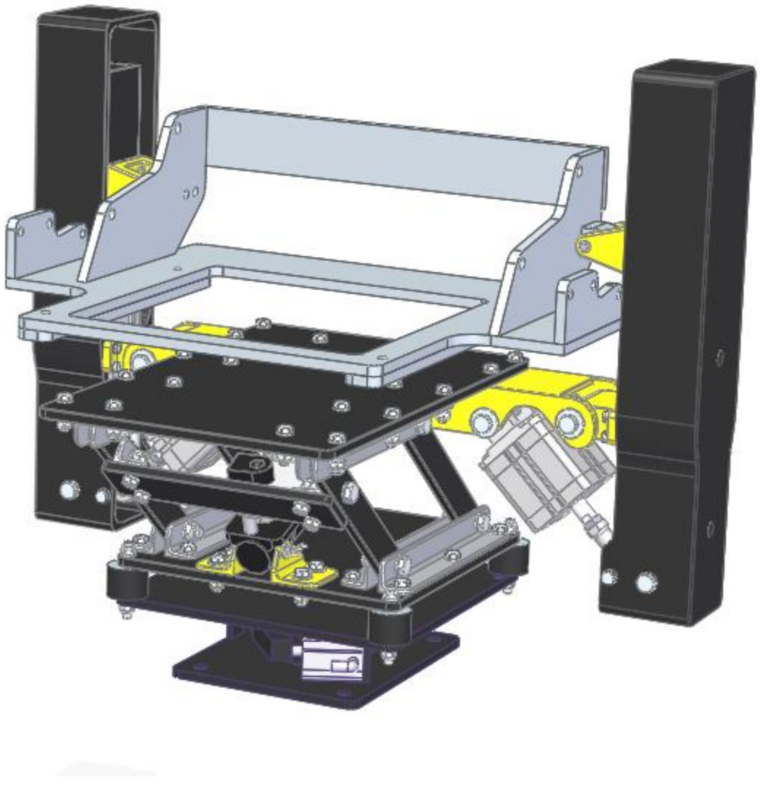
Development and realization of an operator seat with active suspension

Ahmed El Shobaki



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Abstract

In this thesis work, generated design concepts during previous thesis work performed by Doroftei Teodor and Osorio Omar. in 2013 for an operator seat for forestry machines, have been examined. This examination was made in order to design a full-scale prototype of an operator seat featuring active suspension. With the help of this analysis, a proposal for a prototype, which decreases the effects on the operator caused by vibrations in the cabin, was developed. By damping the vibrations in the operator seat, the operator is not exposed to in-time harmful injuries and can therefore work for a longer time. With an actively suspended operator seat, new ways open up for a more effective forestry industry as the work becomes more convenient for the operator.

This report presents how the work was performed. The first chapters of the report concern the background for the work, the components and subassemblies of the previous generated design concepts as well as standard components needed in order to realize an active operator seat suspension. Then, the most applicable concept is further developed to a full-scale prototype using CAD-modelling and evaluated using FEM-analysis. Based on the CAD-models, a prototype was manufactured, the vibration control is discussed and conclusions and proposals for future work are presented.

Keywords: Operator, seat, active, suspension, pneumatic

KTH Industriell teknik och management	**Examensarbete MMK 2014:x MKN 111** **Utveckling och realisering av en operatörstol med aktiv dämpning** Ahmed El Shobaki	
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Sammanfattning

I det här examensarbetet har designkoncept för en aktivt dämpad operatörstol för skogmaskiner, framtagna av Doroftei T & Osario O under tidigare examensarbete som ägt rum år 2013, analyserats för att därefter resultera i en prototyp. Med hjälp av dessa analyser har ett förslag utformats i form av en prototyp som minskar påfrestningarna hos maskinoperatören orsakade av vibrationer i hytten. Genom att dämpa vibrationerna i operatörstolen, utsätts inte operatören för skadliga påfrestningar och kan därmed arbeta längre. Med en aktivt dämpad operatörstol öppnas nya möjligheter för en effektivare skogindustri då arbetet blir bekvämare och mindre påfrestande för operatören.

