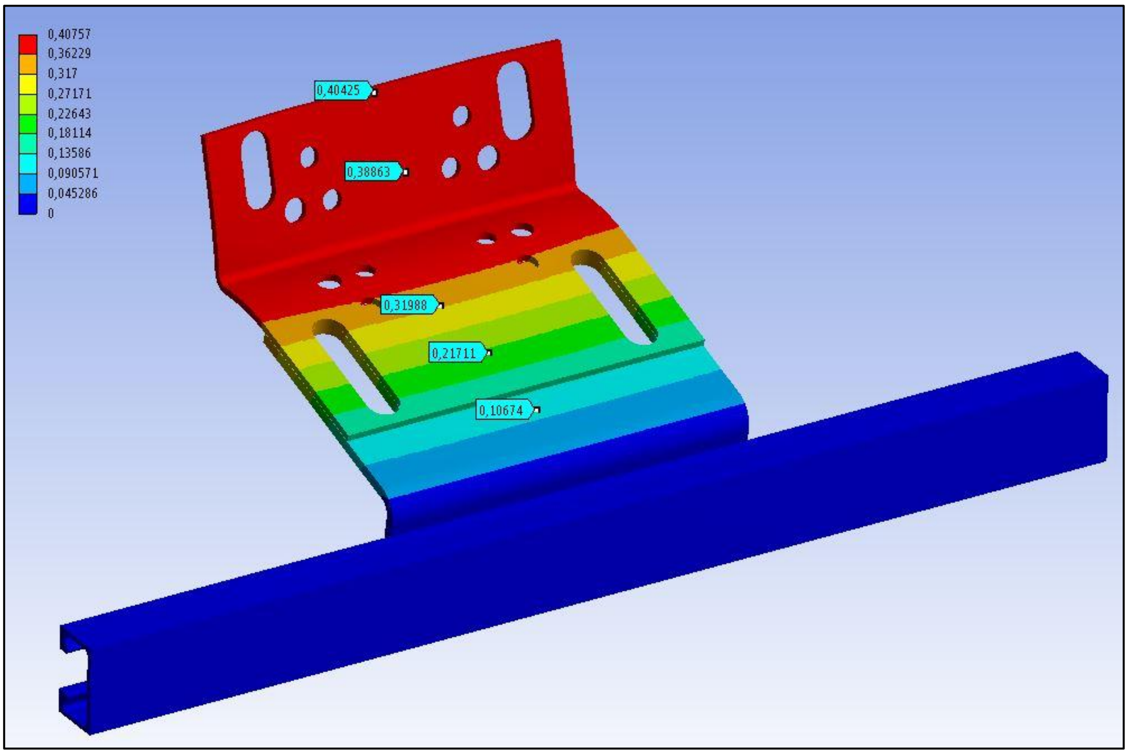
Equivalent stress in MPa, alternative view



Equivalent stress in MPa, Halfen rail

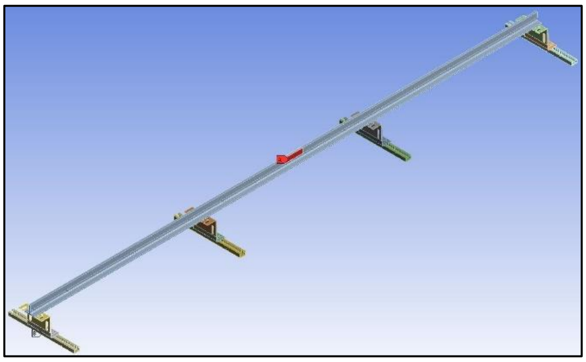
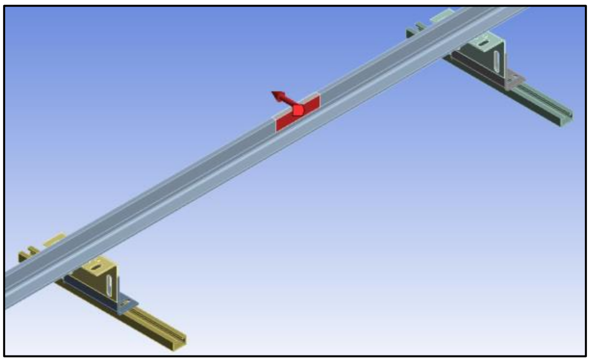


Displacement in mm, bracket

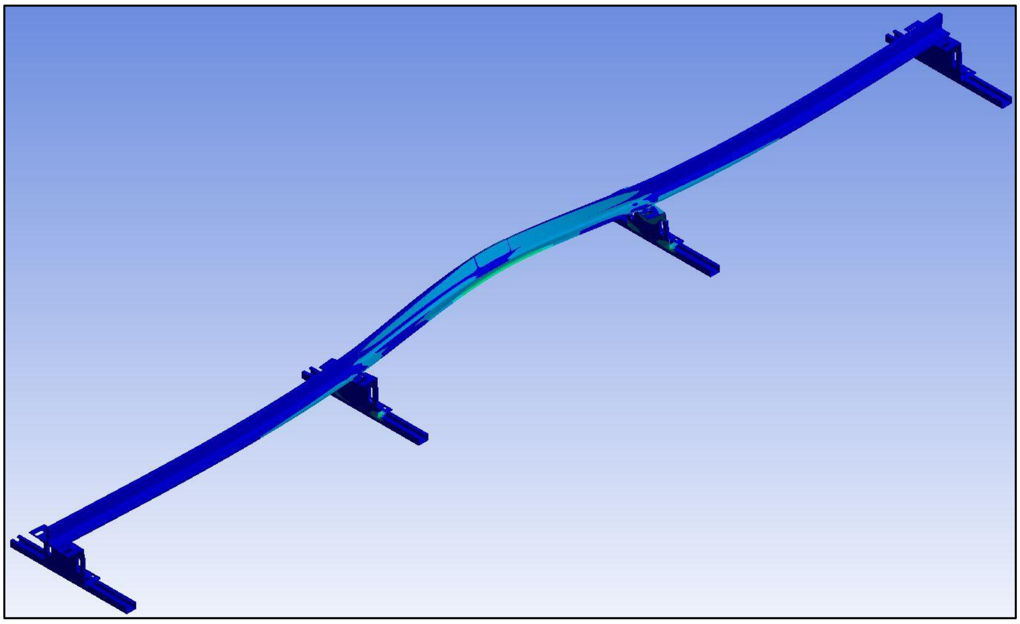
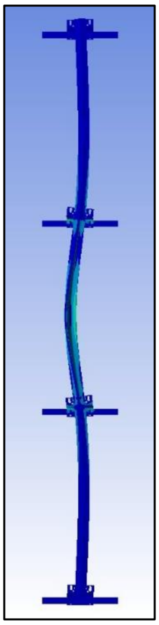


3.2 Force in x-direction

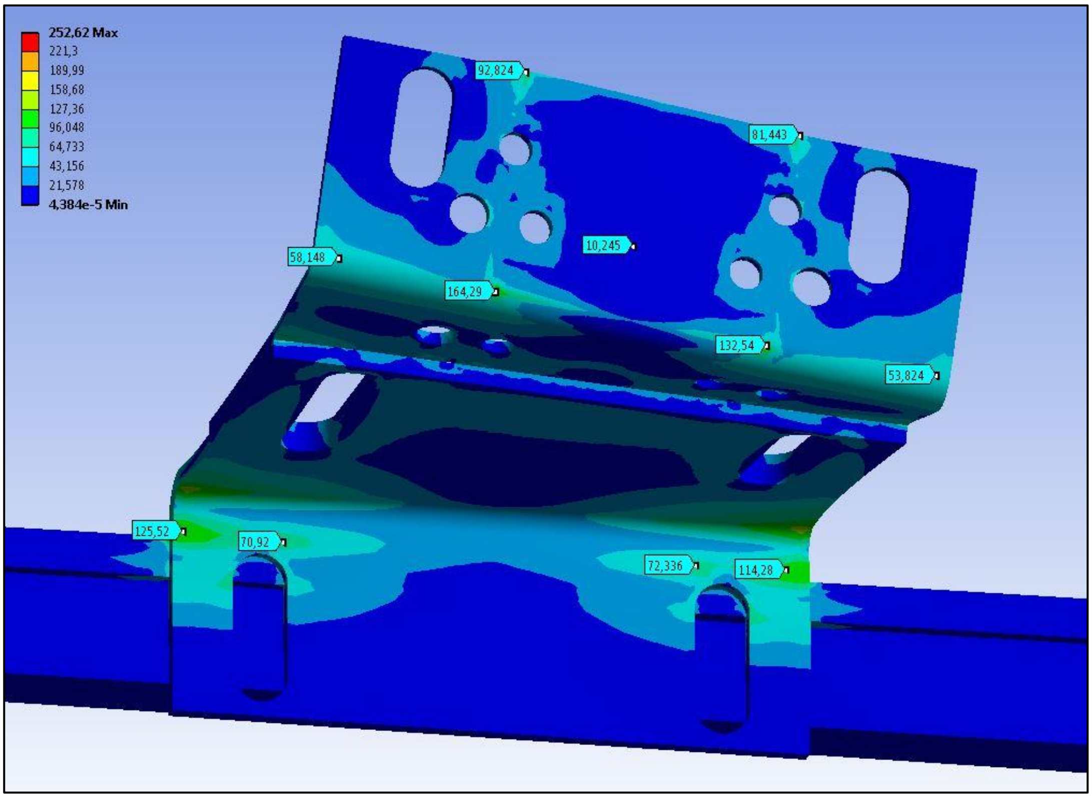
Applied force, F_x = 5,5 kN

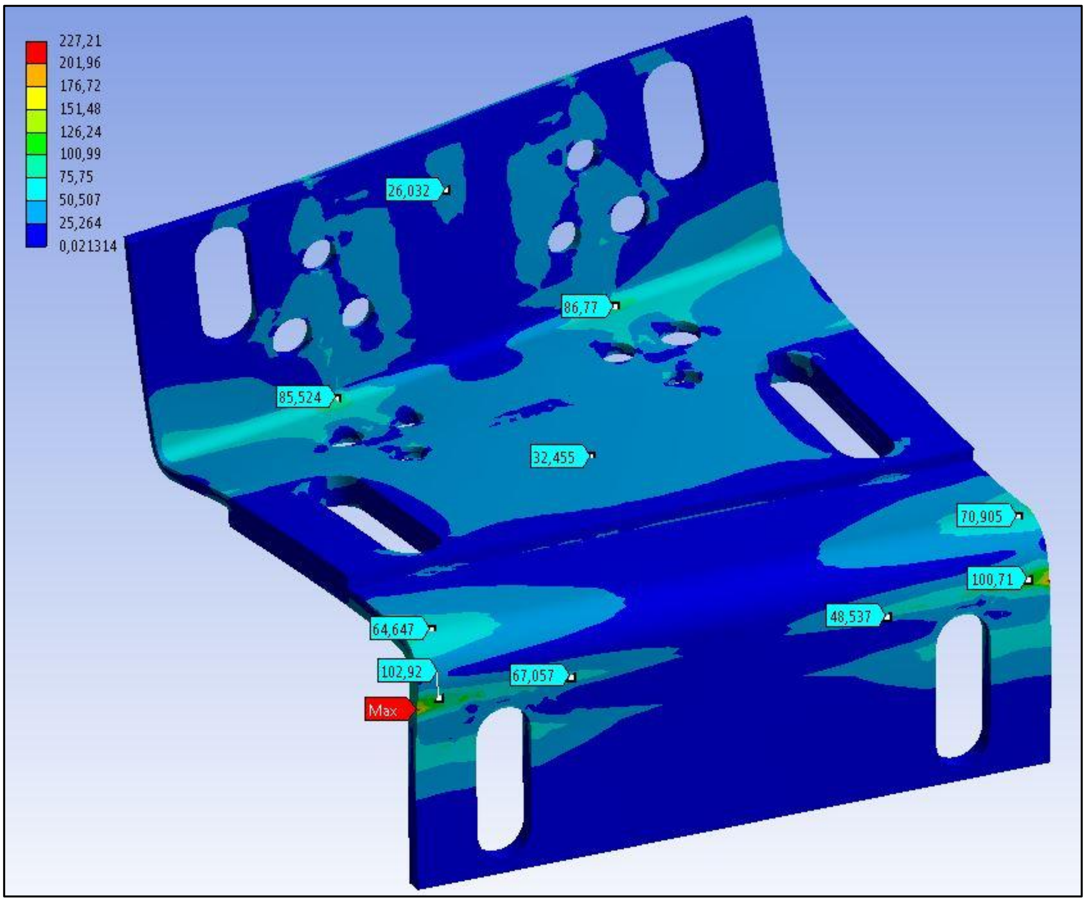
Deformation, isometric and side view

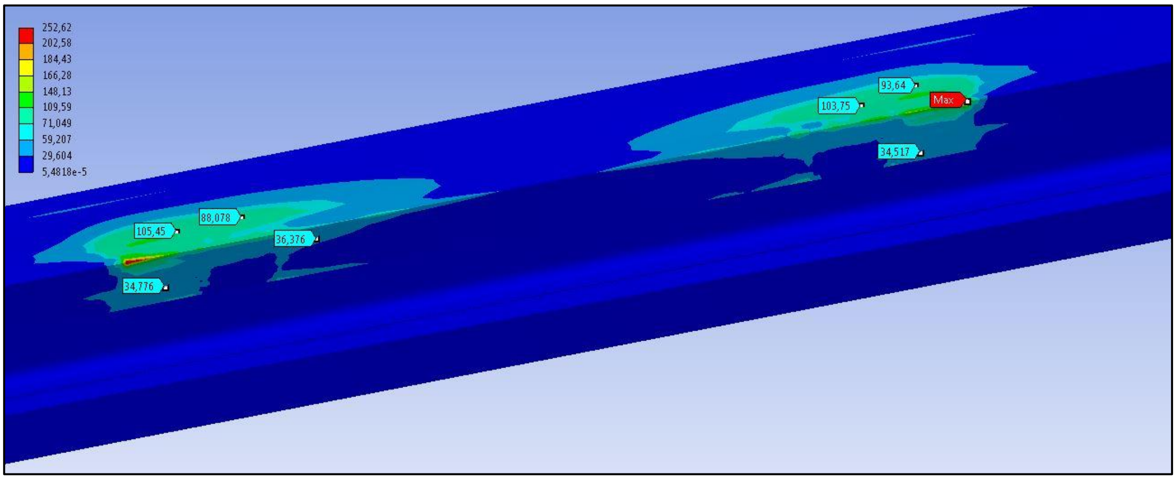
Equivalent stress in MPa, bracket



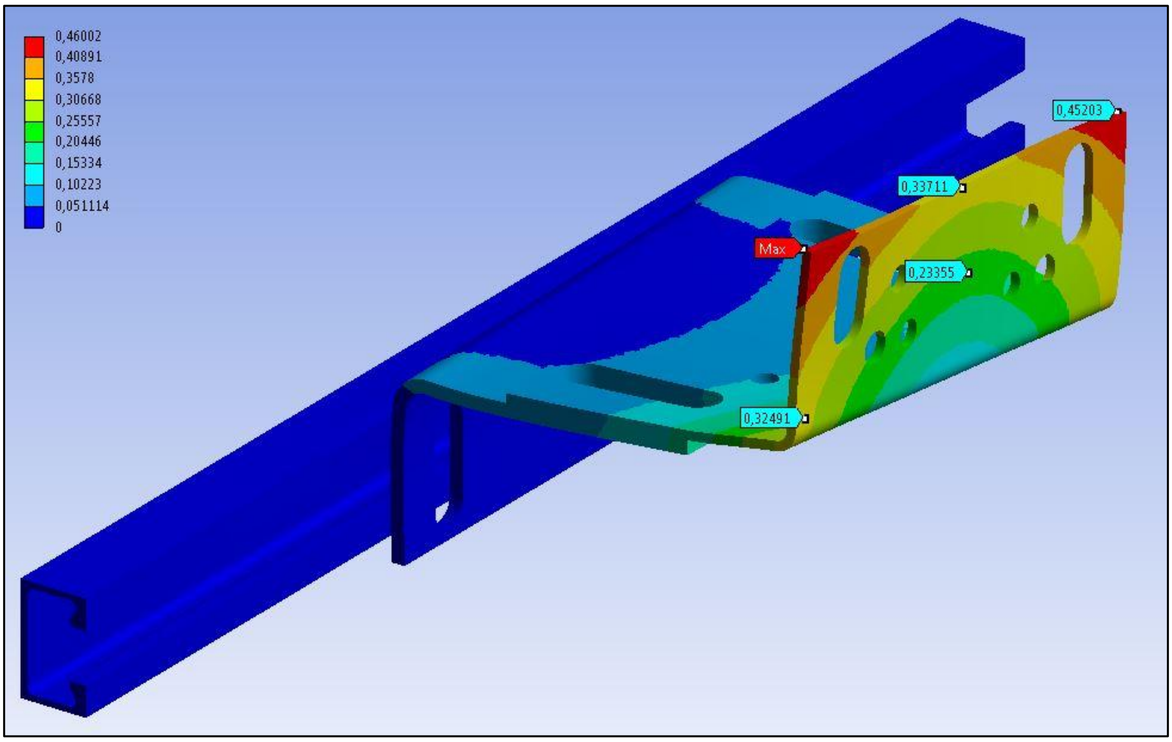
Equivalent stress in MPa, without Halfen channel



Equivalent stress in MPa, local analysis of Halfen channel



Deflection in mm, bracket



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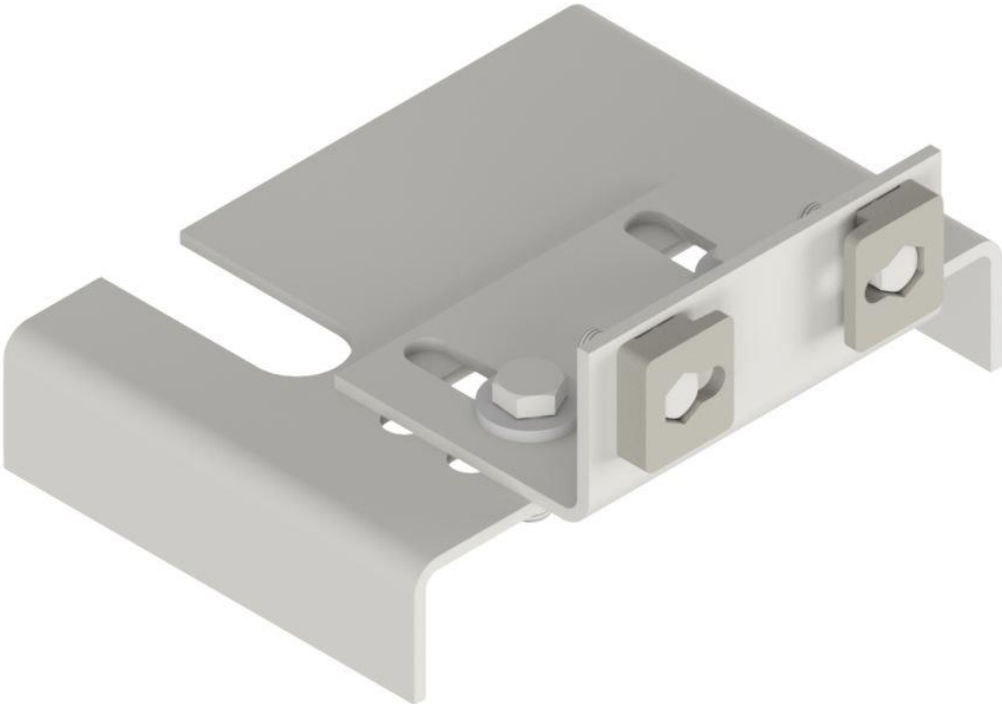
APPENDIX E

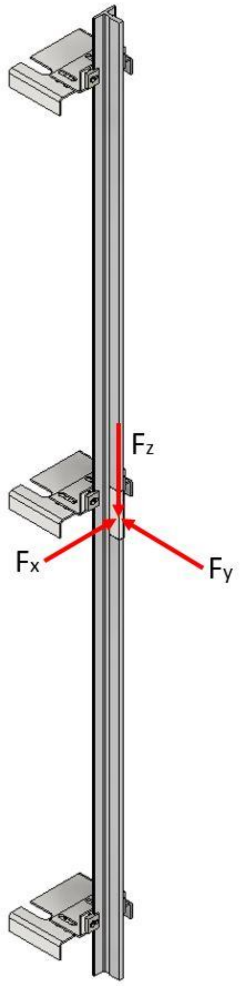
Customized bracket solution

Structural analysis in ANSYS



1 Introduction

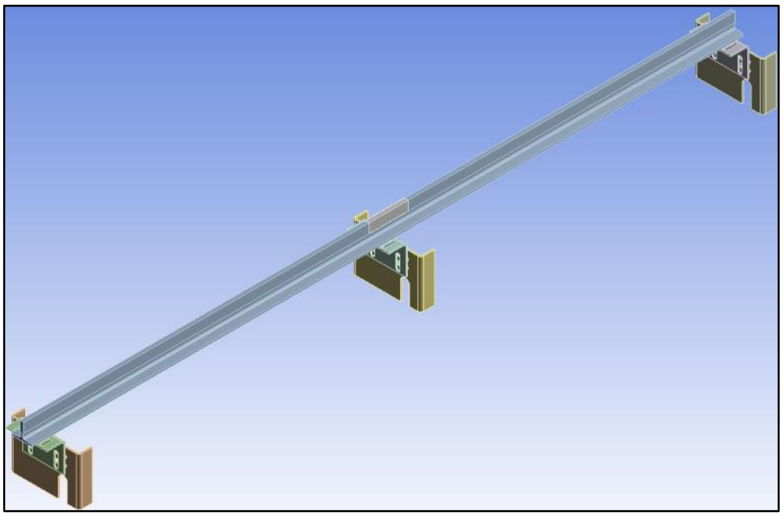
1.1 Applied loads



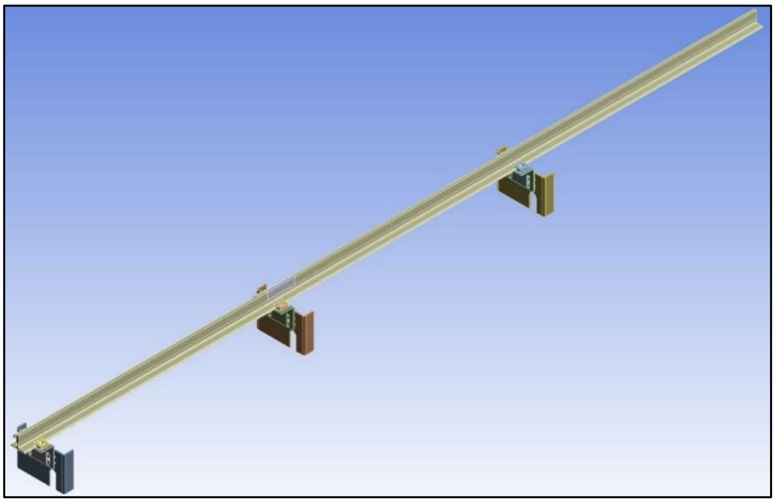
Direction	*Force*	*Unit*
Parallel to trunk, x-direction	$F_x = 11$	kN
Normal to trunk, y-direction	$F_y = 5{,}5$	kN
Parallel to trunk, z-direction	$F_z = 60$	kN