I den här rapporten redogörs hur arbetet gått tillväga. Rapporten inleds med bakgrunden till examensarbetet, uppbyggnaden av de tidigare framtagna designkoncepten samt nödvändiga standardkomponenter för att aktivt kunna dämpa operatörstolsvibrationerna. Därefter presenteras det mest tillämpbara konceptet som i sin tur utvecklas till en fullskalig prototyp med hjälp CAD-modellering och utvärderas med hjälp av FEM-analyser. Baserat på de CAD-modeller som tagits fram, tillverkas en fullskalig prototyp, därefter diskuteras den aktiva regleringen. Rapporten avslutas sedan med slutsatser och förslag på framtida arbete.

Sökord: Operatör, stol, aktiv dämpning, pneumatik

FOREWORD

This page is to thank everyone who gave me support and helped me during my thesis

I would like to thank my supervisor Ulf Sellgren at KTH and contact person Björn Löfgren at Skogforsk, for this amazing opportunity to do this wonderful project towards the forestry industry, and thank them for their support and guidance throughout the project.

Special thanks to Ponsse, especially Juha Inberg and Matti Leinonen for their inspiration and support and for hosting a great study visit at their factory in Vieremä, Finland! Special thanks do also go to Petrus Jönsson at Skogforsk for providing with technical support and measured vibration data.

Great thanks to every member of my family and friends for their endless love, support and patience. You truly are the best!

Ahmed El Shobaki

Stockholm, June 2014

NOMENCLATURE

Notations and abbreviations used in this report are listed below.

Notations

Symbol	***Description***
E	Young´s modulus (Pa)
r	Radius (m)
t	Thickness (m)
a	Acceleration (m/s^2)
m	Mass (kg)
F	Force (N)
P	Pressure (Pa, bar)
X	Longitudinal direction, front and back (x-axis)
Y	Lateral direction, side-to-side (y-axis)
Z	Vertical direction, perpendicular to y-plane, up and down (z-axis)
Roll	Rotation around the x-axis
Pitch	Rotation around the y-axis
Yaw	Rotation around the z-axis

Abbreviations

CAD	Computer Aided Design
FEM	Finite Element Analysis
KTH	Kungliga Tekniska Högskolan (Royal Institute of Technology)
CNC	Computer Numerical Control

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1 INTRODUCTION

This chapter gives an insight of the background for the thesis work, the definition of the problem, the goal and scope as well as delimitations.

1.1 Background

1.1.1 Swedish forestry industry, Skogforsk and Ponsse

The industry of the Swedish forestry plays an important role of the Swedish economy (Svensén, 2012). Within the forestry industry, export, turnover, generation stands for 10-12% of the Swedish economy. The export is the most focused aspect as the raw material is mostly domestic and can reach up to 90% of the mass- and paper production, and up to 75% of the harvested tree products (Svensén, 2012)

In order to develop the forestry industry of Sweden to increase the effectiveness, there is a need of deep research in order to find proper solutions. Skogforsk is a Swedish research institute that provides applicable knowledge and solutions for a more effective and sustainable forestry industry. To achieve this, Skogforsk has several collaboration partners in the Nordic machine manufacturing namely Ponsse OYJ (hereafter called only Ponsse), Komatsu and Deer & Company (more known as John Deer) (Skogforsk, n.d.). With its about 100 employees whereas 65 are researchers, the institute is financed by the industry and the government (Skogforsk, n.d.).