1.2 FEM models

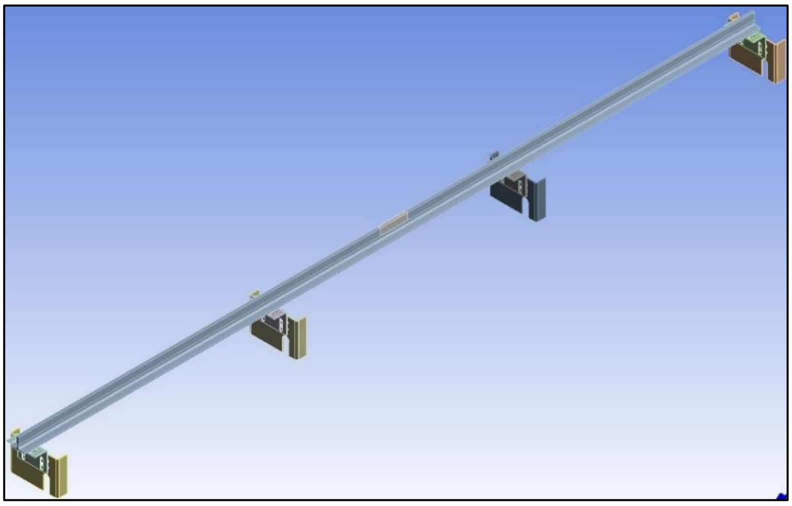
Load case 1 Force acting on one bracket, x-direction and y-direction



Load case 2 Force acting on one bracket caused by safety gear, z-direction

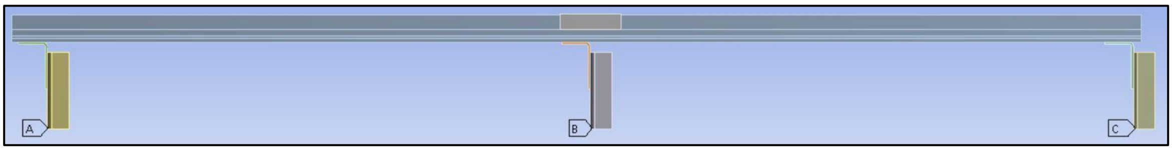


Load case 3 Force acting between two brackets, x-direction and y-direction



1.3 Boundary conditions

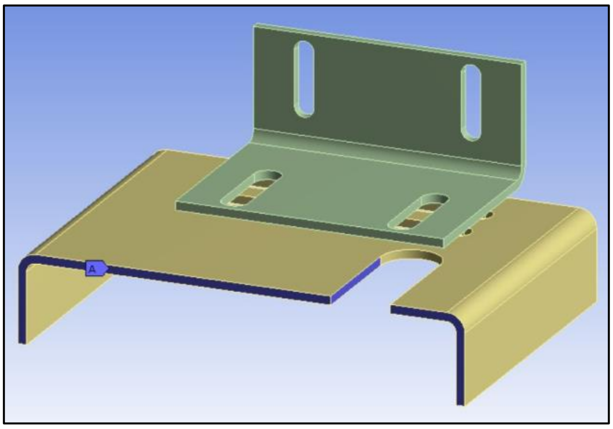
Load case 1



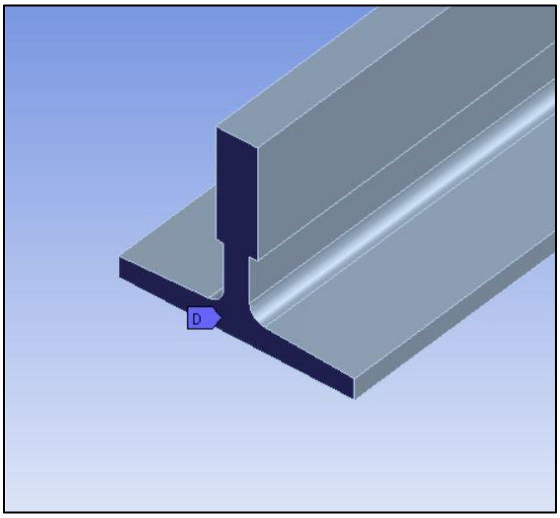
Load case 2

Load case 3

Brackets

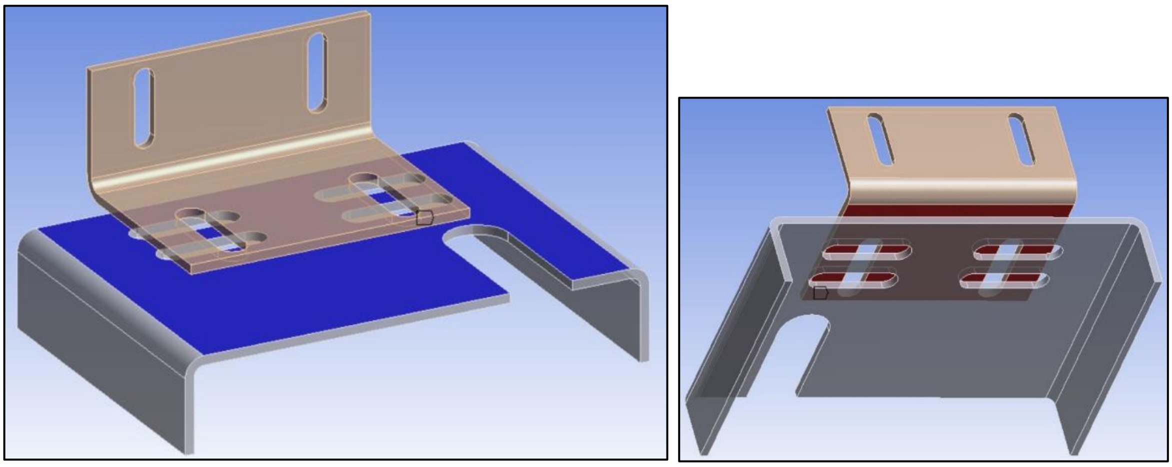


Baseplate (for load case 2)

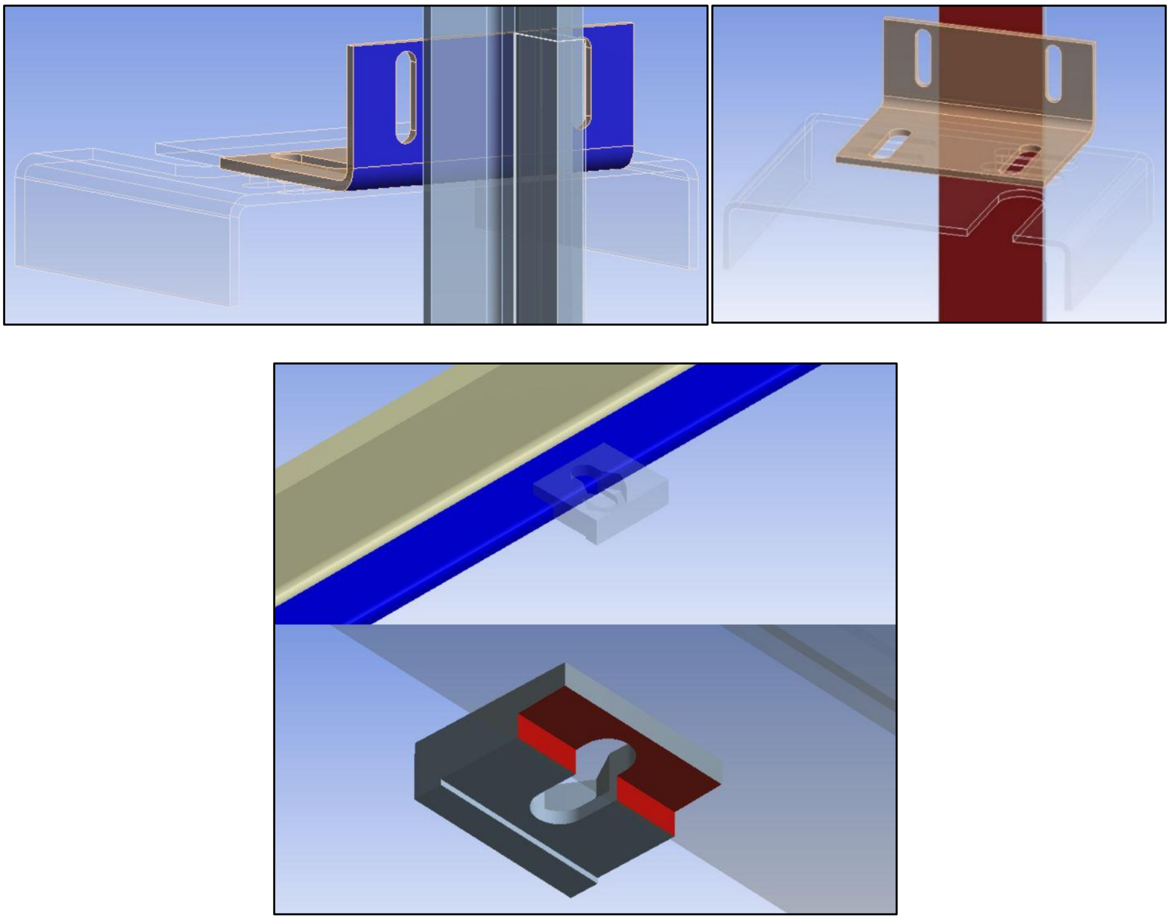


1.4 Contacts

Bonded contact for all load cases

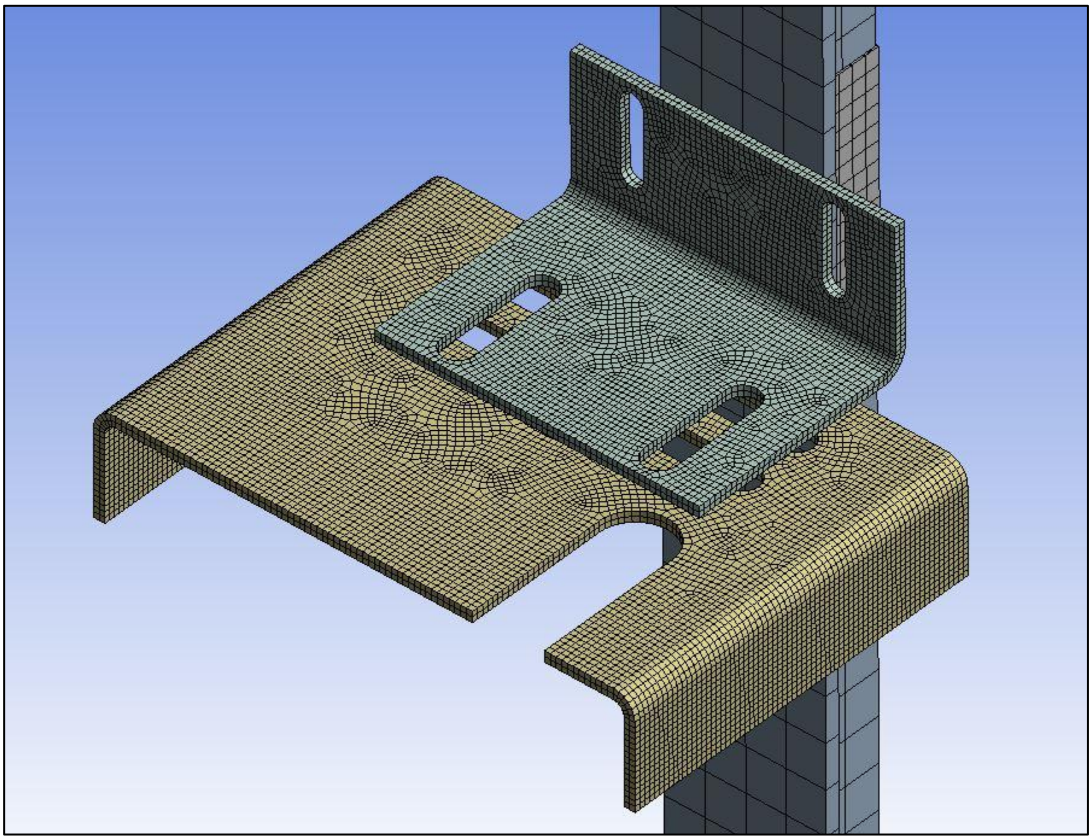


Frictional contact for the following parts in load case 2

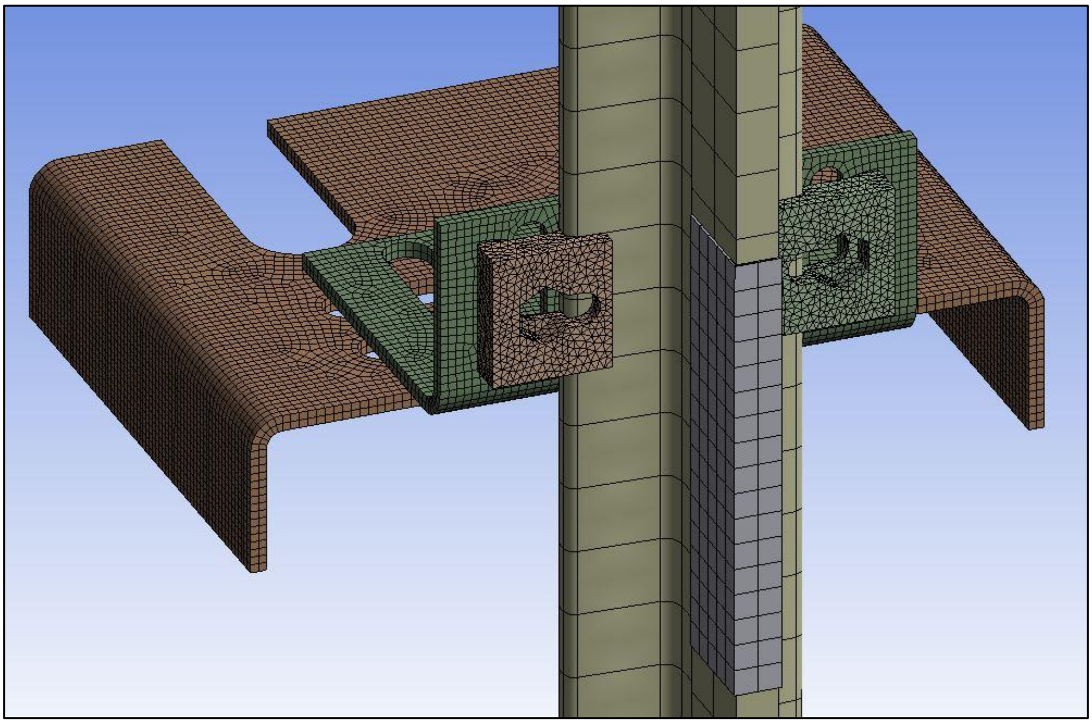


1.5 Mesh

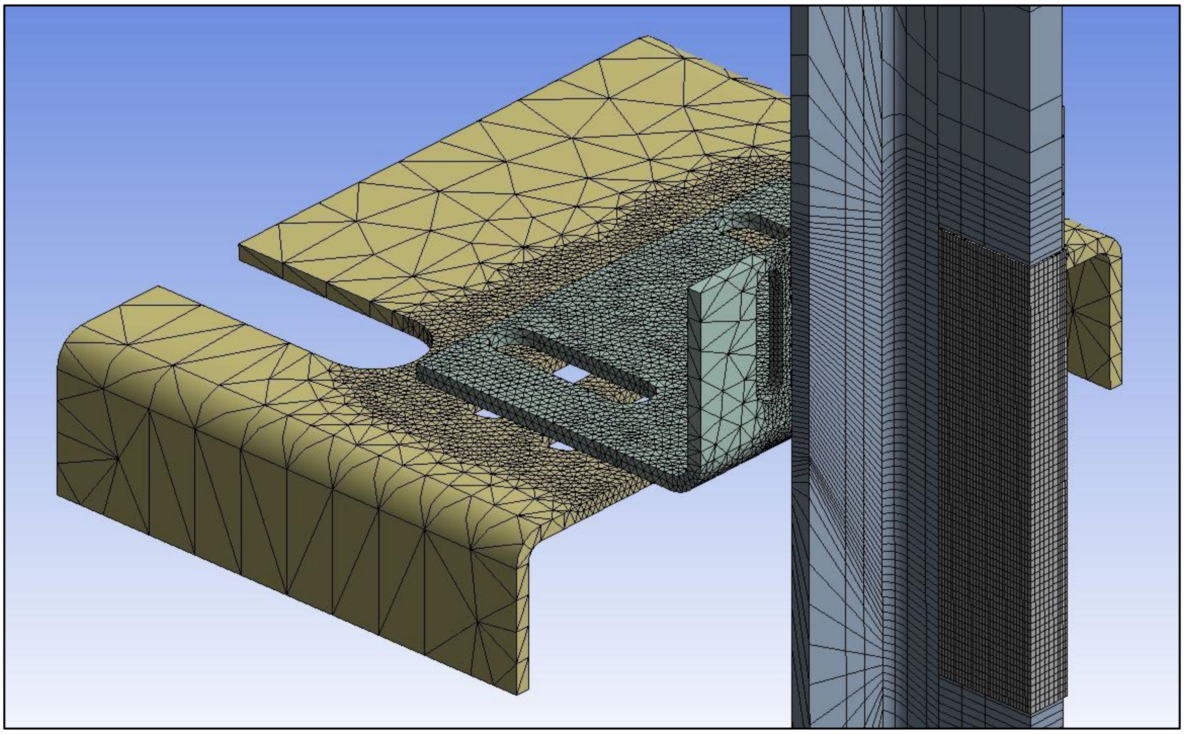
Hex meshed bracket (L-profile and U-channel), guide rail and guide shoe



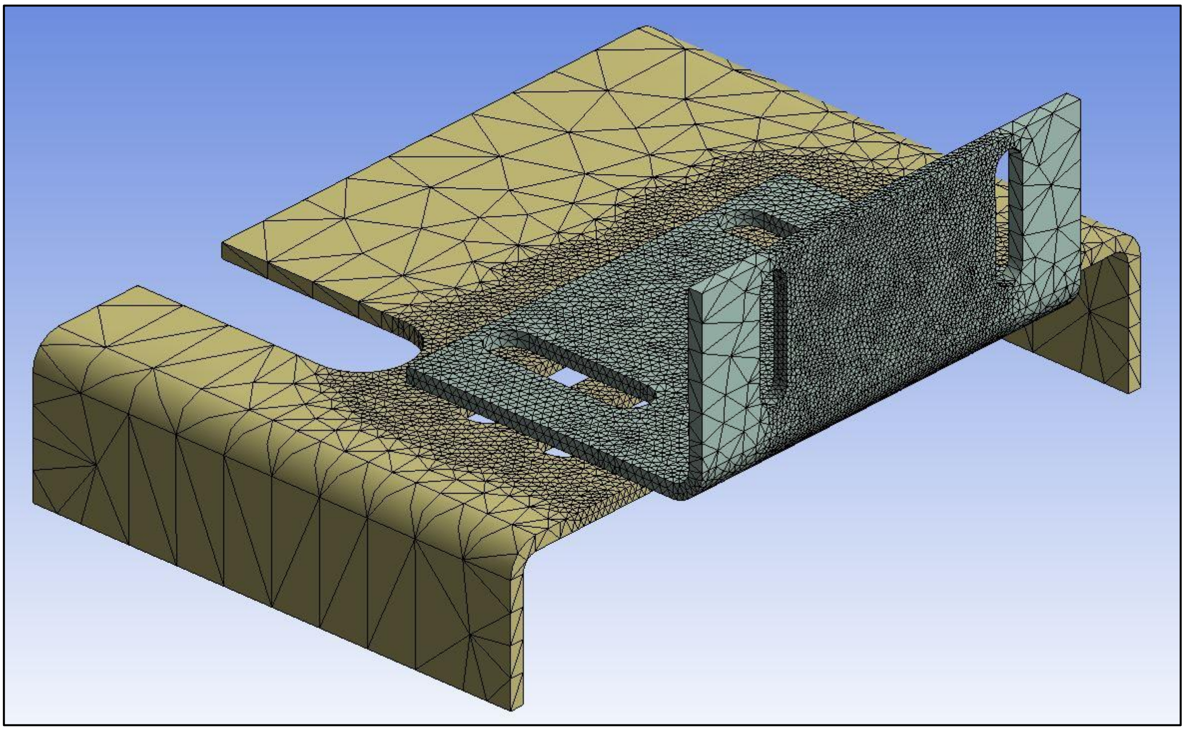
Tetra meshed rail clips



Alternative mesh, contact sizing



Alternative mesh, without guide rail



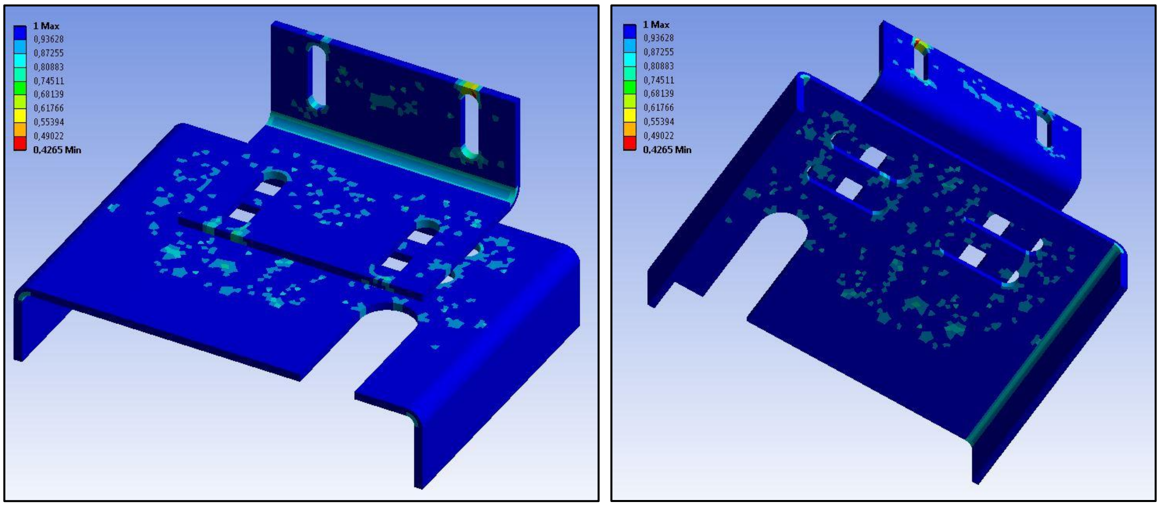
Model	***Nodes***	***Elements***
Load case 1	166 656	28 429
Load case 2	195 677	45 887
Load case 3	305 439	52 753

Components	***Mesh type and method***	***Element size***
Evaluated bracket(s)	Hex mesh, MultiZone	3 mm
Other brackets	Hex mesh, MultiZone	10 mm
Guide shoe	Hex mesh, Hex Dominant	10 mm
Guide rail	Hex mesh, Hex Dominant	25 mm
Rail clips, for load case 2	Tetra mesh	3 mm

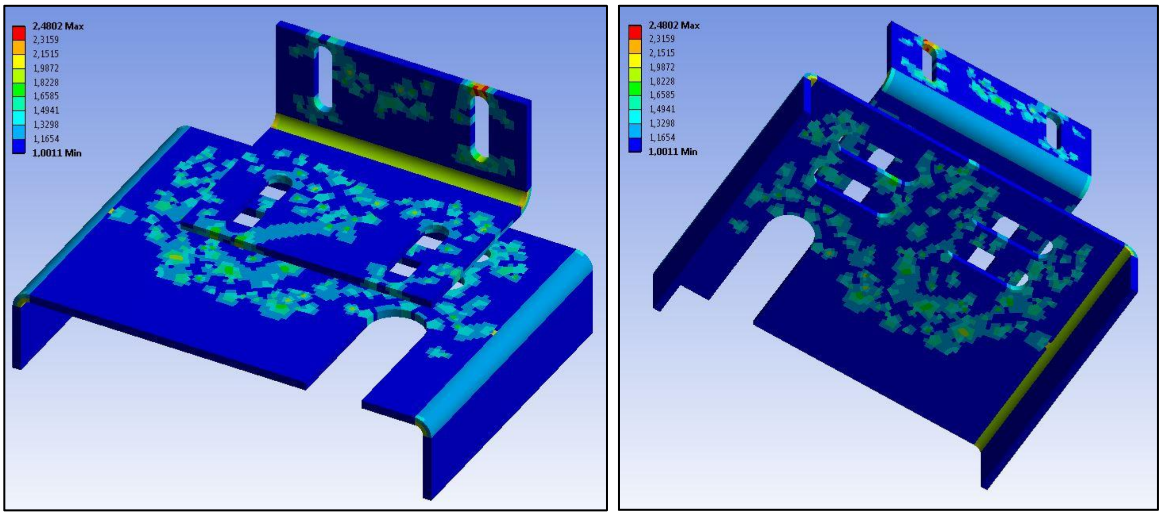
Model	***Nodes***	***Elements***
Load case 1, alternative mesh	193 626	89 089

Components	***Mesh type and method***	***Element size***
All	Tetra mesh, contact refinement	2 and 3 mm

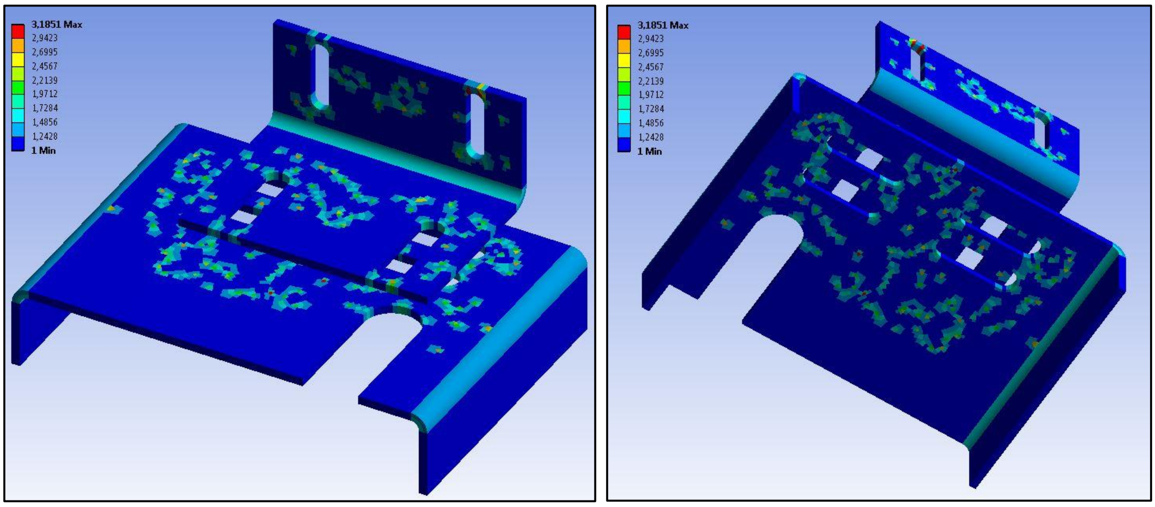
Element quality, bracket



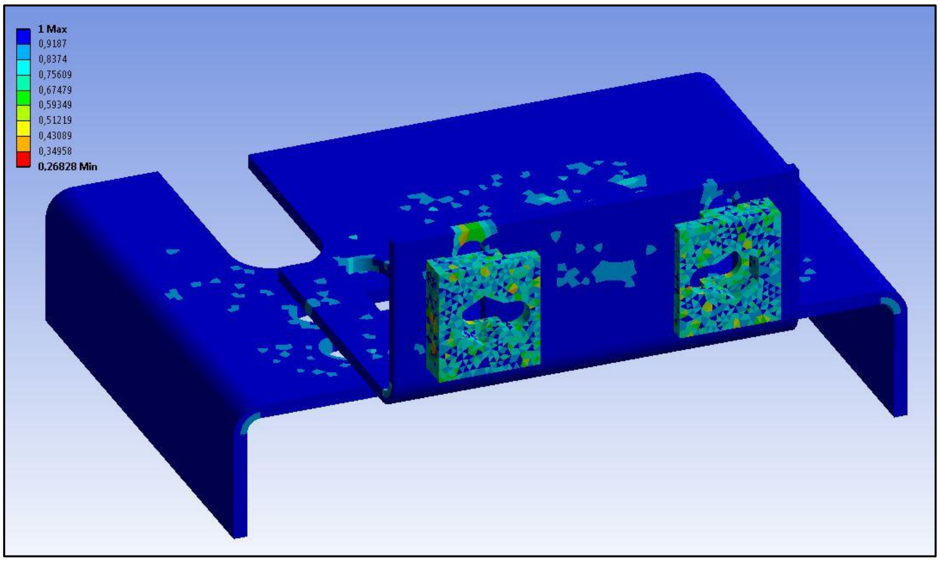
Aspect ratio, bracket



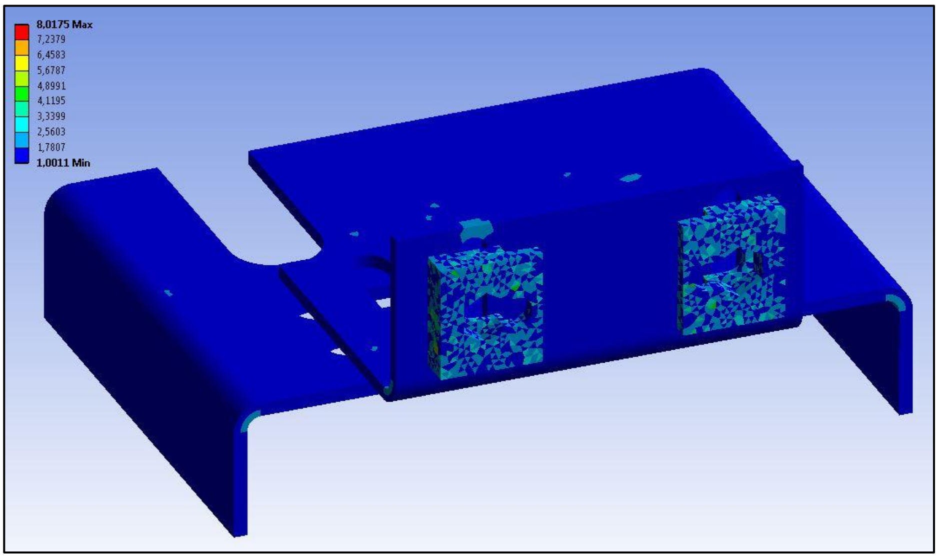
Jacobian ratio, bracket

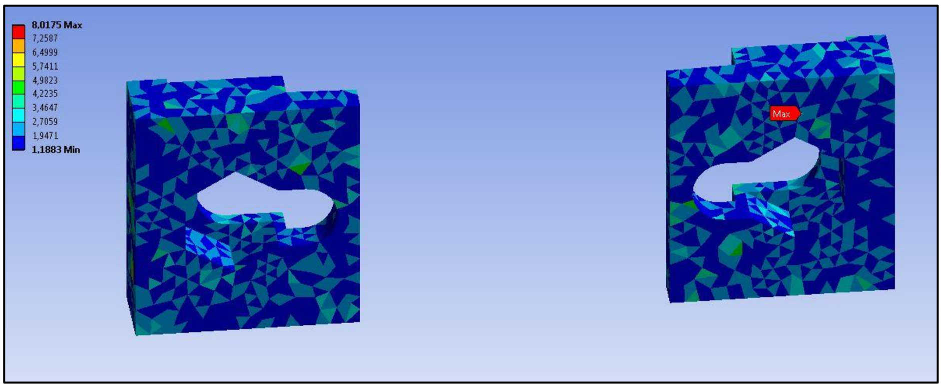


Element quality, rail clips

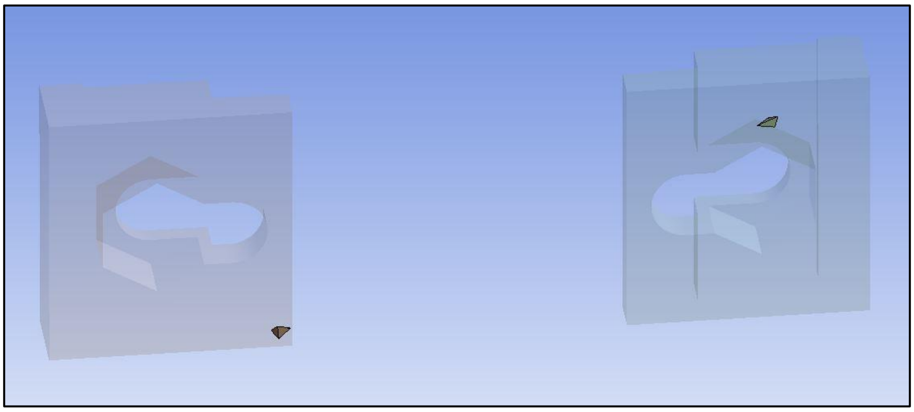


Aspect ratio, rail clips

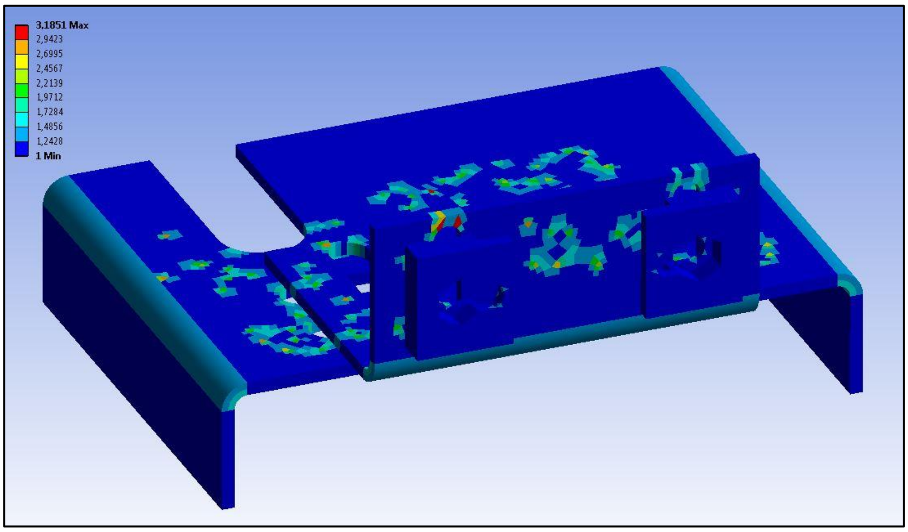




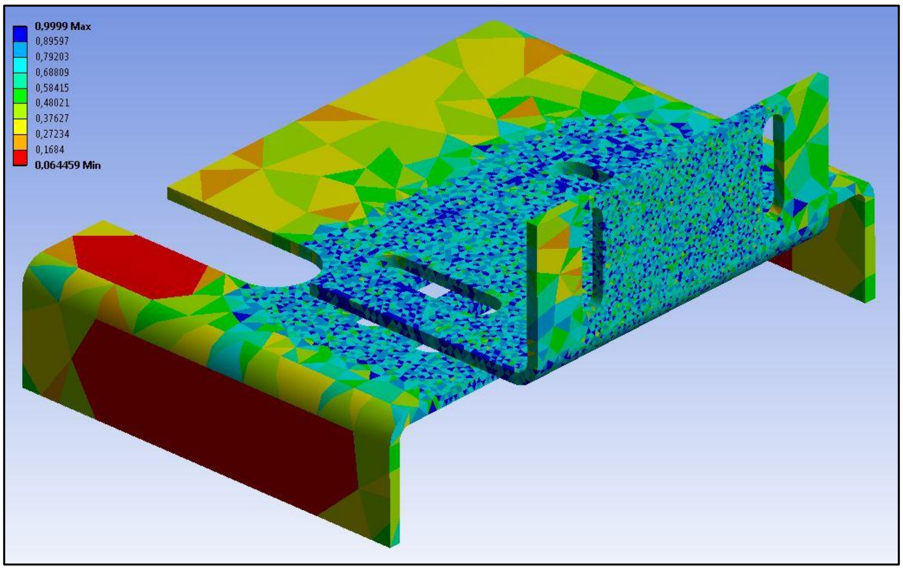
Element with max. aspect ratio



Jacobian ratio, rail clips



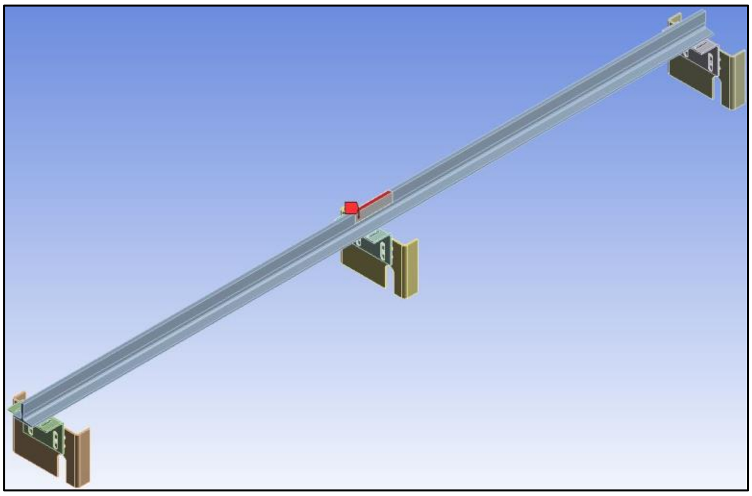
Element quality for the alternative mesh