Ponsse is a forest machine manufacturer which is currently one of the leading manufacturers in the world, operating in over 40 countries. Some of these countries are Sweden, Norway, Russia, USA, China, Russia and Brazil. All Ponsse forestry machines are still today manufactured in Vieremä, Finland since the company was founded by Einari Vedgrén in 1970 (Vidgrén, 2013)

1.1.2 Machines used in the forestry industry

The forest machines used in the Swedish forestry industry are divided into three types; harvesters, forwarders and harwarders (Ponsse, 2013)

A harvester is a forestry machine that locates the tree that is going to be cut and then it de-limbs and cuts it down to a desired length. The main work of a harvester is to cut down trees in a specified region depending on their size and value. As the work is performed, the harvester unit leaves the location with the logs in sorted piles.

The forwarder however is forestry machines that locates the piles of logs left by the harvester machine, and load itself with the logs. As it finishes with the loading, it then forwards the load to a checkpoint where the raw material is handled.

A harwarder machine is a hybrid of both a harvester and a forwarder machine, thereof the name. This type of a machine does the work that both of the machines do together; harvesting, loading and forwarding.

1.1.3 The operator

The forest machine is operated by a trained person, sitting inside the cabin. From there, the operator controls all the functions of the machine by operating functioning buttons, levers etc. As the cabin features a lot of buttons and control units, the operator has to make several decisions in as short time as possible to work effectively. According to Löfgren B. saying: "A forestry machine operator harvests a tree in 35 seconds and within that time, the operator makes up to 12 decisions." (Löfgren, 2014)

By this saying, it gives an image of how stressful the environment is for an operator, whereas there is need of a more convenient cabin. This aspect affects the operator mentally. However, there is another aspect which affects the operator physically, vibrations in the cabin that inherits from the driveline, driving and operational motion and movement of the forestry machine. As most of the forest machines do not have a chassis suspension, the operator seat is exposed to vibrations affecting the operator negatively. As the machine operator works in a stressful and vibrating environment, the operator suffers from motion sickness and in time the vibration amplitudes cause injuries in the neck, back and shoulder. Therefore, the machine operator needs a seat that can overcome the vibrations exposed within the cabin.

1.2 Problem definition

The operator seats used in the forest machines are made for trucks, construction machines and tractors. These seats were not originally designed to be used in such rough terrains like forests. This result in that the operator gets shoulder, neck and back pains and at low frequencies the operator can feel motion sick. The problem lies in the existing seat undercarriage which only dampens the vertical vibrations, leaving the other directions of vibrations from being dampened.

Another issue with the existing undercarriage is that it tends to fail or break due to the vibrations in the cabin, bumps on the track as well as shakings which the forest machines are subjected to. The cause of the failure is that the seats are not designed with respect to the high lateral forces that appear as the machine is driven every day through a rough terrain or as in this study, the forest where most of the machines do not have chassis suspension.

1.3 Delimitations

In this project, the following delimitations have been defined:

- Development and realization of an undercarriage with active suspension for an operator seat will only be focused on generated design concept from previous master thesis work by Teodor Doroftei. & Omar Osorio (Doroftei & Osario, 2013).
- The design will only consider the Ponsse cabin design.
- Interfaces of both cabin floor and seat will not be changed.
- Only active suspension solution will be investigated.
- No changes of cabin floor or other cabin components shall be made.
- Static analyses of the undercarriage will be made.
- Realization and prototype manufacturing of an undercarriage featuring an active suspension.
- The prototype will only be tested if there is sufficient amount of time.
- Programming the regulations of the undercarriage will not be made.

1.4 Purpose & Aim

The purpose of this thesis work is to show that by using modern tools such as CAD and FEM, a new solution for a seat undercarriage will be realized. This solution for the undercarriage should withstand the forces and motions in vertical and lateral directions. The solution should also damp the vibrations in the cabin floor by minimum 50%, in order to minimize the negative effects on the operator.

The first task is to examine and develop further the proposed design concepts generated during previous thesis work by Doroftei T. & Osario O. in 2013. Then, the design concepts have to be analyzed using FEM. If the proposed design concepts need further improvements, it has to be made to become optimal in withstanding the forces and dampen the vibrations in all directions.

The second task is to prepare and manufacture a prototype based on the analyzed design concept.