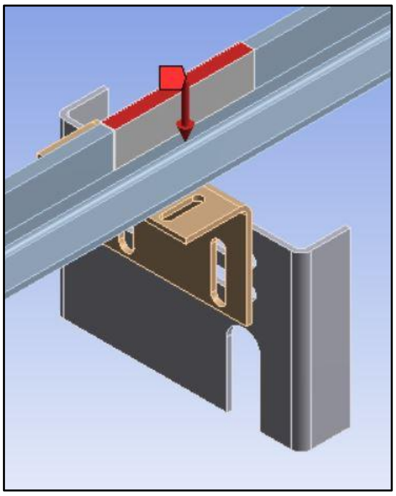


2 Forces acting on one bracket

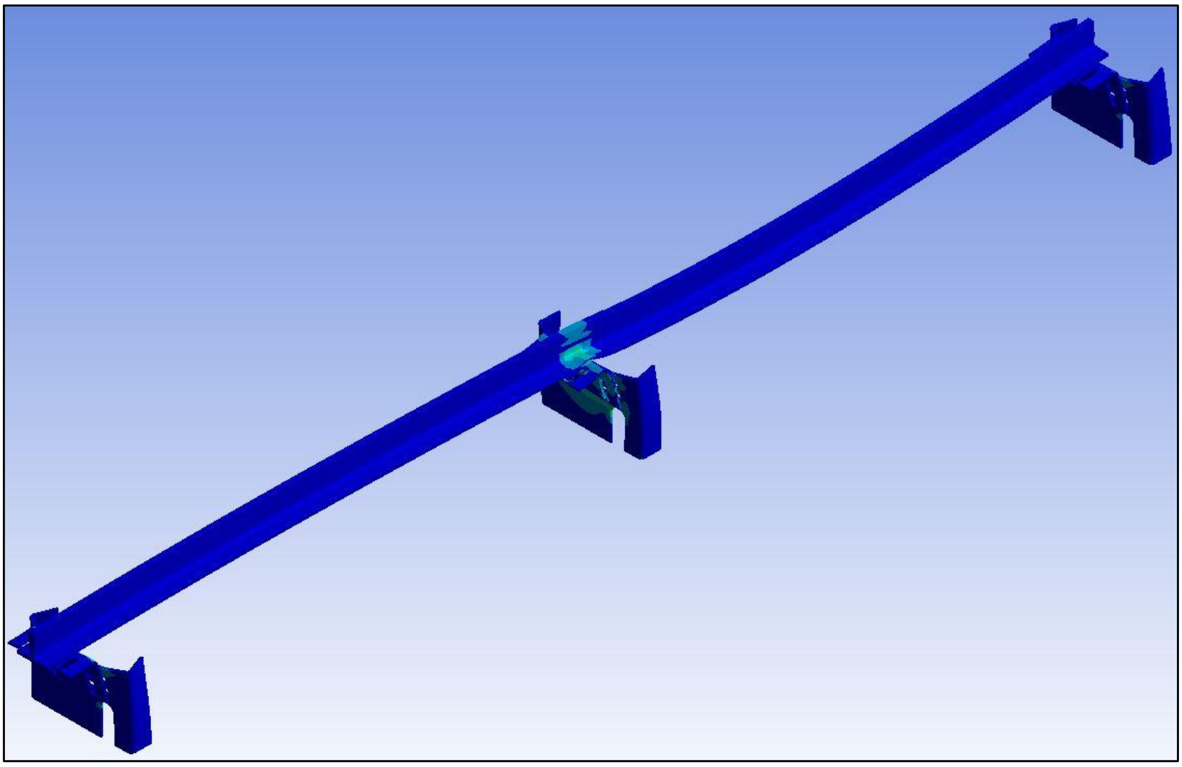
2.1 Force in y-direction

Applied force, F_y = 11 kN

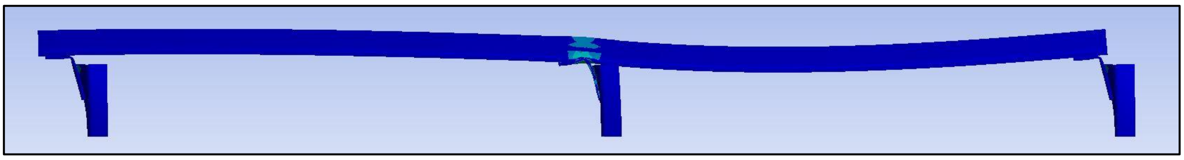




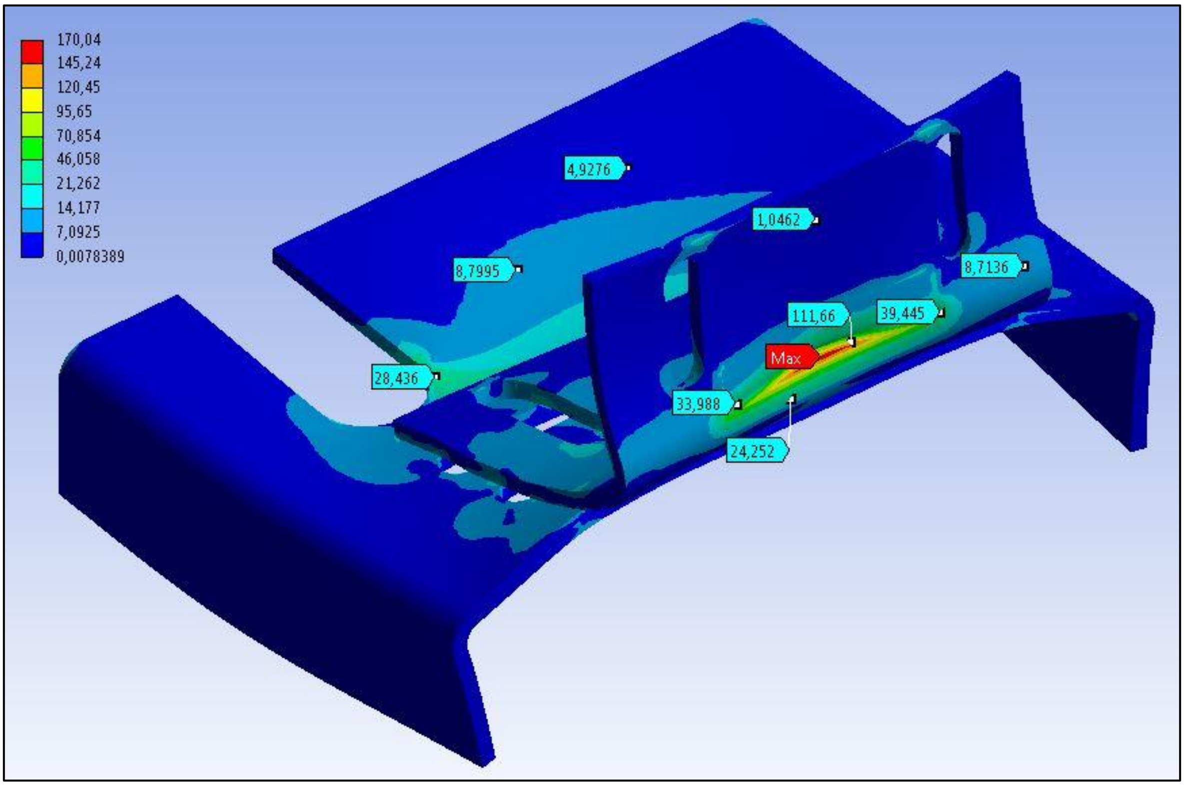
Deformation, isometric view



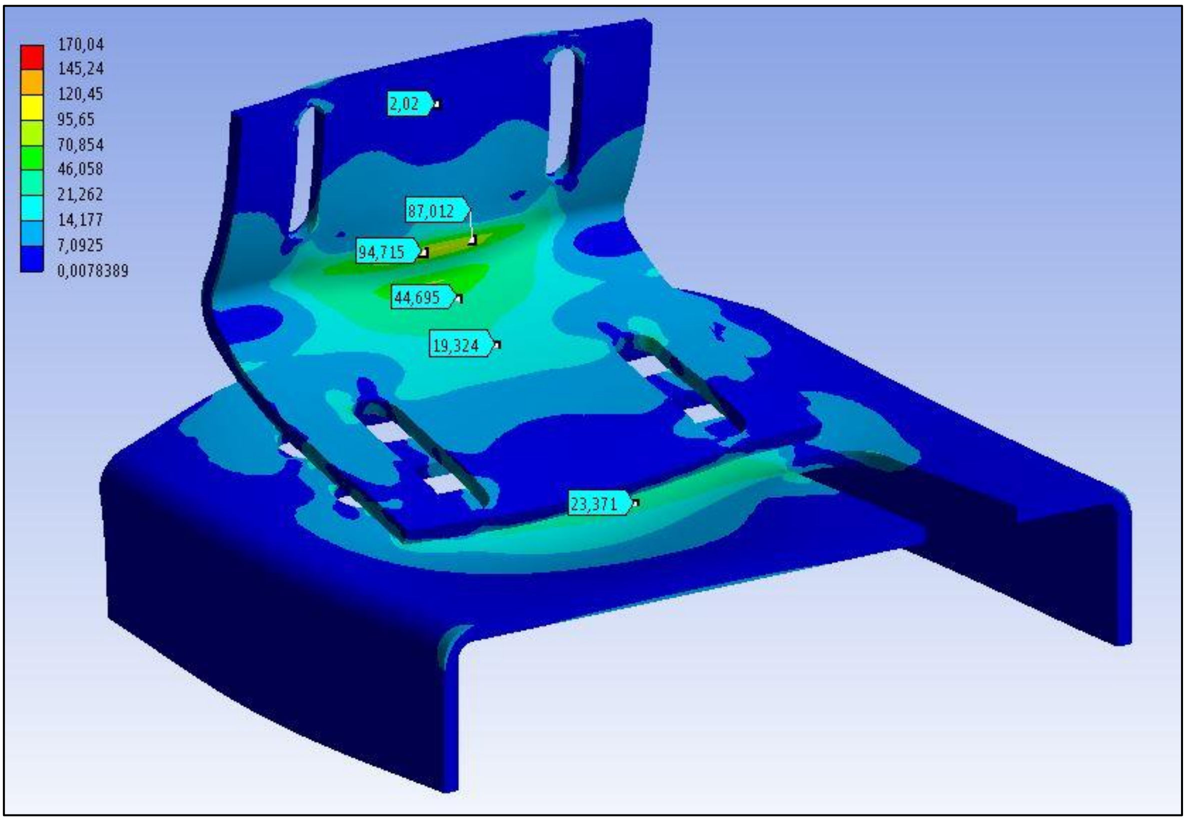
Deformation, side view



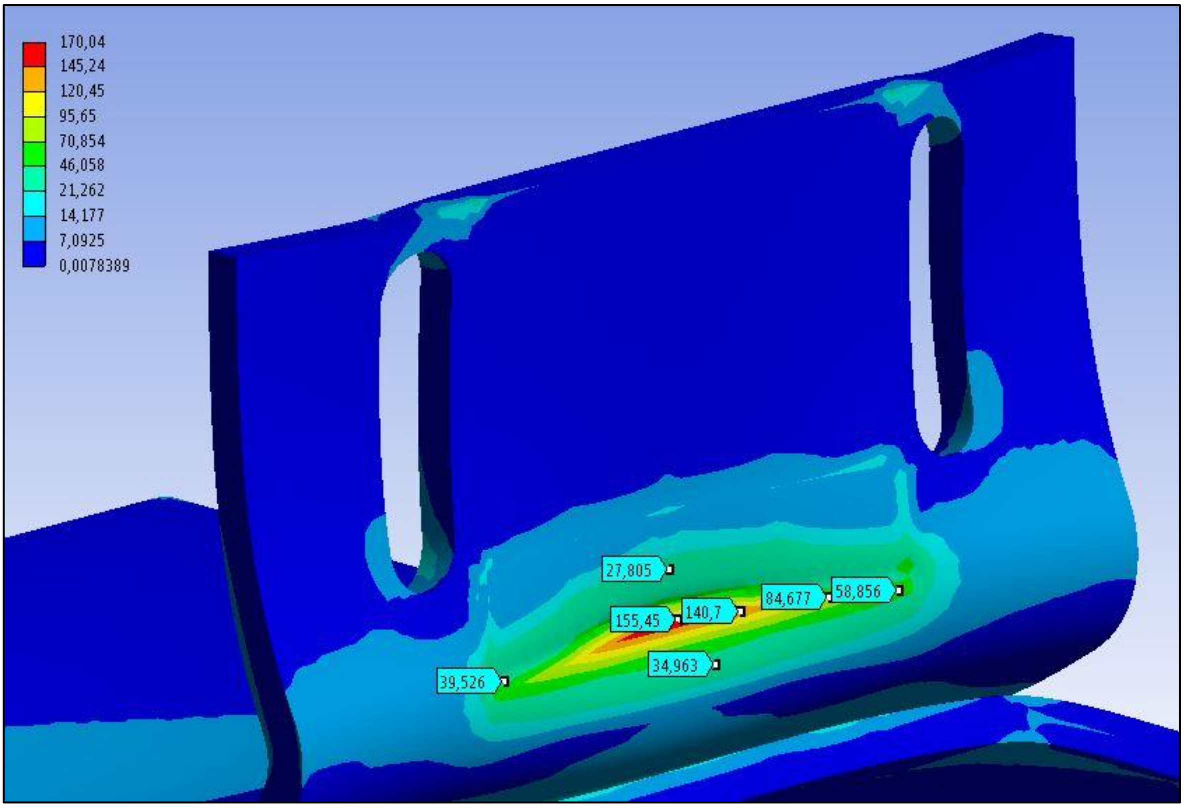
Equivalent stress in MPa, bracket



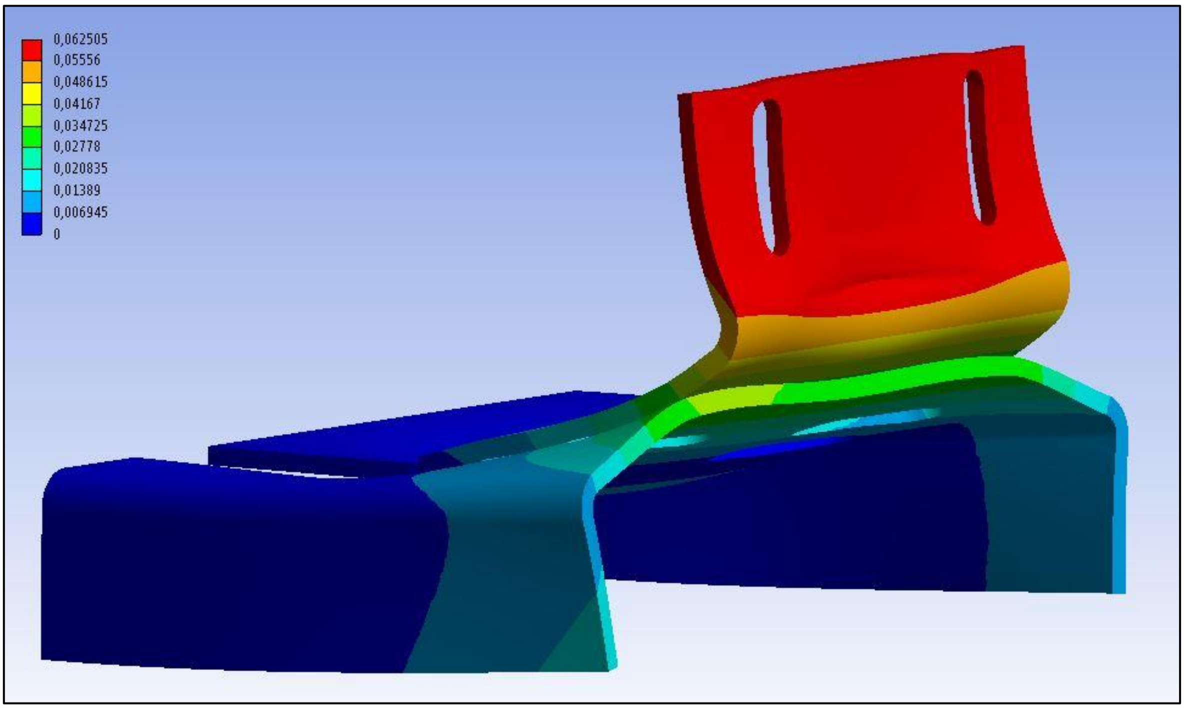
Equivalent stress in MPa, alternative view



Equivalent stress in MPa, local analysis

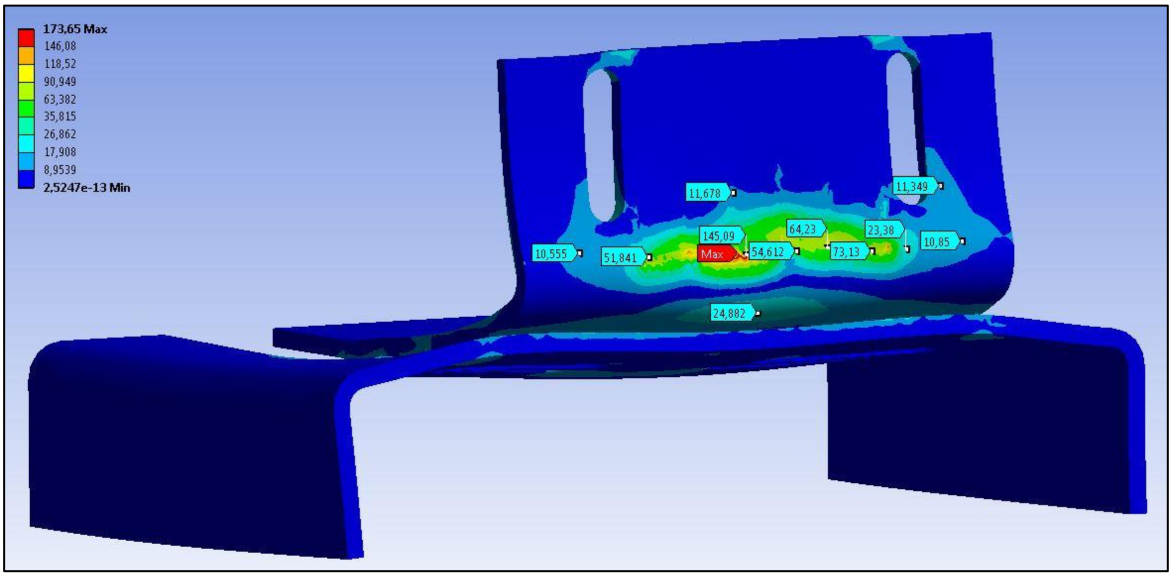


Deformation in mm, bracket

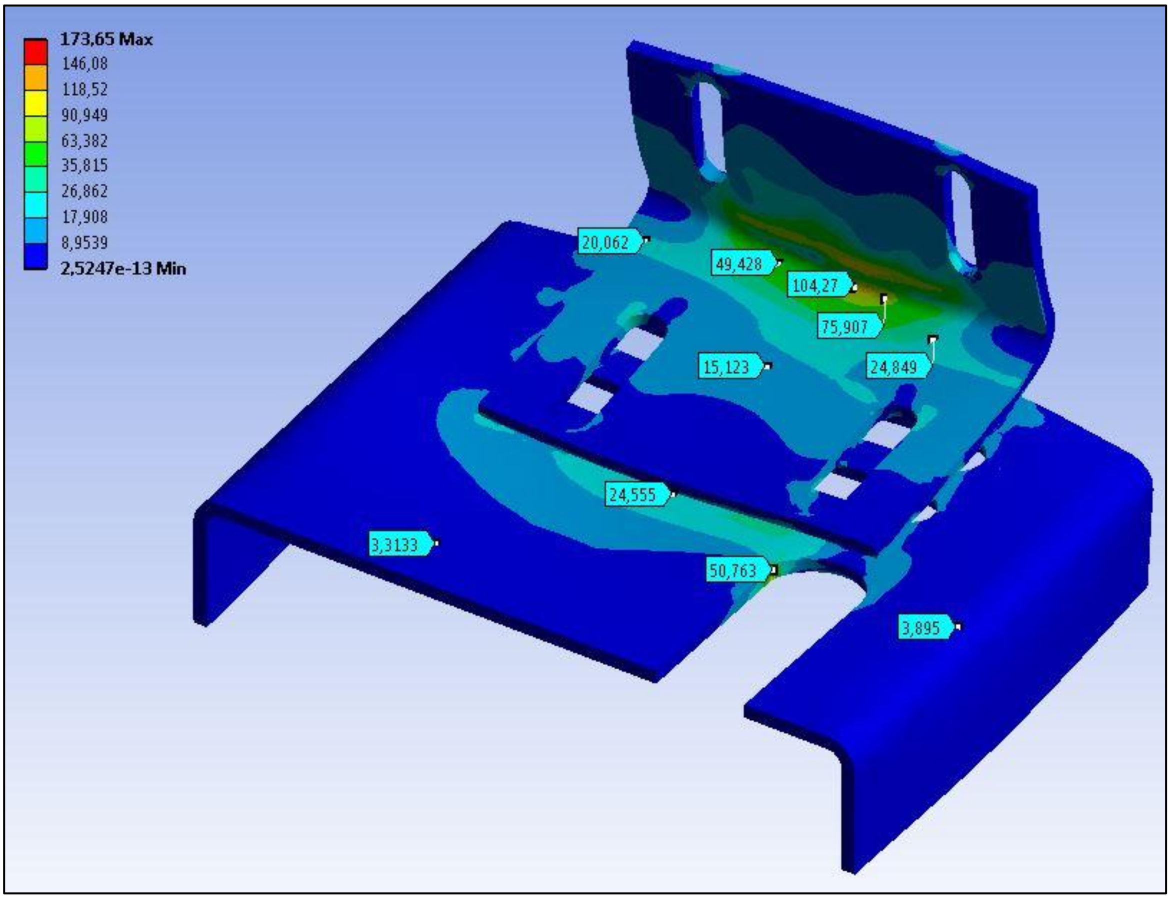


2.2 Force in y-direction, using the alternative mesh

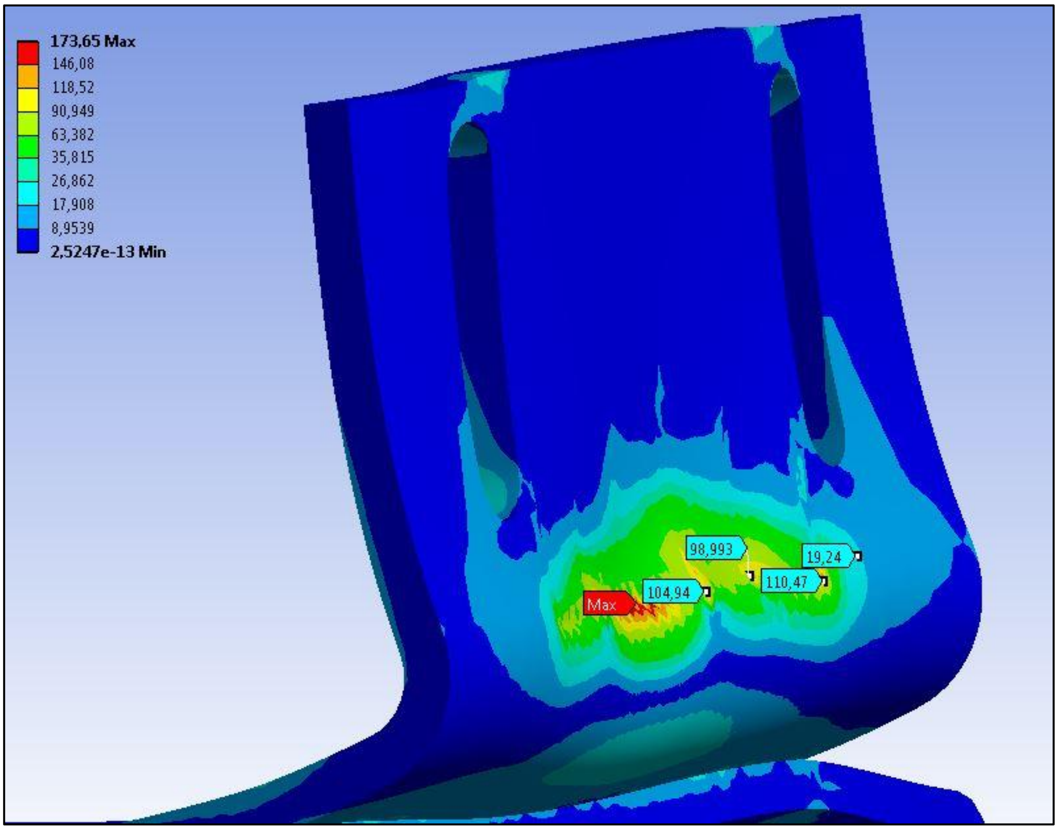
Equivalent stress in MPa, bracket



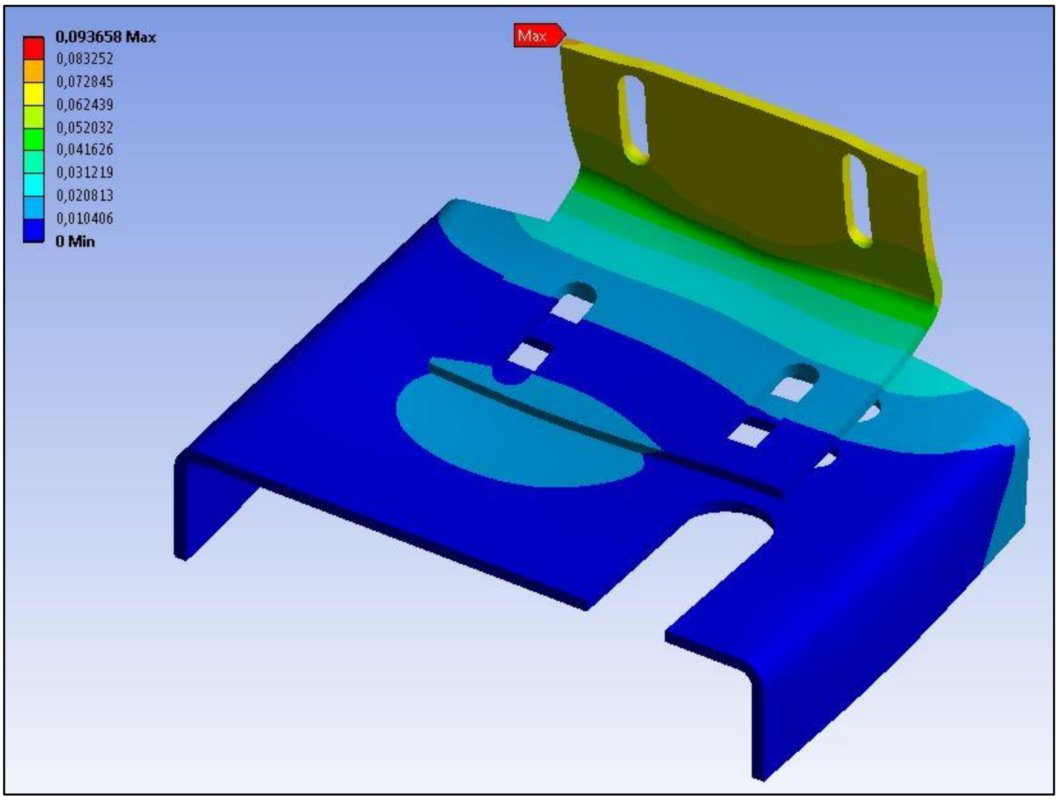
Equivalent stress in MPa, alternative view



Equivalent stress in MPa, local analysis

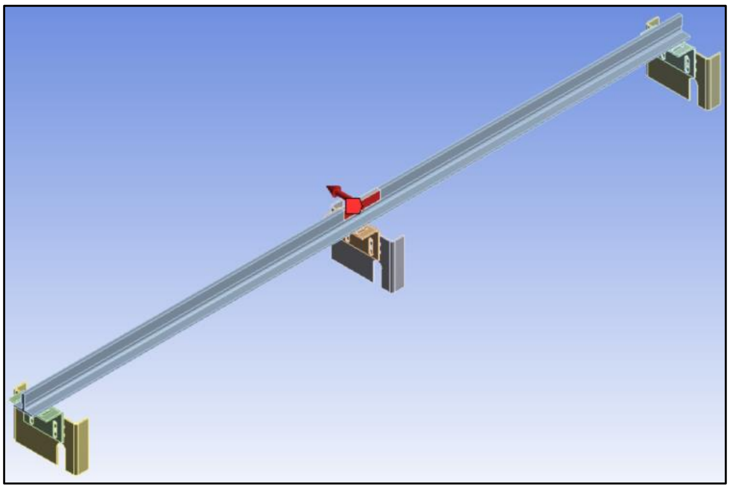


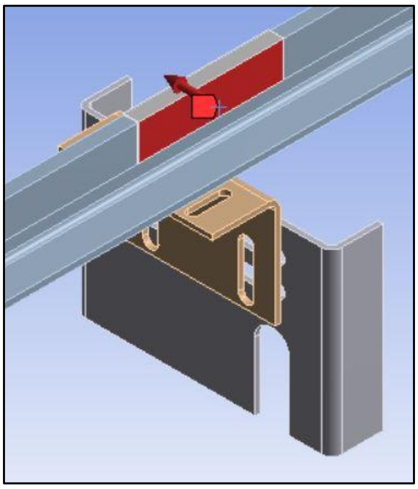
Displacement in mm, bracket



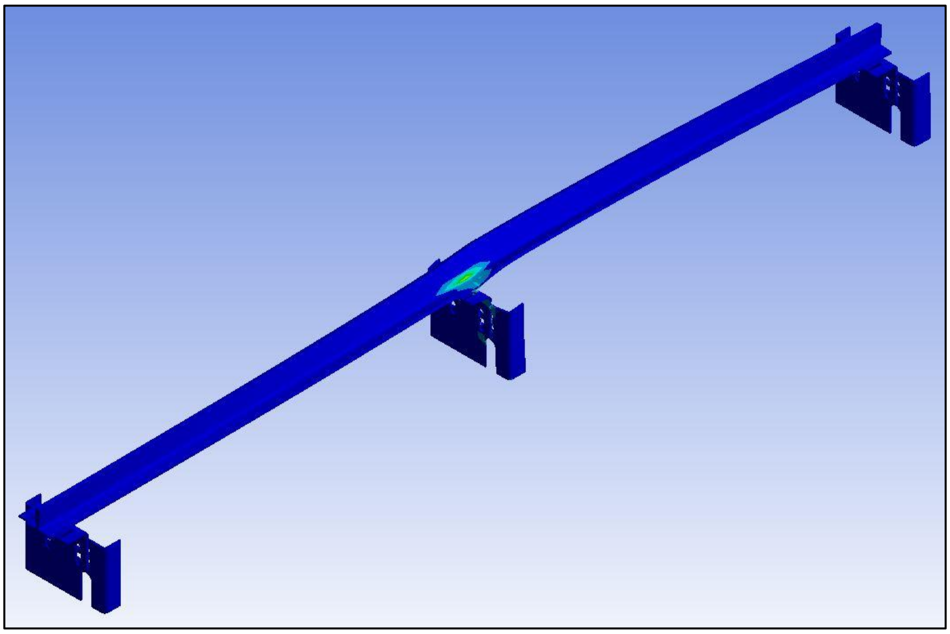
2.3 Force in x-direction

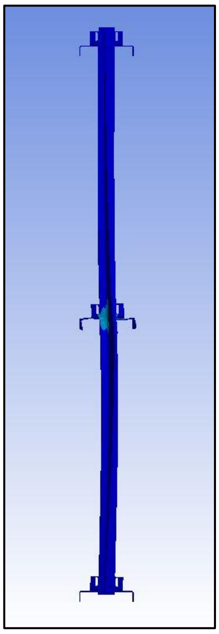
Applied force, F_x = 5,5 kN



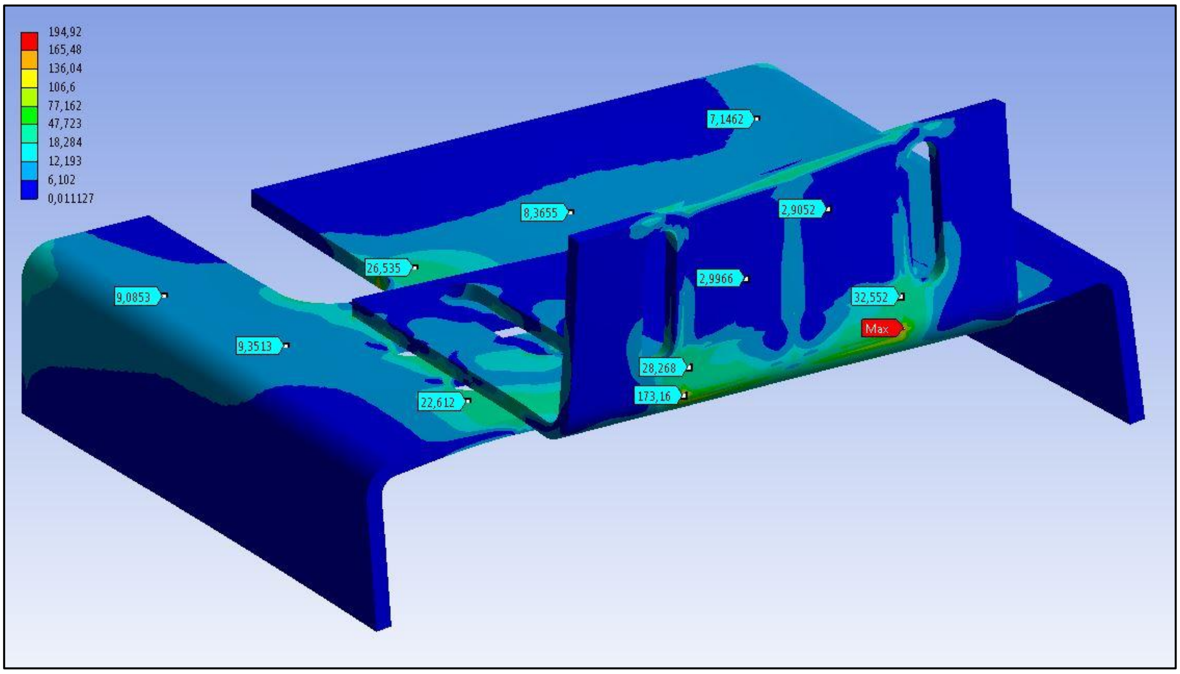


Deformation, isometric and front view

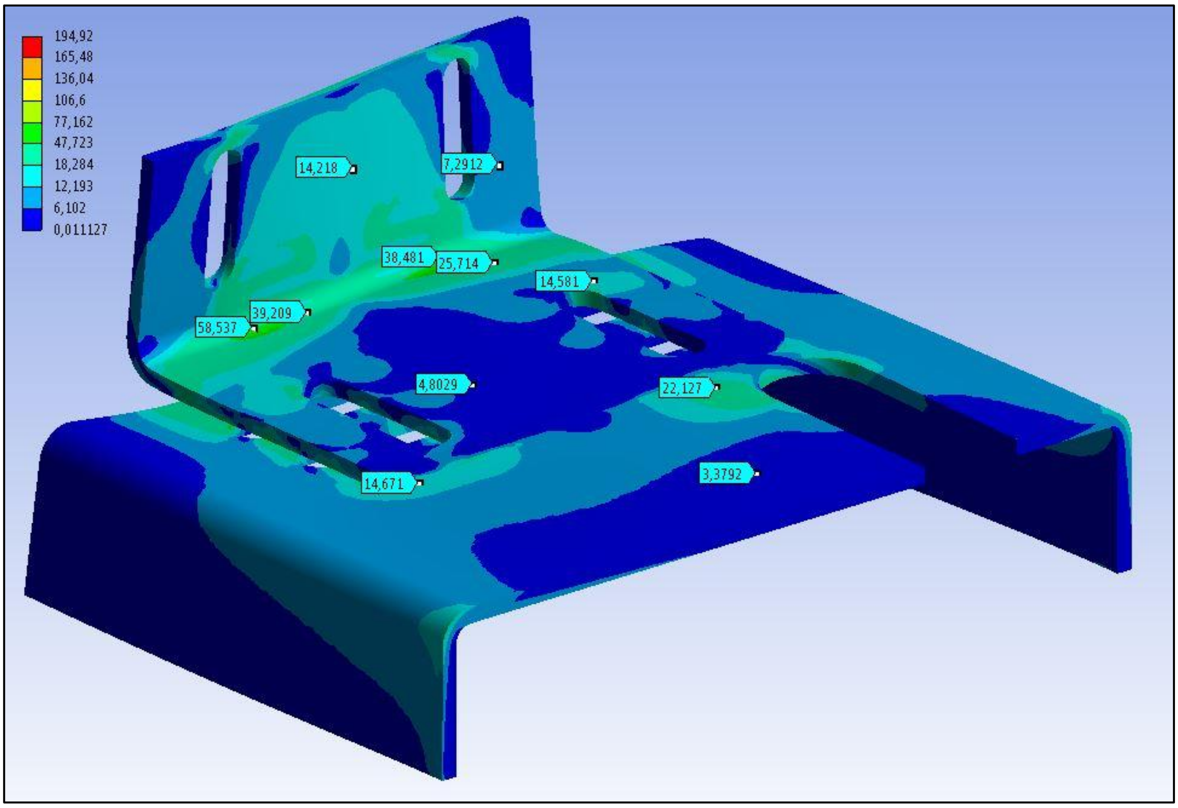




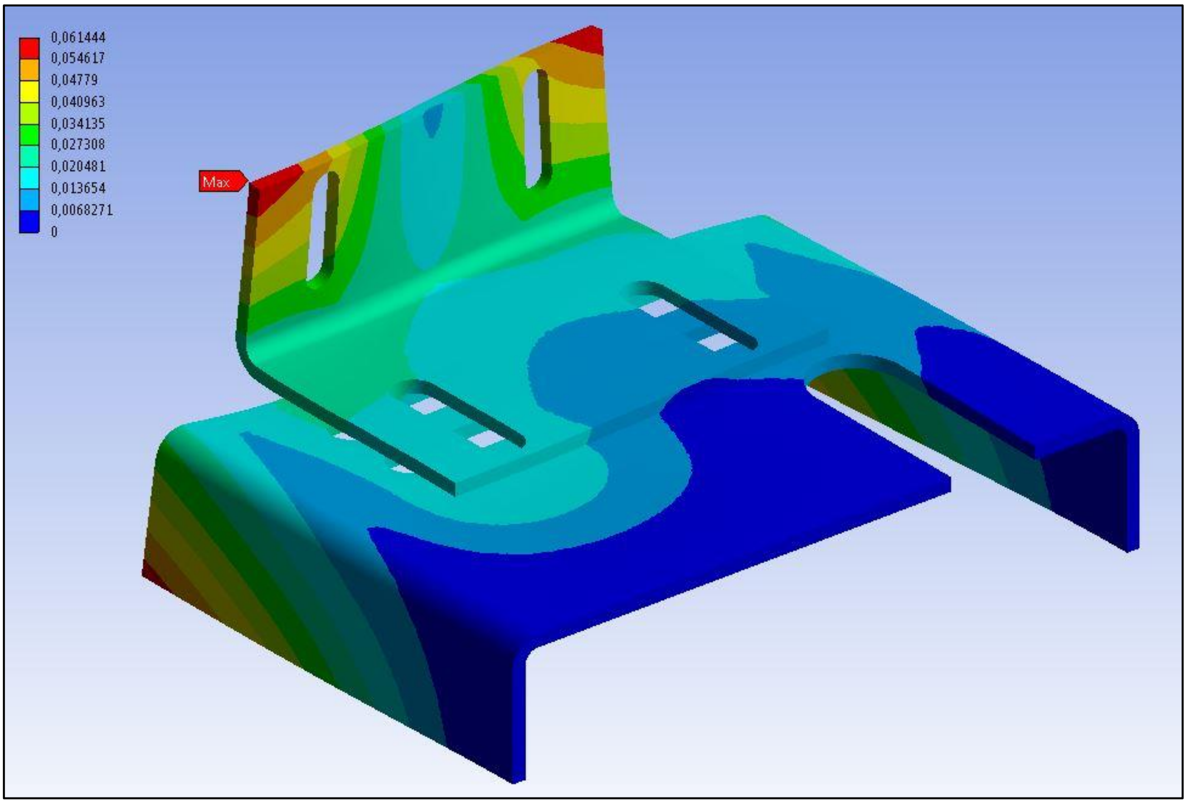
Equivalent stress in MPa, bracket



Equivalent stress in MPa, alternative view

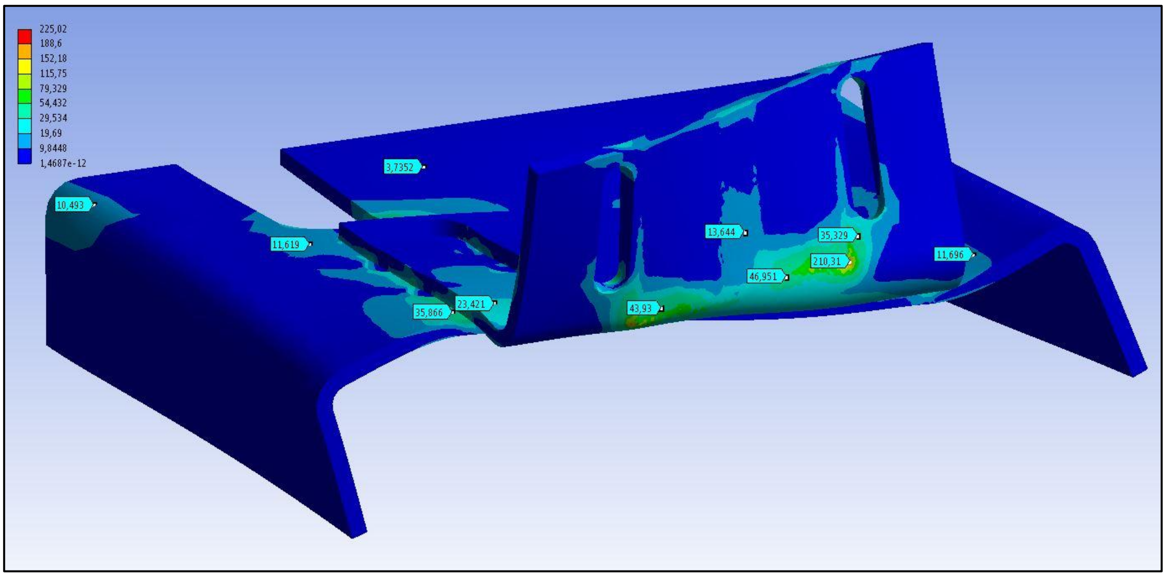


Displacement in mm, bracket

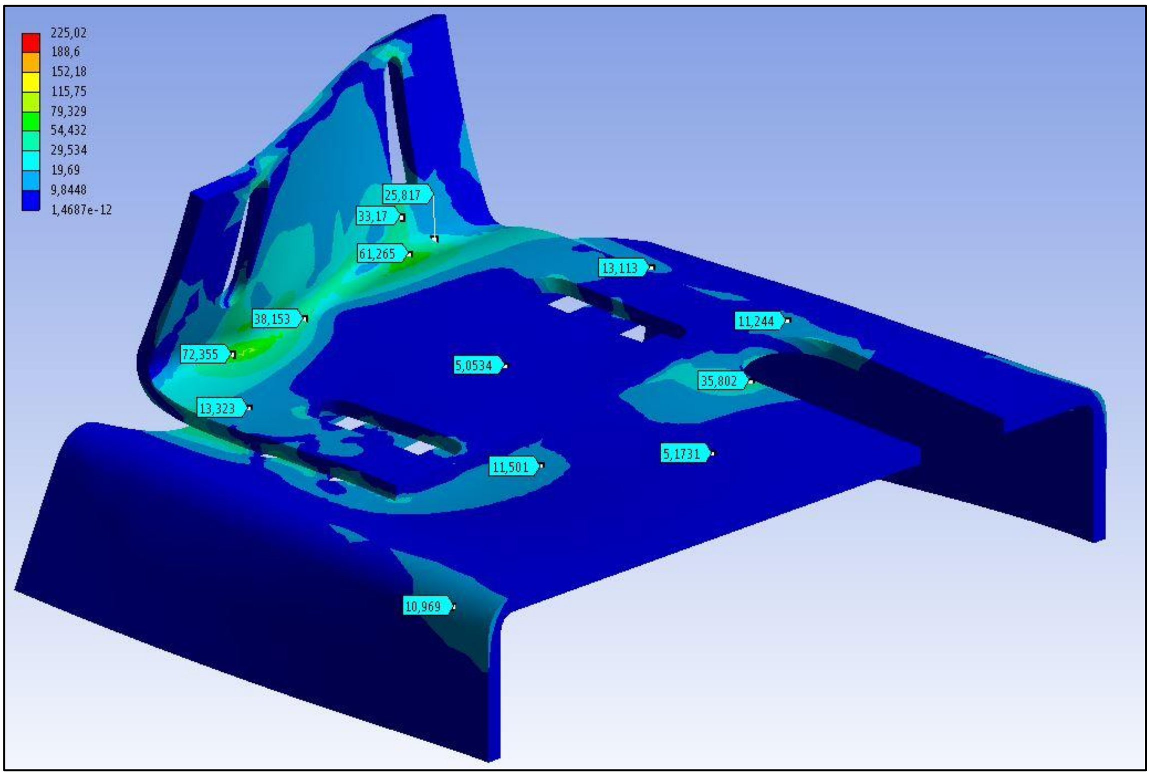


2.4 Force in x-direction, using the alternative mesh

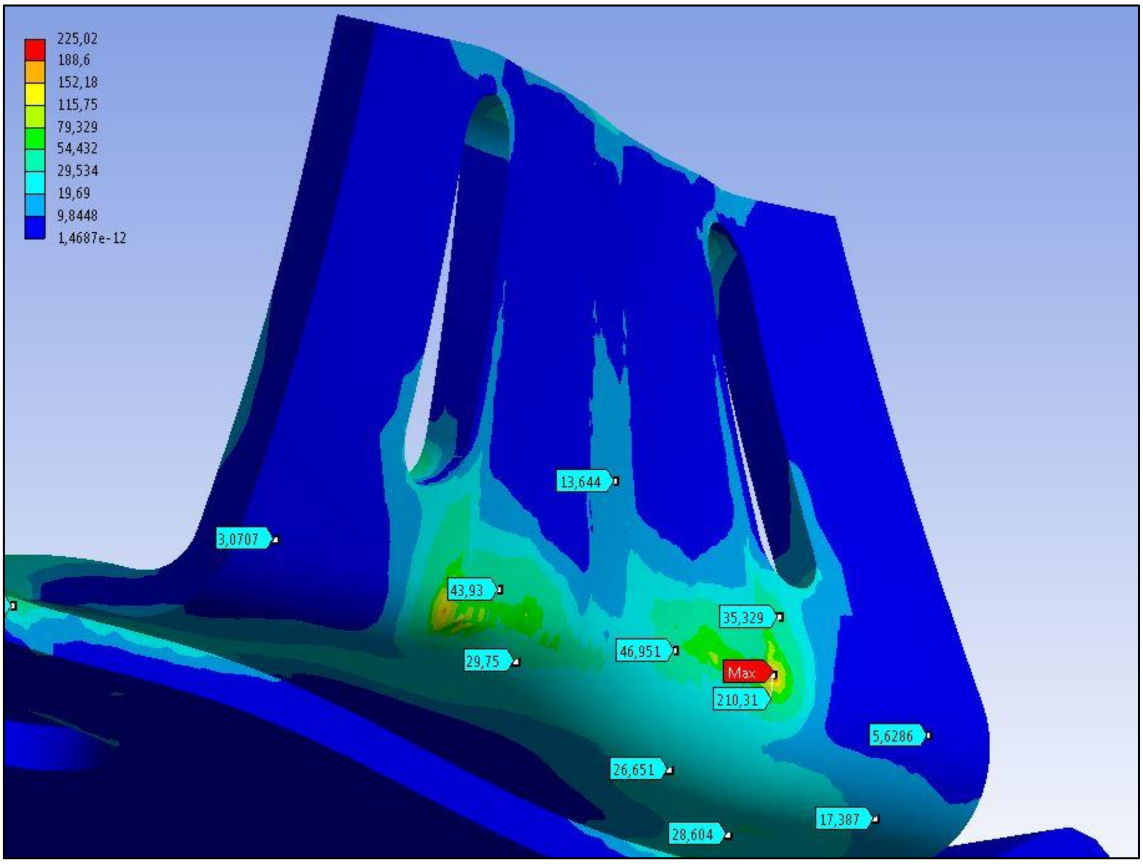
Equivalent stress in MPa, bracket



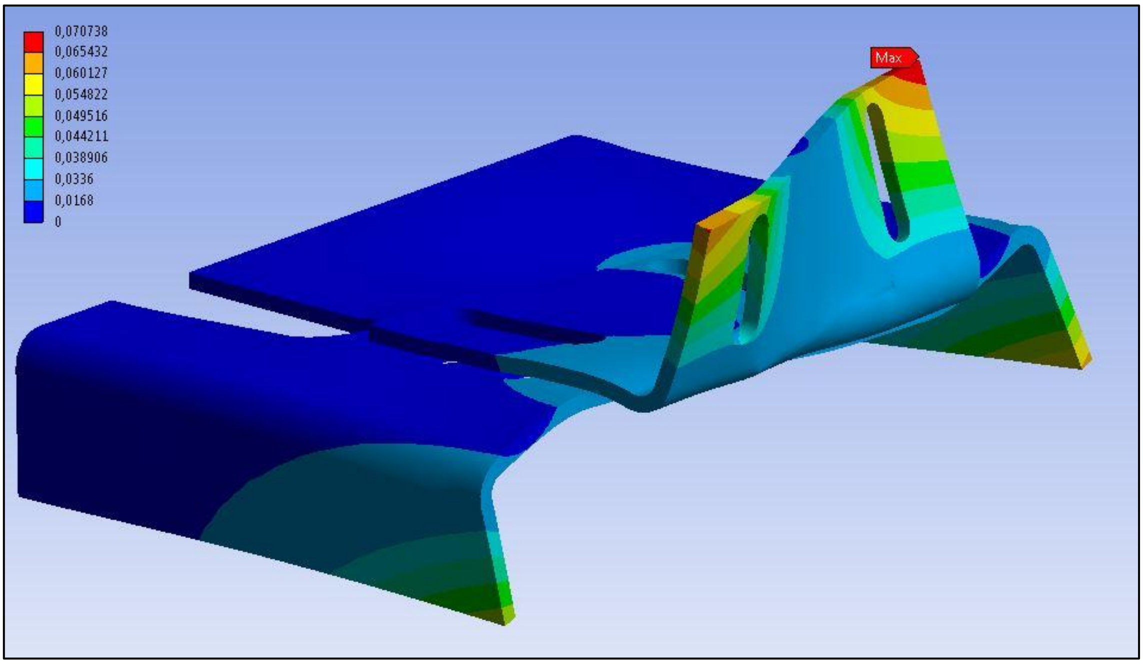
Equivalent stress in MPa, alternative view