1.5 Method

The method used in this project is called "the Stage-gate model", which is development model and consists of an n-number of stages/phases and gates. After each stage/phase, a gate takes place where decisions are made whether to continue the work done during the phase or to stop. Each gate is a gathering meeting where the process of the project is discussed and important decisions and priorities are made. In figure 1, an example of how the Stage-gate model works is presented.

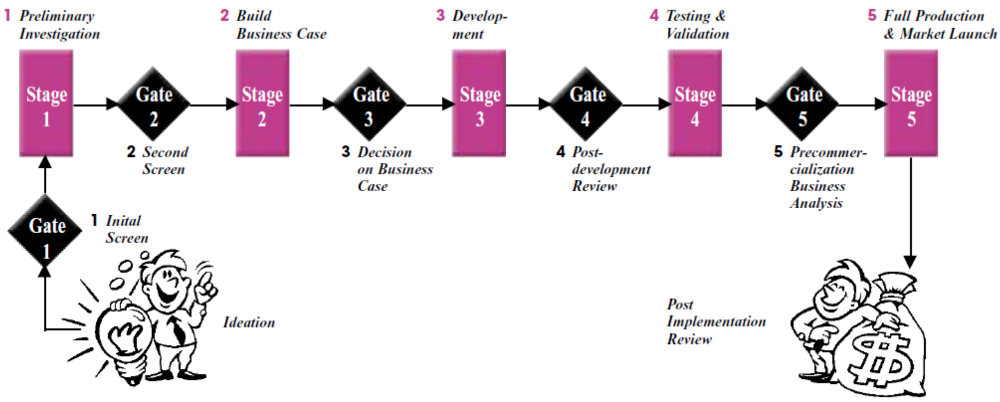

Figure 1: An example of how the Stage-Gate model works (Cooper & Kleinschmidt, 2001).

The method chosen for this project suits it very well since the project reflects the same process. However, since this project does not take testing and validation in consideration, only the first four gates and three stages are held dividing the project into three stages: information gathering, concept development and analysis and finally, the final design and manufacturing of a prototype.

In the first stage, the preliminary investigation, previous thesis reports and are studied to gather as much information as possible about seat suspension and CAD-models for the generated designs are examined. In chapter *2 FRAME OF REFERENCE*, valuable information is presented. The generated design concepts are examined and one of these is chosen for further improvements and realization with help of academic and industrial supervisor.

In the next stage, build business case, the CAD-models for the chosen concept are developed further for a more robust and stiffer system as the system is dimensioned. As the CAD-models get improvements, the model is evaluated using FEM in order to prove that the model is feasible and sound. By this, the model is finalized for the upcoming stage where both academic and industrial supervisor are part of planning the manufacturing process.

The final stage, manufacturing drawings are made and standard components are ordered for a prototype manufacturing. Most of the prototype manufacturing is held at KTH production facility and Da Vinci with help of academic personnel as well as by the author of this report, while some parts may need external manufacturing. Finally, the components are assembled into a full-scale prototype.

2 FRAME OF REFERENCE

This chapter presents gathered information which is relevant for the project. This chapter explains how the system is planned to work and about the major parts which are needed for the system. This chapter does also describe the computer software used to solve the defined problems.

2.1 Previous thesis work

2.1.1 Concept focus and requirements

In order to manufacture a fully working prototype, the first step is to examine previous thesis work. As the previous thesis work suggests two design concepts, seen in figure 2, the focus will in this thesis lye on developing one of these into a prototype. The most applicable design concept for this goal is therefore chosen based on the following requirements founded by Doroftei T. and Osario O. in their thesis work (Doroftei & Osario, 2013), as well as additional requirements by the author of this report:

- The solution should be adaptable to the interfaces of current seats, but mainly the Be-Ge 3100.
- Withstand the lateral strains and shakes that the current undercarriage does not.
- The suspension shall work independent of the height adjustment.
- The height adjustability shall be as much as the original maximum and minimum height.
- 360° swiveling without collision with anything in the cabin.
- The modified seat undercarriage must fit in existing operating cabins.
- Dimensioning according to ISO standards.
- Capable of length adjustments in height, length and around lateral axis for operator comfort.
- Safety, no flammable substances, no sharp edges, fingers should not be able to get squeezed.
- The modified seat undercarriage must be able to be manufactured with standard manufacturing methods.
- The system must not be over-dimensioned to reduce the risk of drawer-effect.
- The system should be designed using as many standard components as possible to reduce the manufacturing cost.

Figure 2: Operator seat design concepts (Doroftei & Osario, 2013).

The undercarriage prototype must fulfill the following:

- Be able to work up to 10 years (roughly 20 000 working hours) before breaking.
- Materials used should be easy to recycle
- Materials should withstand the vibrations in the cabin
- Service and maintenance should be easy
- Should be adaptable for operator heights between 1.6m and 2m
- Should handle operator weights between 50kg and 150 kg

2.1.2 The concepts

The seat suspension module

The seat suspension module consists of two identical suspension modules, the seat base of the existing operator seat as well as a pair of angled dampers. The suspension module features several components but the essential components are the following:

- Upper bracket
- Lower bracket
- Sliding beam with rail guides
- Vertically mounted damper
- Suspension module housing
- Fixing shaft for angled damper (lower end)

The seat base is fastened in the upper brackets of the two suspension modules. These in turn are mounted on the sliding beams which are connected to the dampers. All of these components are packed into the suspension housings. Lastly, the angled dampers are fixed to the suspension modules by shafts.

Since there are vibrations in all three directions (x-y-z), the dampers take up forces in the different directions. If the seat starts to shake due to vertical vibrations, the seat will force the sliding beams to move vertically. The vertical dampers start to compress/stretch as they absorb the forces when the sliding beams are moving. If the seat is exposed to lateral vibrations, this causes the angled dampers compress/stretch. The complete assembly can be seen in figure 3:

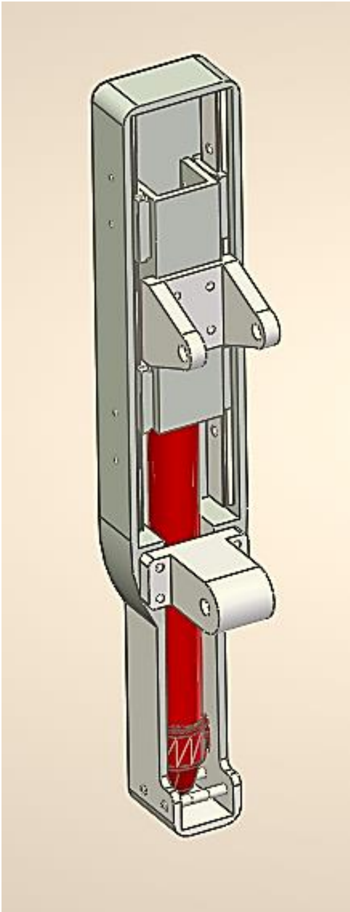

Figure 3: Complete assembly of the suspension module.

Undercarriage concept 1: Stump with seat suspension module

Stump is concept of a stiff undercarriage which adjusts its height linearly to a wanted height. Since the concept consists of several parts, the essential parts are presented as following:

- Pillar base
- Set of four linear rail guides
- Swivel module; Steel plate with integrated brackets and a lateral beam with holes for the seat suspension module.
- Seat suspension module

The pillar base is telescopic which can be extended/ retracted to a wished height. As the pillar is set to a specified height, it keeps its position accurately due to its internal sets of linear rail guides. Currently, the design features four sets of rail guides which are not good due to the risk of over-dimensioning it as well as the risk of the drawer-effect. The solution is better designed if the pillar features two or three sets rail guides by the maximum. The adjustment of the height is designed to be controlled by a strong pneumatic cylinder.