Equivalent stress in MPa, local analysis

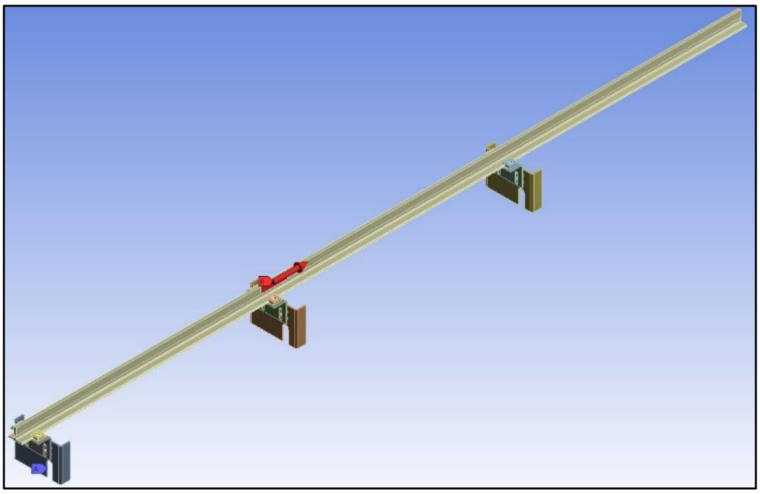


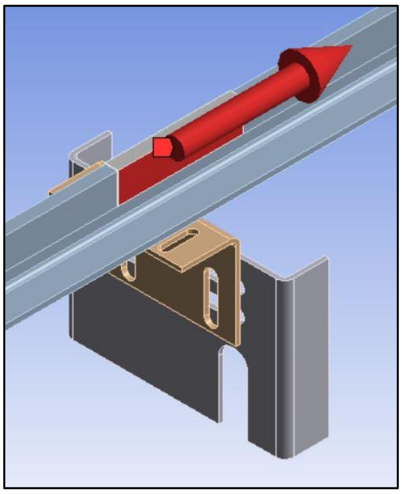
Displacement in mm, bracket



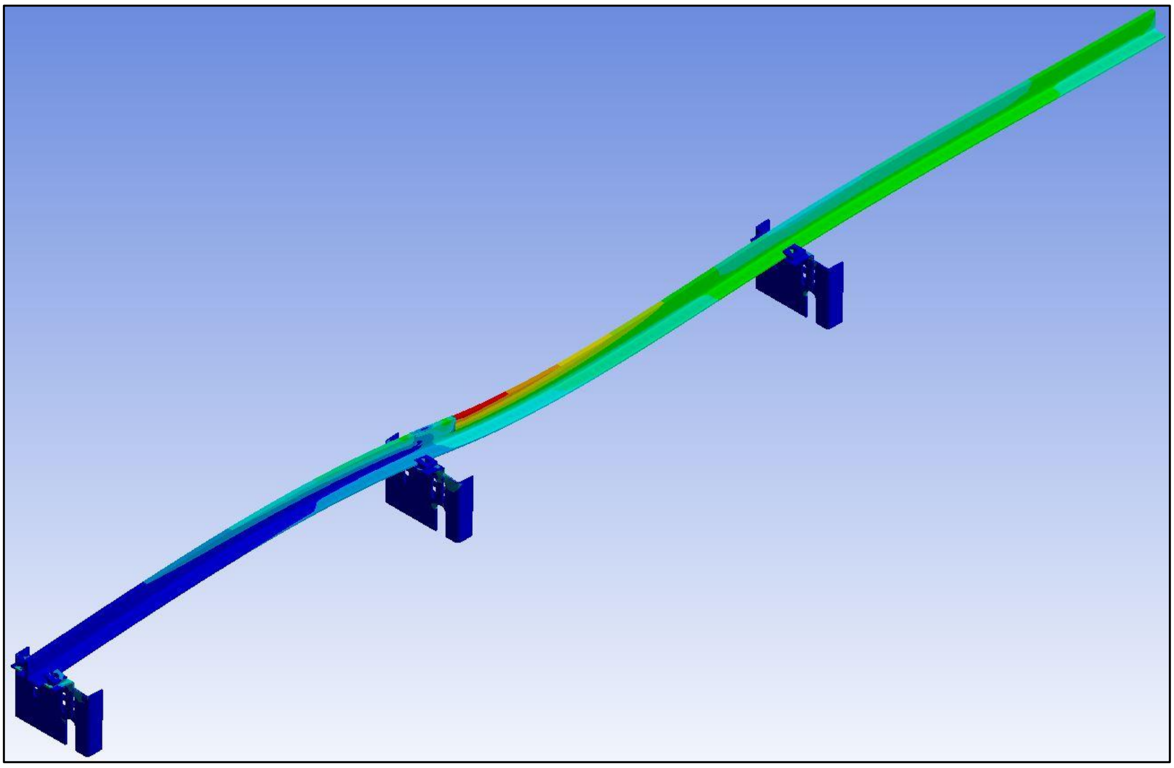
2.5 Force in z-direction

Applied force, F_z = 60 kN

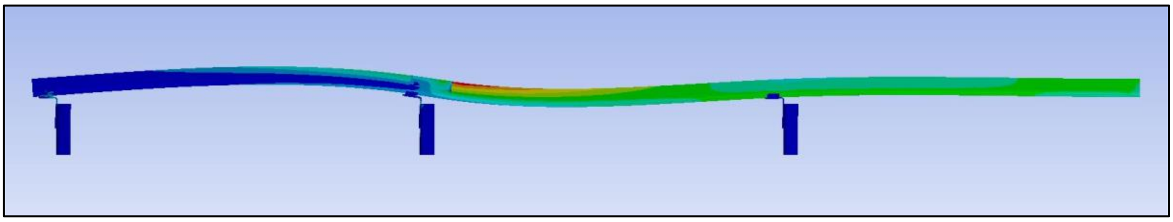




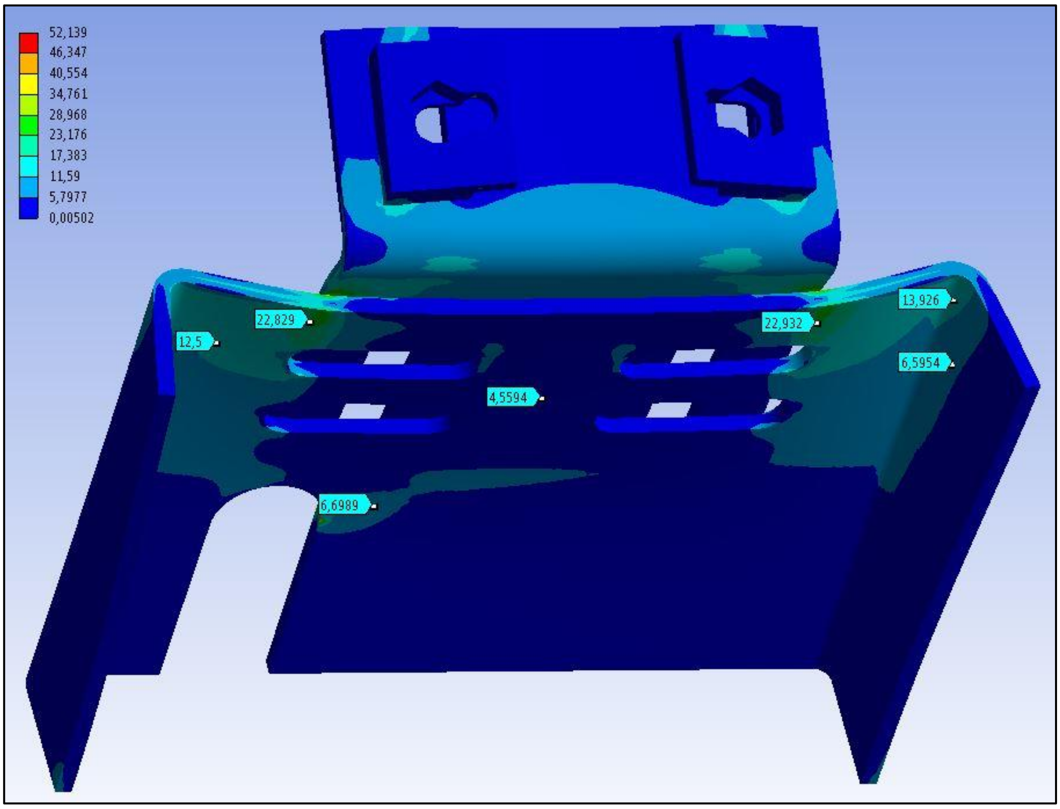
Deformation, isometric view



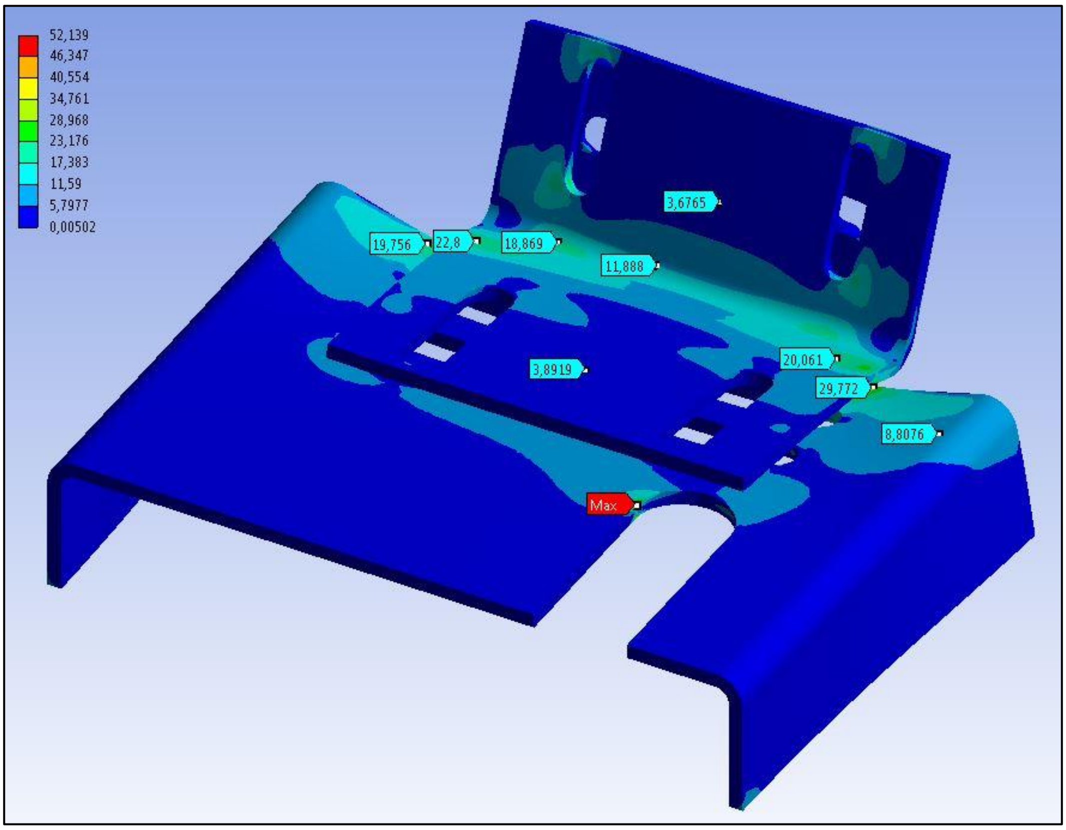
Deformation, side view



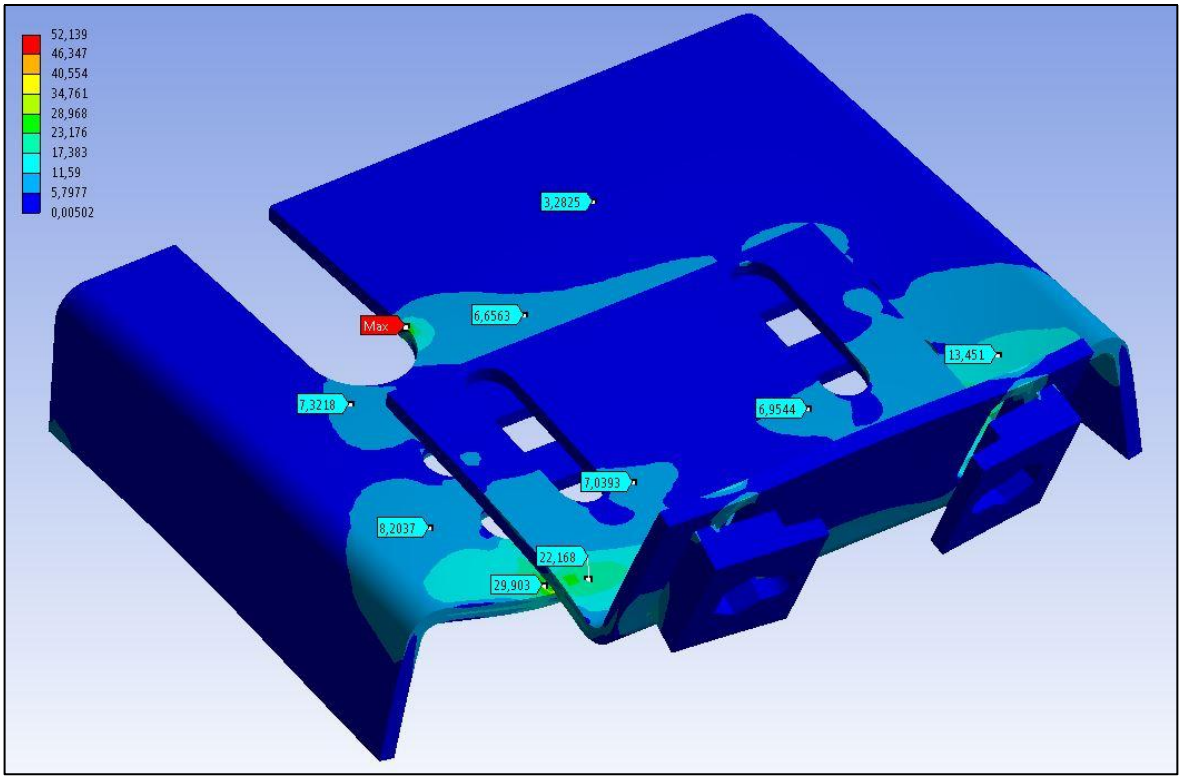
Equivalent stress in MPa, bracket



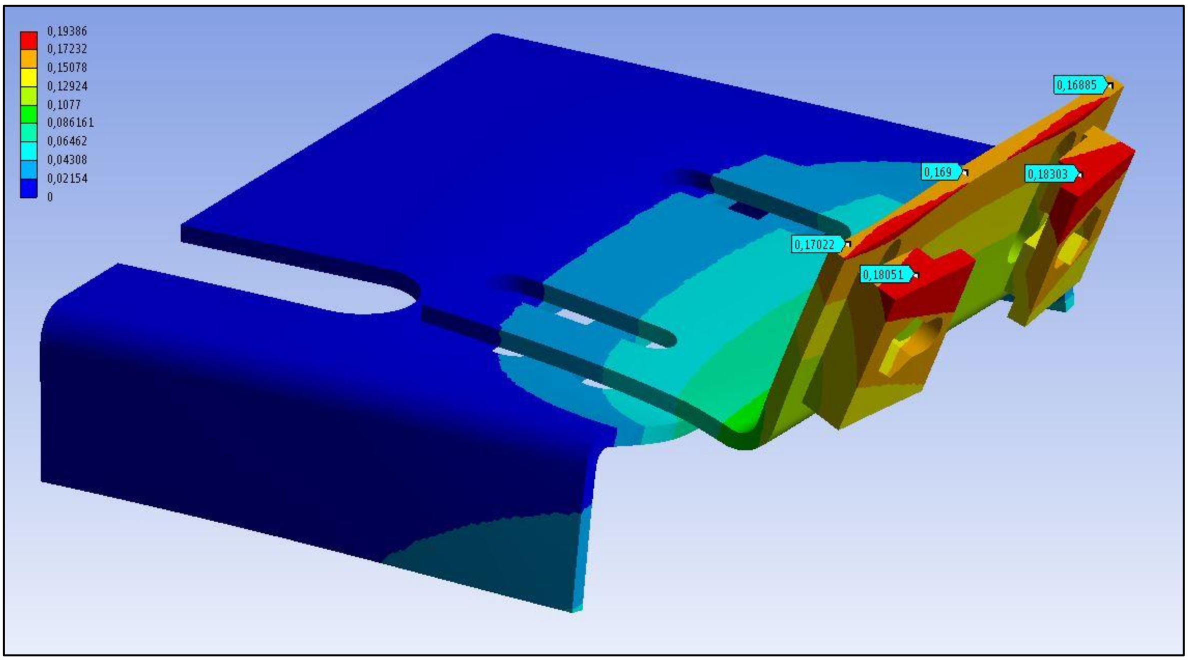
Equivalent stress in MPa, alternative view



Equivalent stress in MPa, alternative view



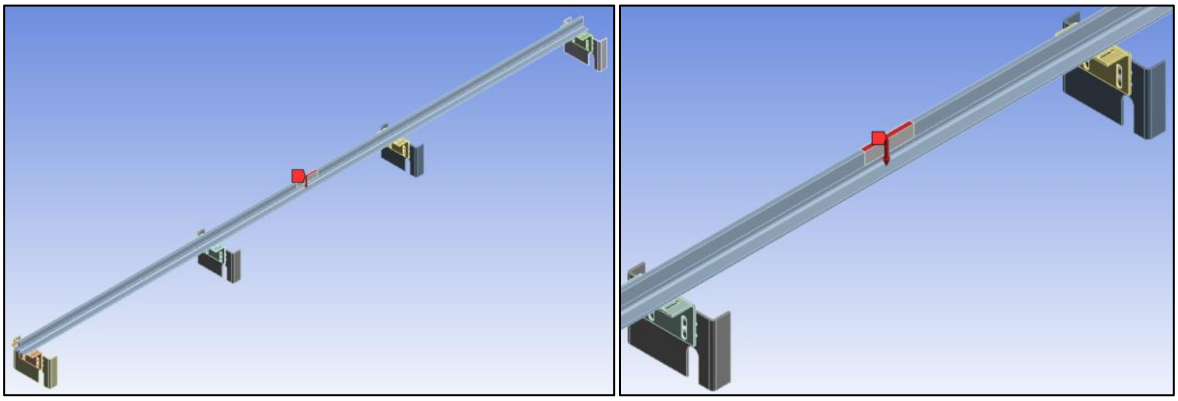
Displacement in mm, bracket



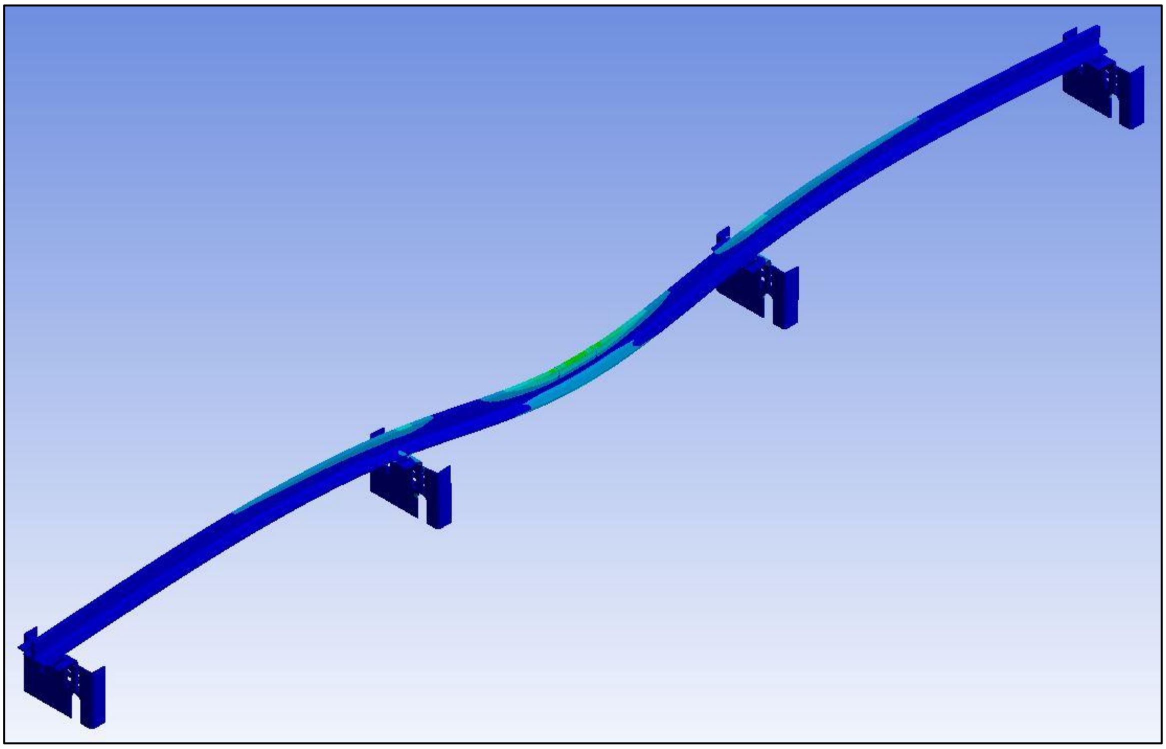
3 Forces acting between two brackets

3.1 Force in y-direction

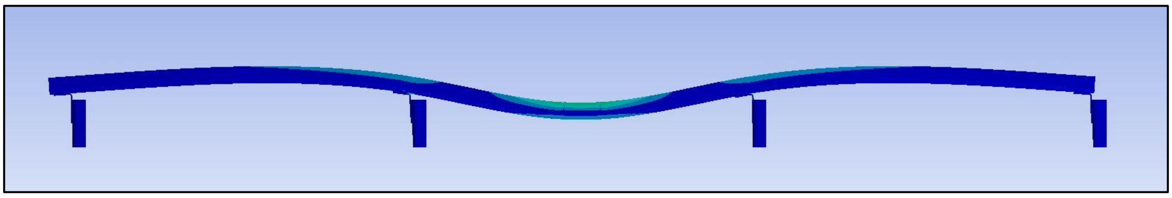
Applied force, F_y = 11 kN



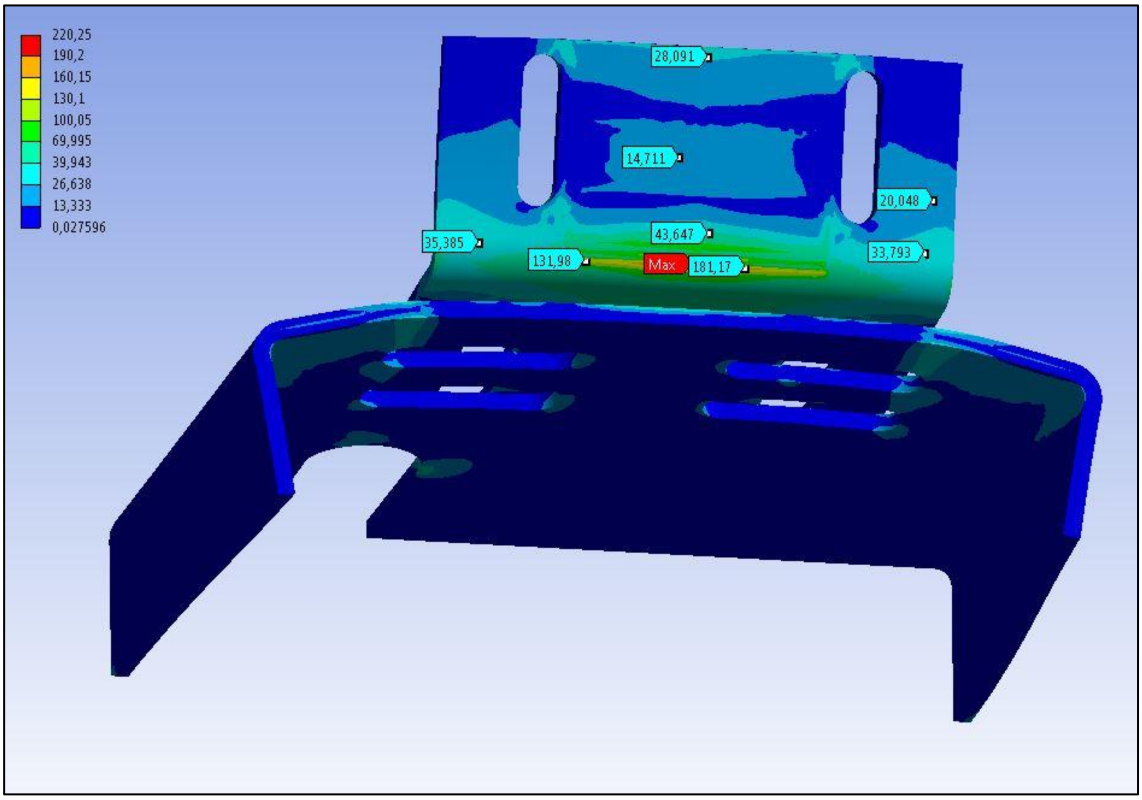
Deformation, isometric view



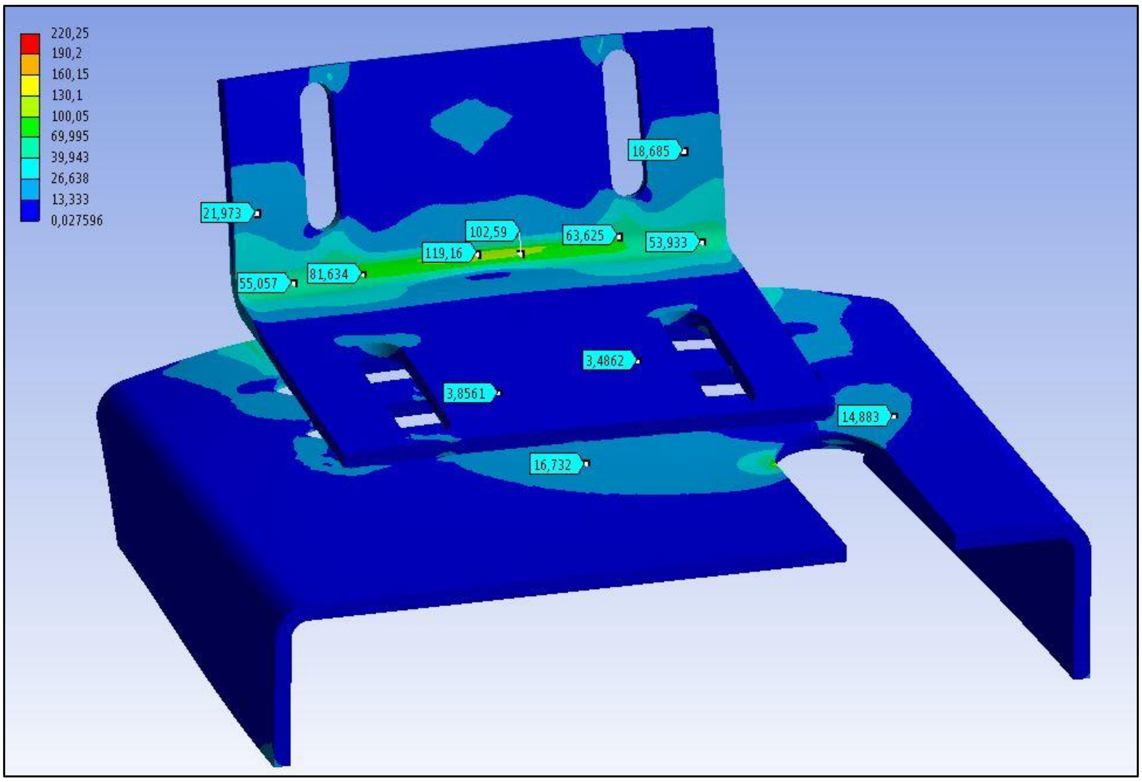
Deformation, side view



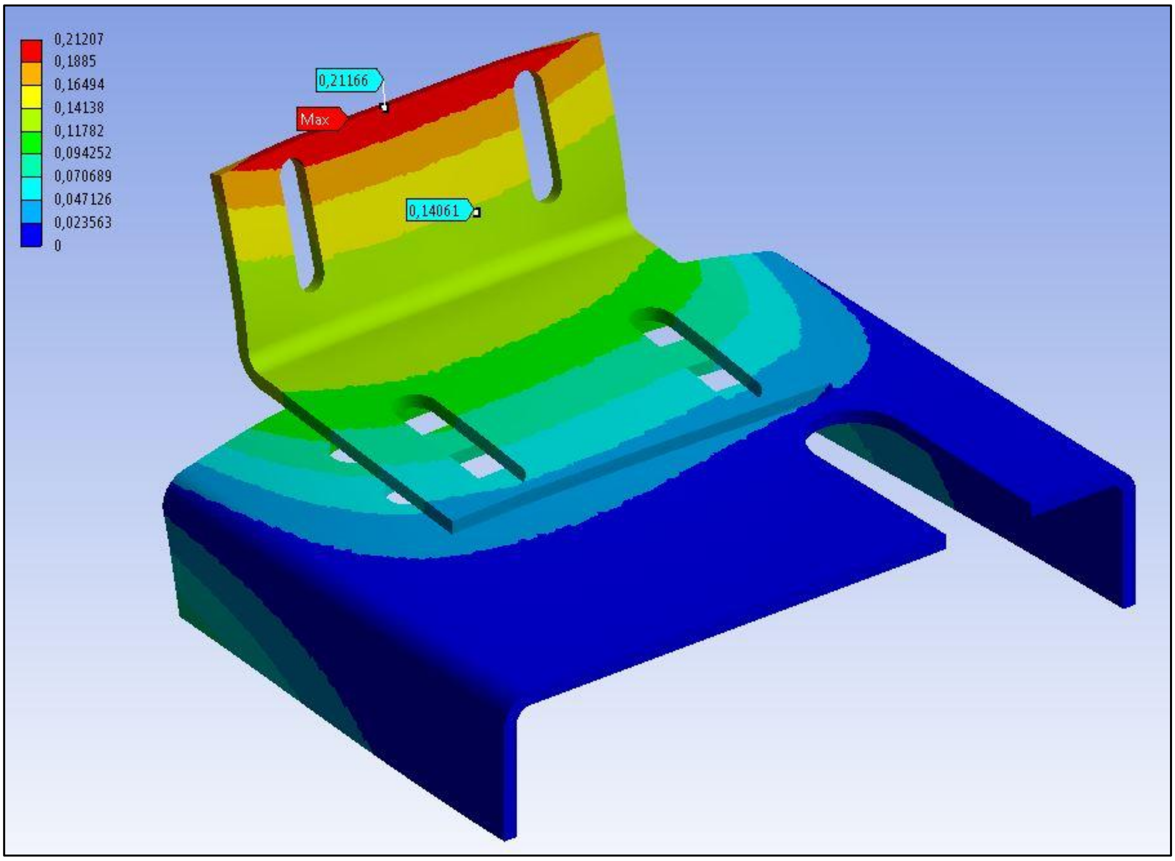
Equivalent stress in MPa, bracket



Equivalent stress in MPa, alternative view

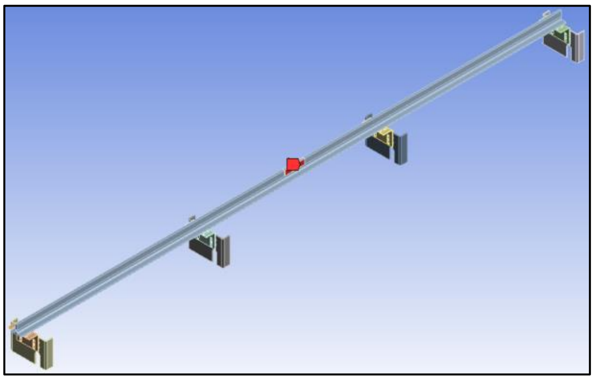


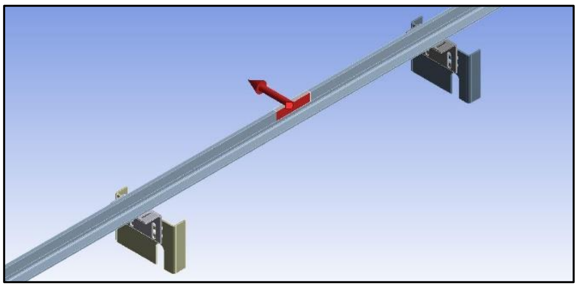
Displacement in mm, bracket



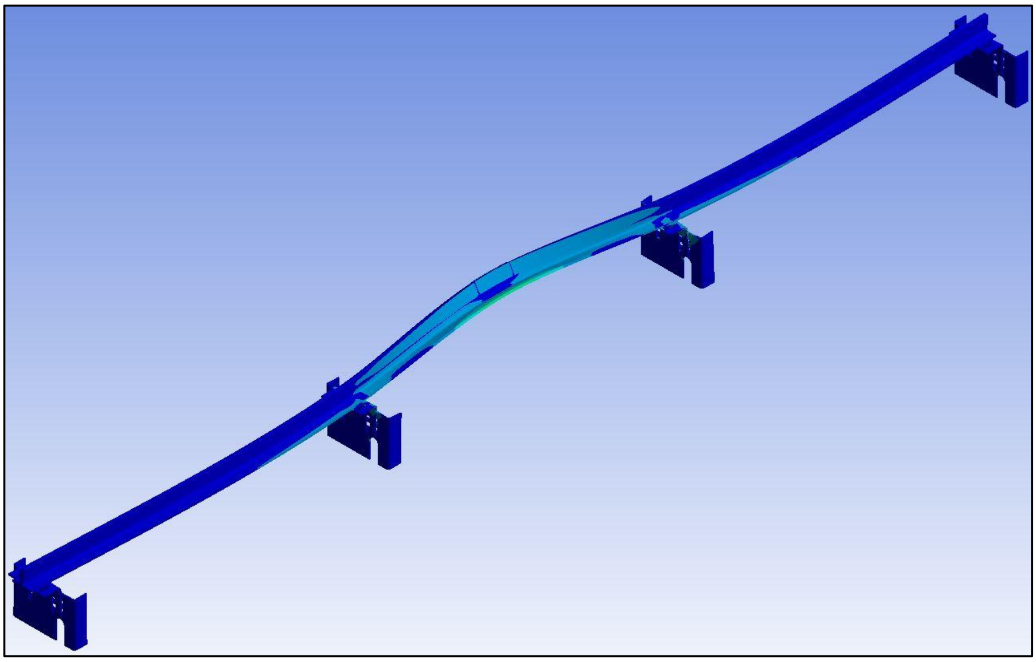
3.2 Force in x-direction

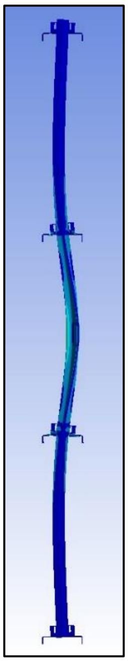
Applied force, F_x = 5,5 kN



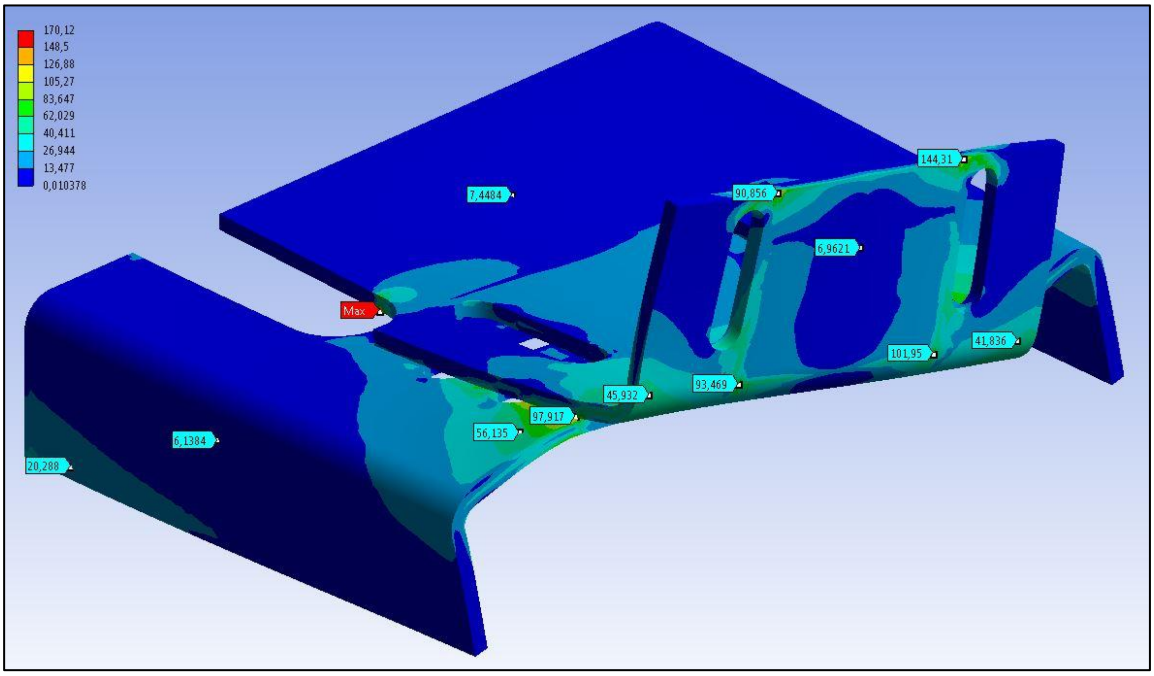


Deformation, isometric and side view

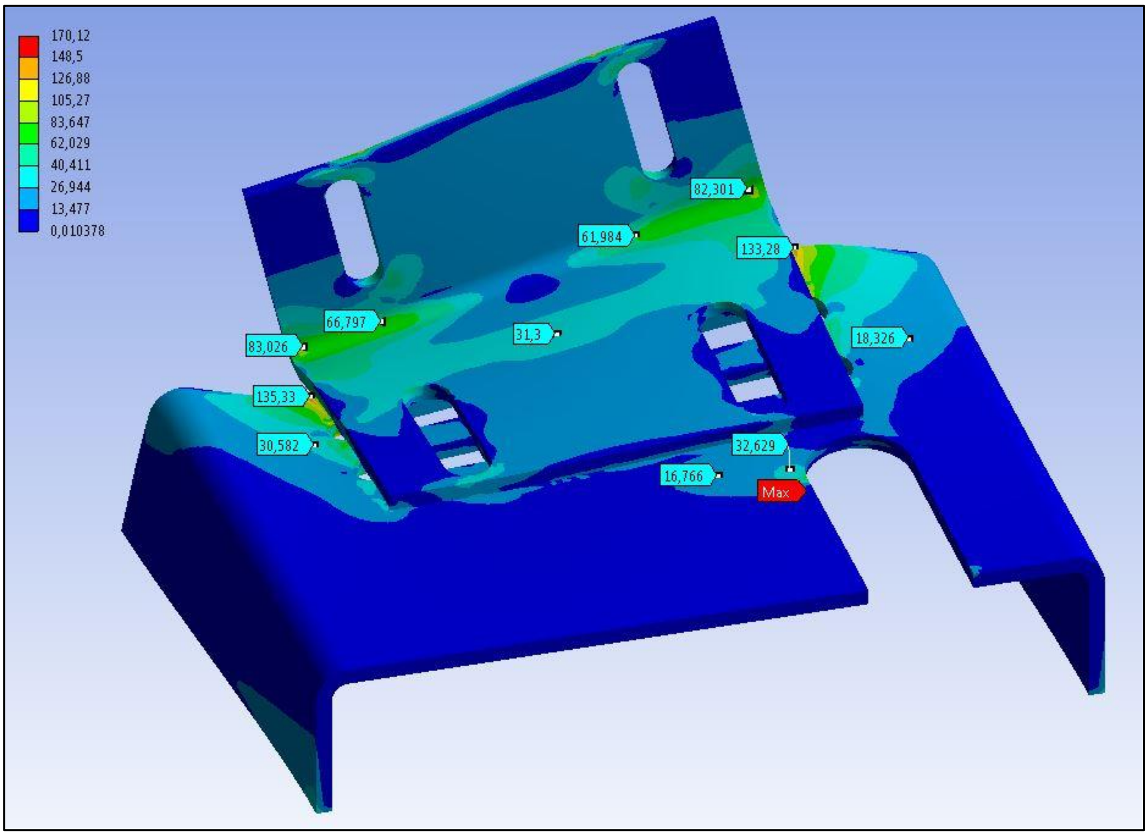




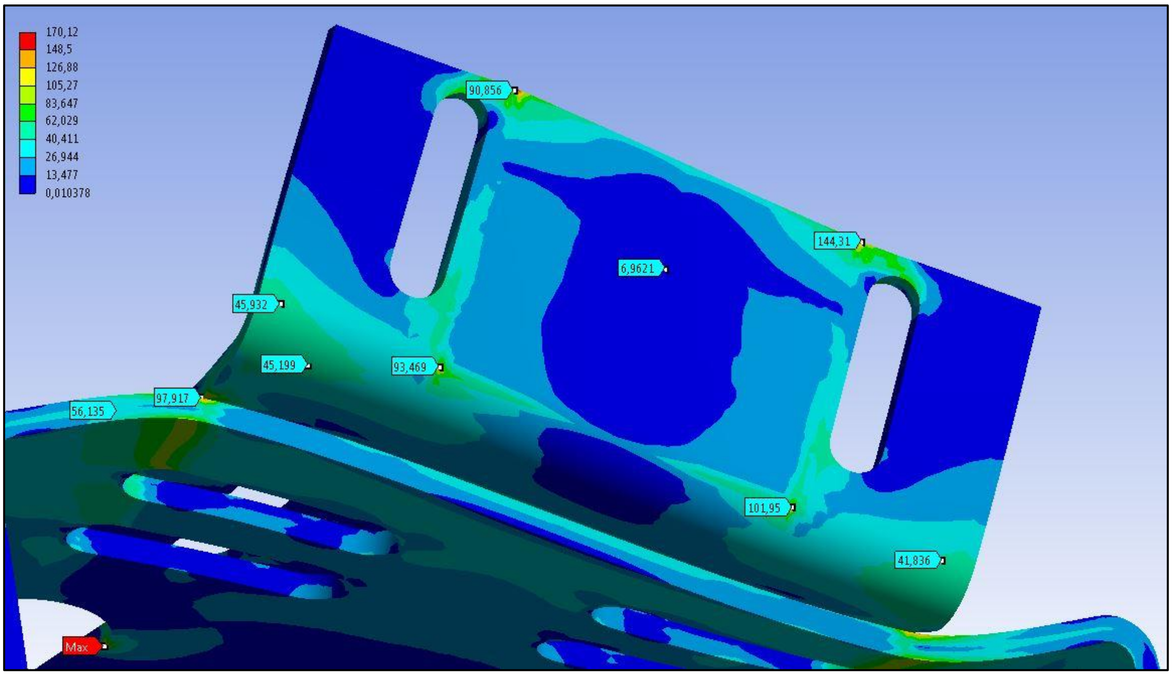
Equivalent stress in MPa, bracket



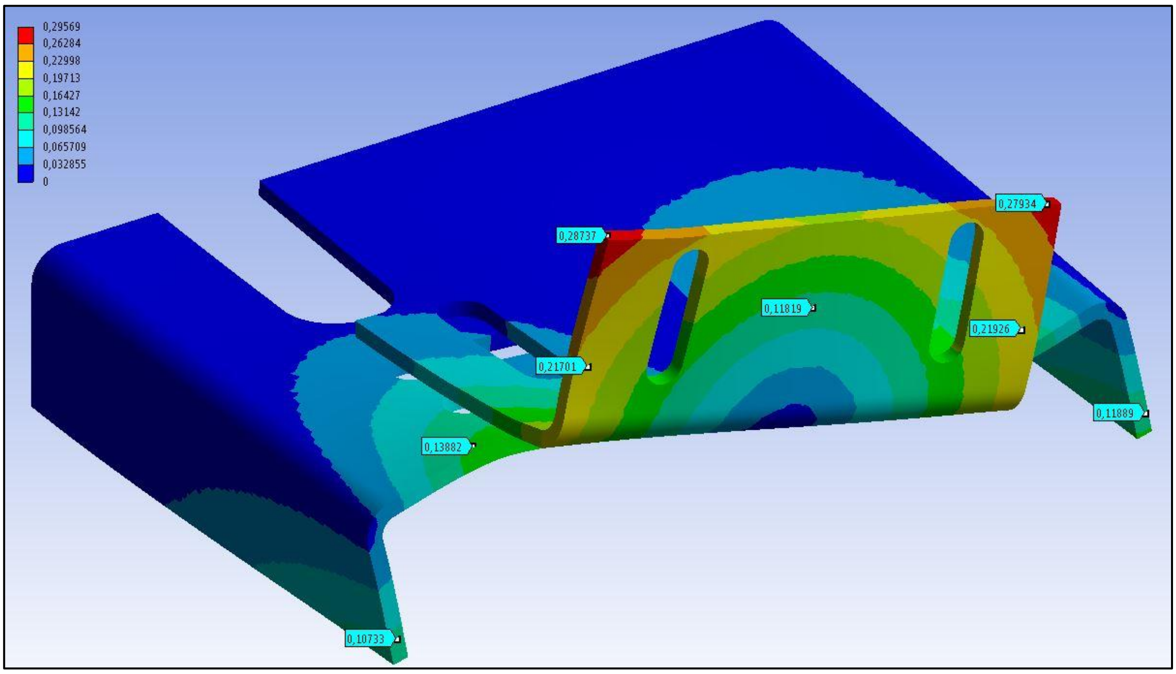
Equivalent stress in MPa, alternative view



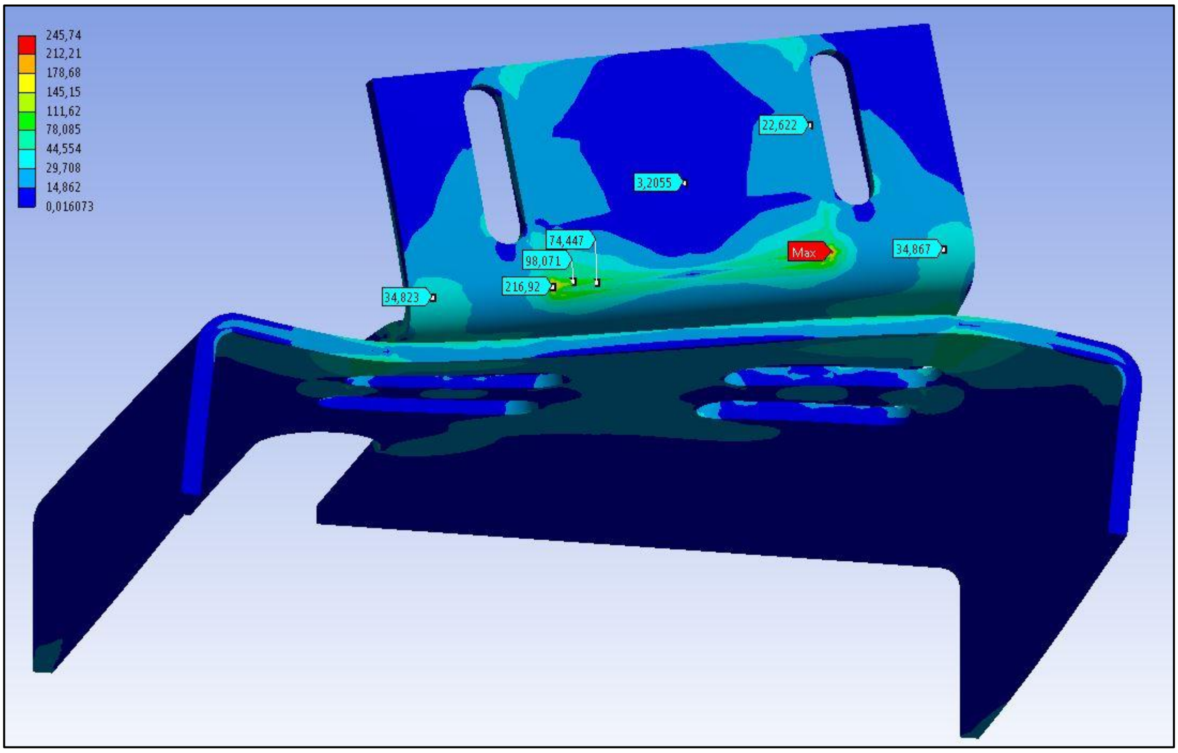
Equivalent stress in MPa, local analysis



Deflection in mm, bracket

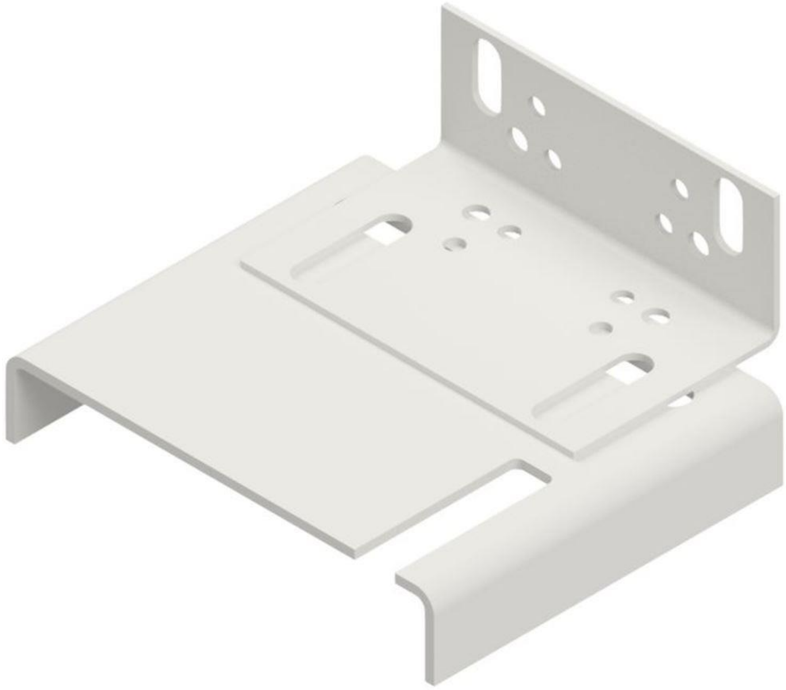


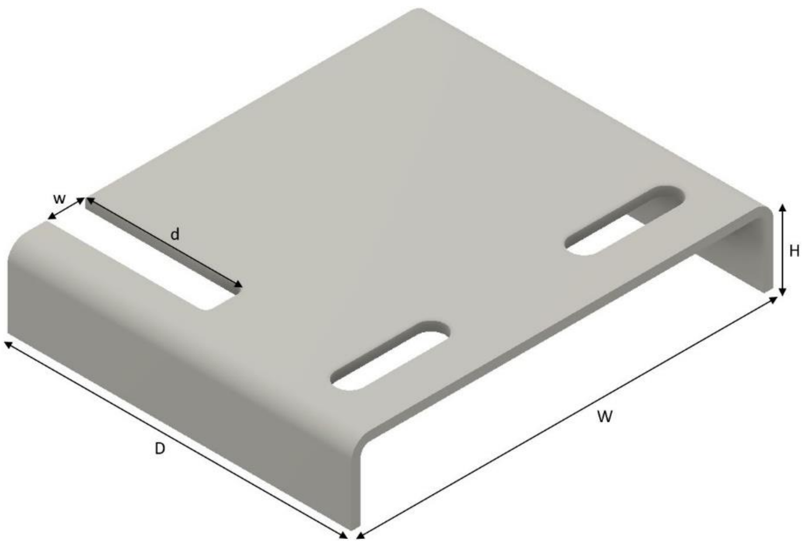
Equivalent stress in MPa, second bracket