The swivel module consists of a steel plate which is screwed onto a wheel which rotates on top of the pillar. As the operator needs to swivel with the seat inside the cabin for better view, this system enables the swiveling function very smoothly thanks to the low friction bearing which is mounted between the wheel and the pillar base. To lock the swivel module, there is a brake which is controlled by a pneumatic actuator.

In order to attach the suspension module to Stump, there is an integrated lateral beam on the bottom side of the steel plate which features mounting holes for both the lower brackets of the suspension modules as well as for the angled dampers. The lower brackets of the suspension module and the angled dampers are mounted with the help of bolts, washers and nuts. To enable a smooth rotation of the suspension module, there are plain bearings between the bolts and the lower bracket. The complete assembly can be seen in figure 4.

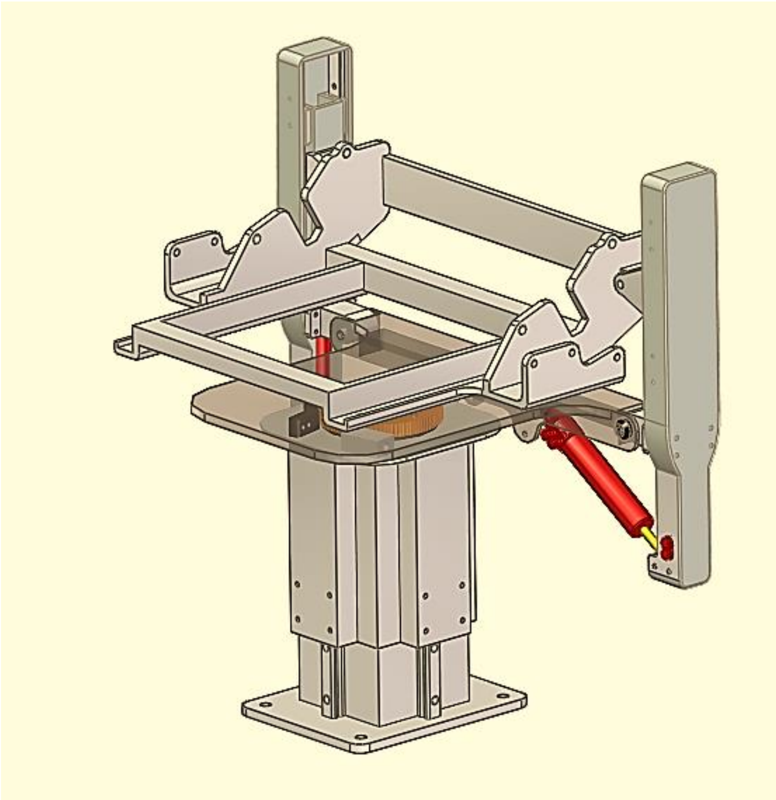

Figure 4: Complete assembly of the Stump-concept.

Undercarriage concept 2: Vesa with seat suspension module

Vesa is a concept which features a smart height adjustment solution which sets the seat more backwards the higher the seat is positioned to. Since taller people need to keep their seat at a higher position, they usually draw their seat more backwards away from the pedals. This concept does both of these adjustments in one motion.

As the concept consists of several parts, the most essential parts are presented as the following:

- Swivel base module from the existing solution from Förarmiljö AB.
- Bottom steel plate with an integrated set of five mounting brackets.
- Set of four scissor arms.
- A powerful actuator which can both adjust the height and keep the seat in a fixed position.
- Top steel plate with five integrated mounting brackets and a lateral beam with holes for the seat suspension module.
- Set of ten fixing shafts with plain bearings.
- Seat suspension module

The bottom steel plate with brackets is mounted on top of the swivel base module using bolts, washers and nuts. To this, the lower ends of the four scissor arms and the lower end of the actuator are fixed inside each bracket with the shafts and bearings. Then, the other ends of the scissor arms and the actuator are fixed in the brackets of the top plate also by using shafts and bearings. Lastly, the seat suspension module can be mounted onto Vesa the same way as presented for Stump. The complete assembly of the Vesa-concept can be seen in figure 5.

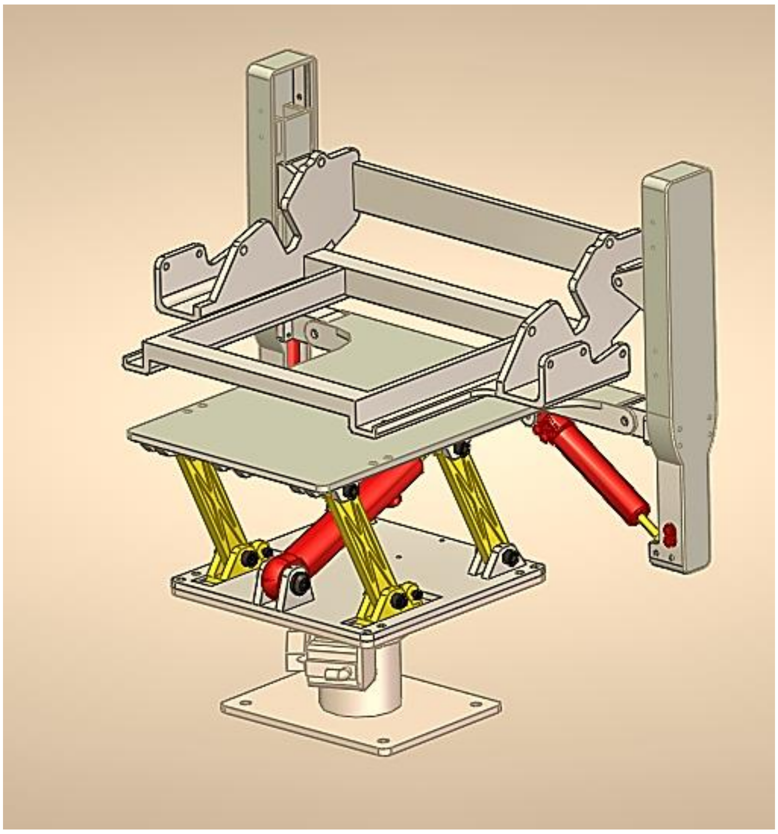

Figure 5: Complete assembly of the Vesa-concept.

Choice of concept to develop further: Vesa with seat suspension module

By examining both of the undercarriage concepts, Stump and Vesa, this raised a couple of questions for each concept to look into and answer. The most interesting of these questions are presented and answered below:

- Q: By the looks, which one of Stump and Vesa seem to be easier to be manufactured?
  A: Stump features a complex design of the pillar base and there is no question about its stiffness since it has wall thickness of 5 mm. But due to its complex shape, the part can only be casted which increases the manufacturing price. Vesa on the other hand features a set of four arms of same size and shape which simplifies the manufacturability and keeps the costs down. To manufacture the scissor arms, they can easily be cut in a water-jet.
- Q: Can any of these concepts design be simplified without changing its function?
  A: Yes, both of the concepts can be developed further into simpler designs without changing each’s functionality. The changes would make the manufacturability easier and cheaper.
- Q: Which of the concepts would be cheaper to be manufactured if simplification of the designs if applied?
  A: Both of the concepts need specific manufacturing processes. To lower the manufacturing costs, some parts need be split into several parts. For example, the top steel plate with integrated lateral beam with mounting holes can be split into two separate parts; steel plate and beam. This does not only lower the manufacturing cost but also it simplifies the manufacturability. However, such simplifications cannot always be applied to parts. For example, the pillar base of Stump can be split into several parts, but this does not simplify the manufacturing process as it still needs to be casted.
- Q: Which of the concepts is more applicable to be manufactured and test its functionality?
  A: By far, Vesa is simpler and cheaper to manufacture than Stump since it features simpler design.

Conclusion: Vesa will be developed further into a fully functional prototype, however the tilting function will not be considered as the main goal is to overcome the vibrations using active controlled components. The design changes and the overall development process will be presented in *3.3 Re-design and development of concept*.