4 Proposed design

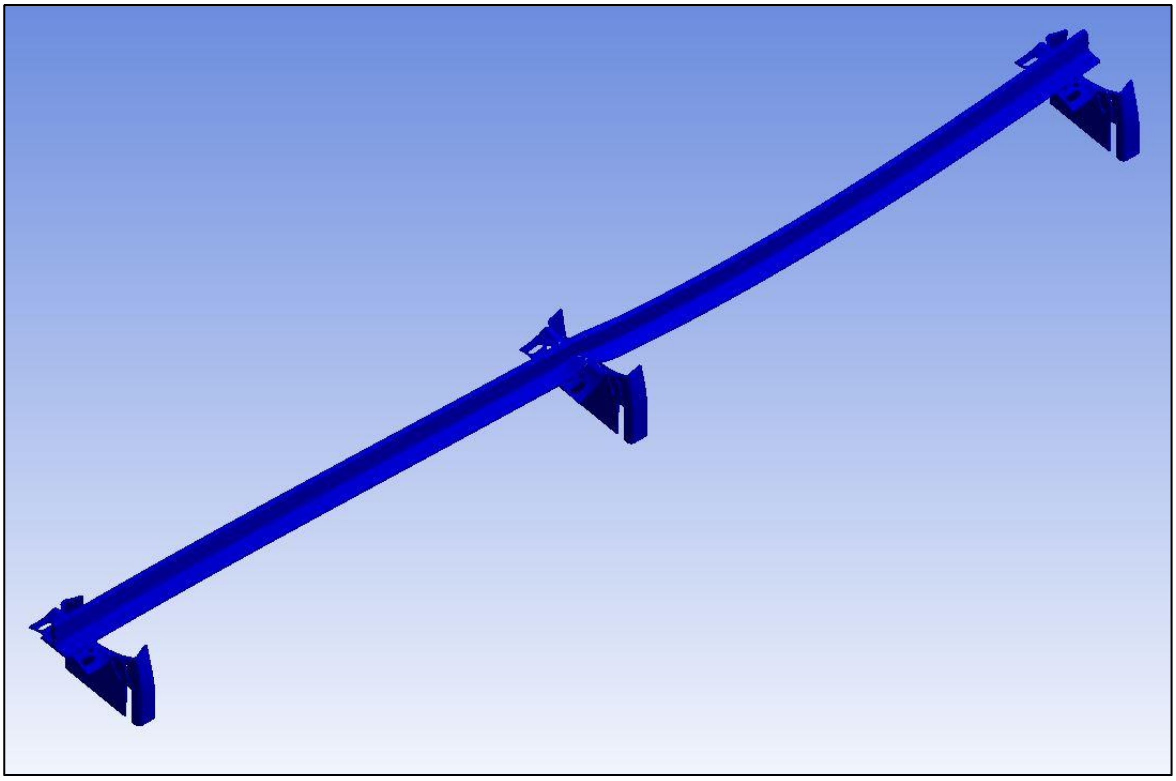
- Standard L-profile from TKA
- Customized U-channel from TK
  - Two holes for bolt connections
  - Rectangular cut out with radius in edges
  - Reduced width and plate thickness



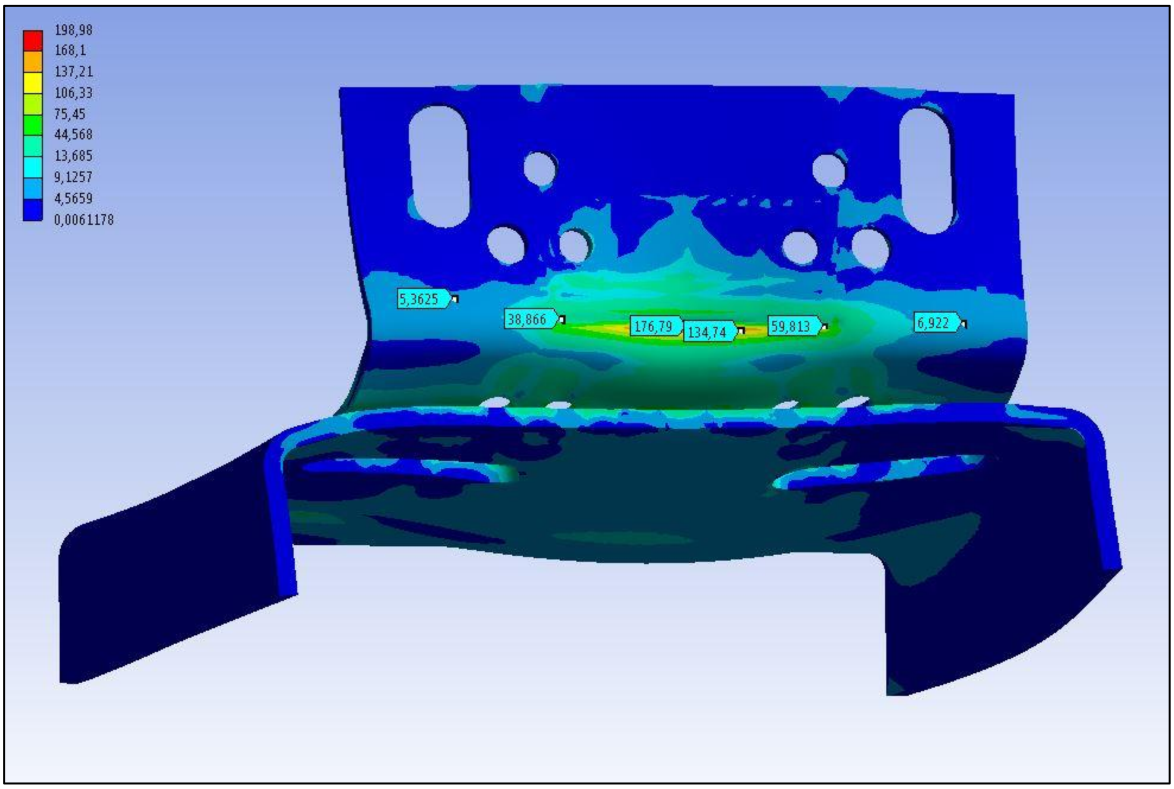


4.1 Force acting on one bracket

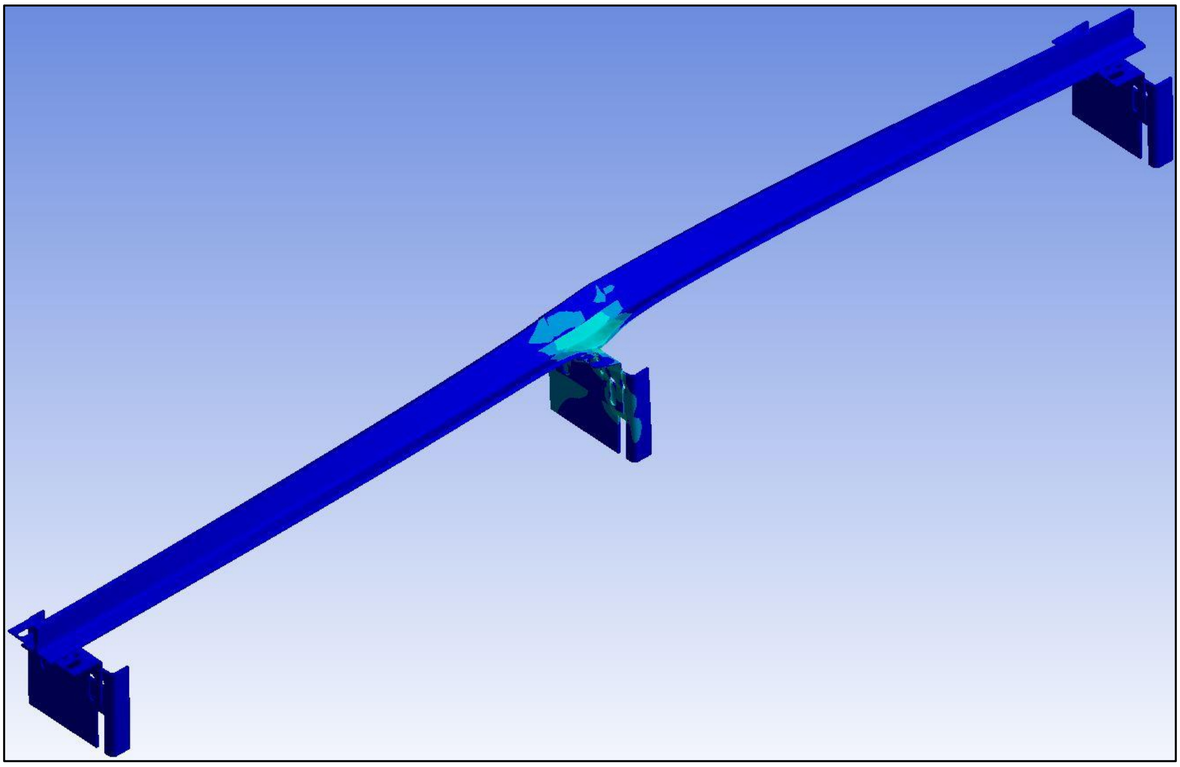
Deformation, F_y = 11 kN



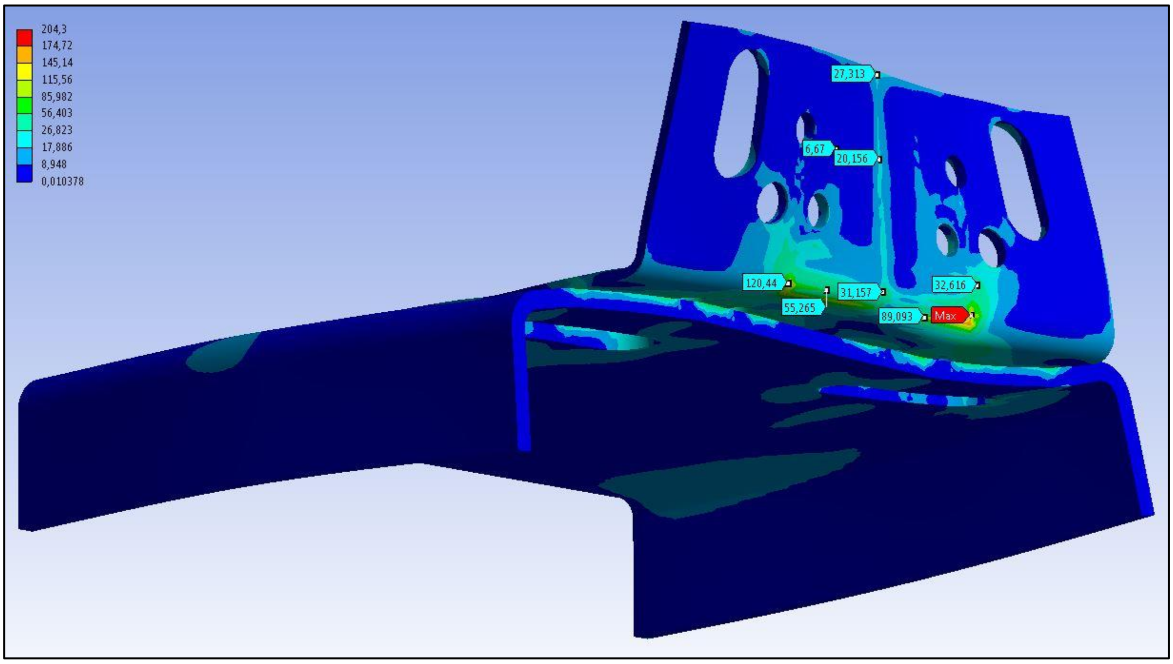
Equivalent stress in MPa, bracket



Deformation, F_x = 5,5 kN

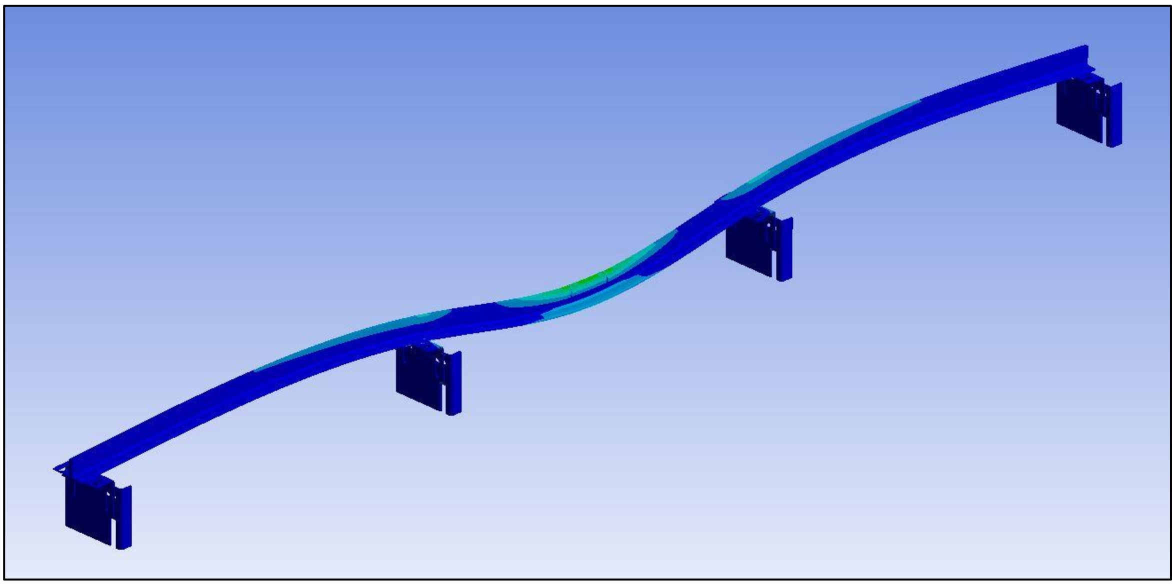


Equivalent stress in MPa, bracket

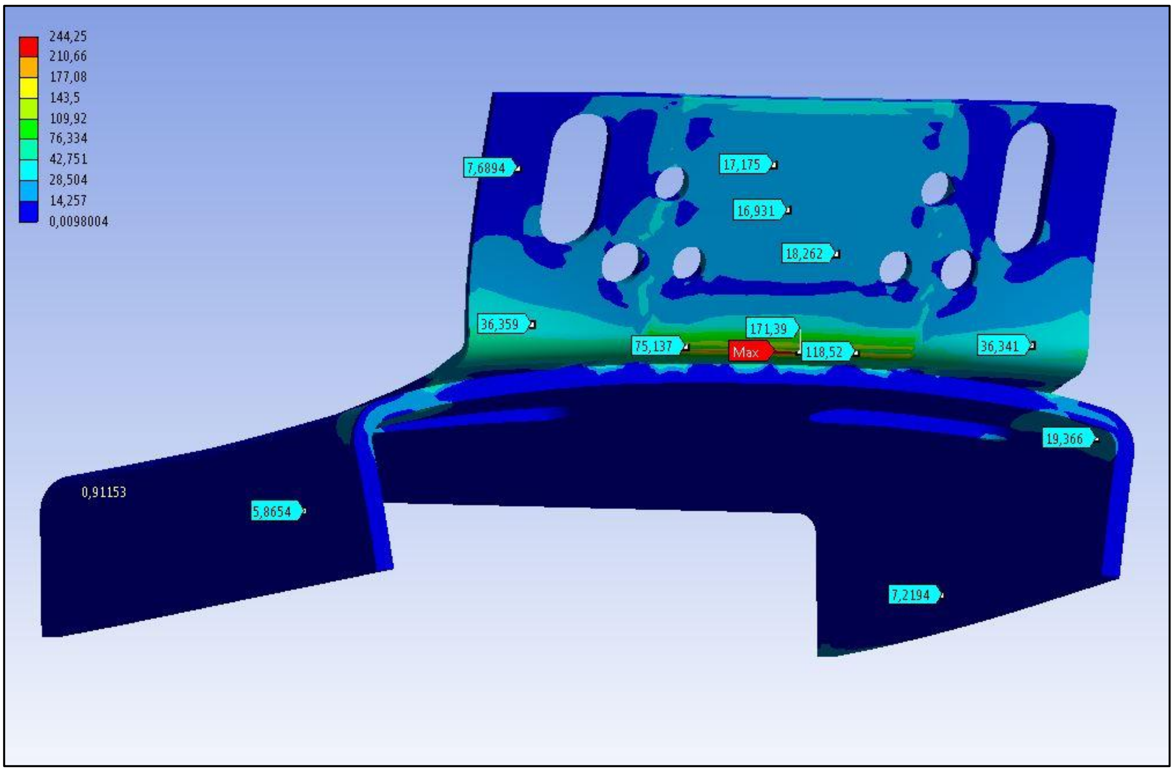


4.2 Force acting between two brackets

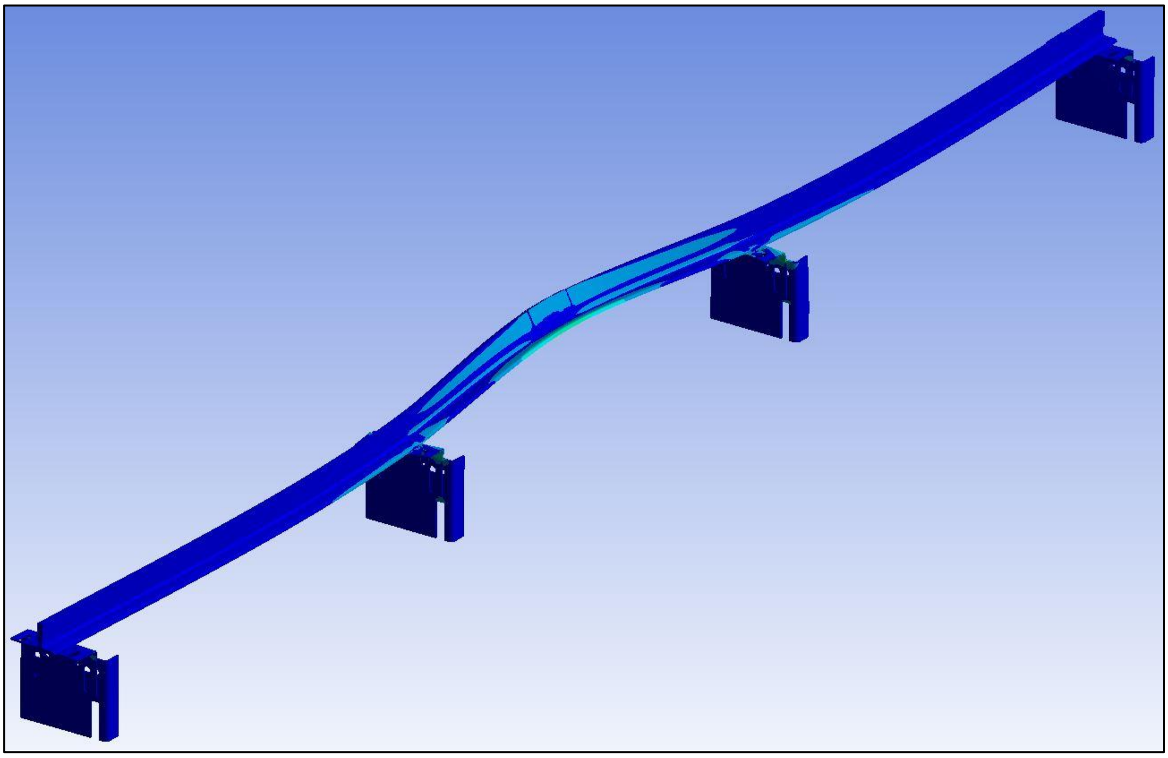
Deformation, Fy = 11 kN



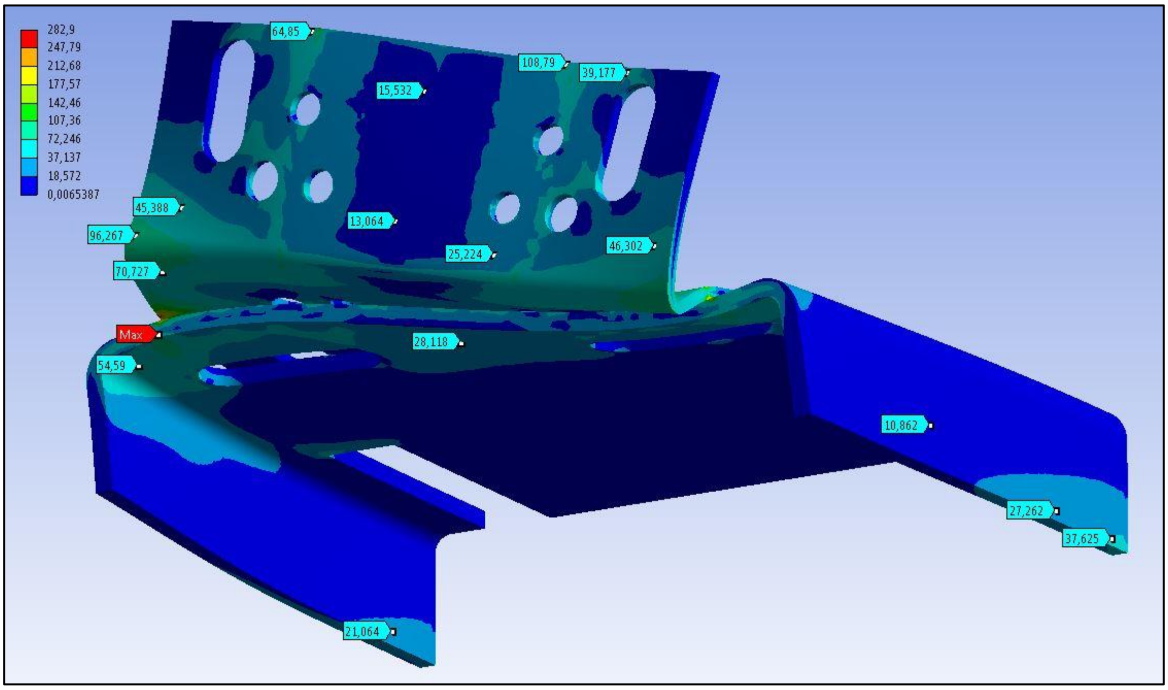
Equivalent stress in MPa, bracket



Deformation, Fx = 5,5 kN



Equivalent stress in MPa, bracket



APPENDIX F

Tentative title and abstract for an article

Publication of Article

Article/paper title

Optimization of Guide Structure for Elevators on Ships

Abstract

The existing design of marine and offshore elevators are mainly based on the expertise out of land-based elevators, with higher safety factors and all existing requirements being fulfilled. In the design of the guide structure, it is usually selected large guide rail dimensions and customized brackets of thick steel plates to account for the unknown load cases. This leads to increased weight of the structure, as well as more cost of material and installation. The cost is always important for the customer, but the weight of the components has also been a major focus the recent years. Less weight would allow more passengers and goods aboard the ship, which is especially important for cruise ships where each passenger results in more income. Together with the cost and the functionality of the product, it is therefore an attractive feature when selecting the elevator supplier. Research on the resulting forces caused by the ship motions has therefore been done, to increase the knowledge of guide structures for elevators operating in the maritime and offshore environment.

A thorough literature study on current rules and requirements for elevators on ships, with special attention to the guide structure, has been carried out. Four different methods for calculating the load case acting on the structure are developed from this. The first method is based on the theory of free harmonic vibration, while the other three are based on rules, requirements and equations defined by DNV GL and Lloyd's Register. Furthermore, a mapping of load cases has been done to evaluate the forces acting on the guide structure caused by the different ship types. This was solved by dividing the ships into three different classes based on similarities in type, dimensions and resulting motions.

The optimization of the brackets involves two different solutions; a standard bracket developed for land-based elevators and a customized bracket developed for offshore elevators. Offshore elevator trunks are built with bulkhead stiffeners on the inside or outside, depending on the ship designers. The stiffeners are in most cases placed inside the trunk, because it is more usual to use this space for insulation instead of placing it in the adjacent corridor. The standard bracket is easier to install, and would be the preferable solution given that the trunk is built with the stiffeners on the outside. Both solutions were evaluated with the use of a structural FEM analysis. This was done to verify if the standard brackets are applicable for offshore applications, and to optimize the customized brackets in terms of weight and installation cost.

The calculation methods are tested on different ship types, and the result shows that it is possible to reduce the guide rail dimensions. Most of the evaluated ships are installed with T127 guide rails for their elevator car. The calculation methods do however show that T90 guide rails could be used instead, since this is the recurring dimension from all the ship examples. Furthermore, it is recommended to use the standard bracket given that its applicable for the trunk layout. The FEM analysis verified that the solution could be used for offshore elevators up to 4000 kg. It is also suggested to reduce the plate thickness and width of the customized bracket, since it still would be within acceptable safety factors. The reduction of guide rail dimensions and usage of the standard bracket could save up to 40 % in cost of installation, based on the examples given in the paper.