2.2 Study visit at Ponsse

As the GANTT-schedule was planned, a study visit at Ponsses manufacturing facilities was planned and held on March 12th. Matti Leinonen along with Juha Inberg and Kalle Einola at Ponsse helped and arranged a whole day-study visit at their facilities in Vieremä in Finland. The day started out with a project meeting where employees of Ponsse not only got the opportunity to be briefed about this projects plans and goals but also to help with information needed for the project. The designers of Ponsse had very good inputs on what to keep in mind when designing a suspension with active suspension based on pneumatic cylinders.

Later that day, a tour of the factory of the forest machines was conducted. As Ponsse both produce and assemble their components in-house, the whole tour was amazing and informative. For this project though, there was one section of the assembly line which was of highest interest. In this section, several operator seats of different models were waiting to be assembled in responding machine. These seats were examined to get a verification of the size, shape and weight which could be of use further on in the project. Another interesting part of the assembly line was the assembly of the operator seat in the cabin, where one could get a deeper insight on how big the space is between the seat and the control units within the cabin. The space will be high importance when dimensioning the operator seat with dynamic simulations.

Lastly, a test ride of one forwarder unit was held out in the fields of Vieremä. During the test ride of the forwarder which was operated by an expert, one could sit in the cabin of the forest machine and get to feel the working environment of a forwarder operator. As the vibrations in the cabin varies as a function of both the ground as well as the weight of the logs when loading and unloading, this ride gave a good insight of how crucial operation seat suspension is. The test ride was shakier than expected which will be kept in mind when developing the operator seat with active suspension.

2.3 Consultation

In order to make a functional prototype, there is need of some important knowledge both from Ponsse and Skogforsk. Some of this knowledge could for example be measurements of vibrations in the cabin, general thoughts to keep in mind when designing a suspension with active suspension as well as requirements needed for an active damped system. To get this vital information, conversations by e-mail were mainly held with Matti Leinonen and Juha Inberg at Ponsse as well as with Petrus Jönsson at Skogforsk.

Both Matti Leinonen and Juha Inberg provided with information regarding the existing undercarriage as well as information about their forest machines which is needed to keep in mind when generating ideas for the new undercarriage.Petrus Jönsson provided with technical support and vibration data, measured during previous tests on the forwarder cabin. This data stood as a base for the quasi-static analysis of the undercarriage.

2.4 Standard components

After consultation with Ponsse employees as well as with Ulf Sellgren, it became clear that the undercarriage will be active damped using pneumatics. This is rather unusual as active suspension is traditionally applied using hydraulics. As the system will feature active flow controlled pneumatic cylinder, the topic gets more interesting. The reason for the system not to feature active controlled hydraulic cylinders is due to safety regulations for cabin environment. When dealing with hydraulic components, great care has to be taken prevent oil leakage as the oil is usually both flammable and toxic ((ATSDR), 1997). By simply substituting the hydraulic components for pneumatics, health issues regarding toxicity do not need any care. However, as there are upsides with using pneumatic cylinders, there are some downsides as well. These downsides can be stated as follows:

- Pneumatic cylinders are much weaker than hydraulic cylinders due to that air can be compressed which fluids cannot.
- Pneumatic cylinders can give out high-pitched noises if not properly isolated.

In order to manufacture an undercarriage with fully active suspension for an operator seat, there is need to define all the major components for it to function. The following sections describe each required component. The standard components were found from a CAD-model and standard component supplier called Solid Components (Solid, 2014), from which all the necessary standard components were downloaded. The pneumatic parts are supplied by AirTec (AirTec, 2014) and fastening elements come from Wiberger AB (Wiberger, n.d.).

2.4.1 Double acting pneumatic cylinders with magnetic piston and adjustable end cushioning

Double acting pneumatic cylinders can be found in many types of industrial solutions. By the name “double acting”, it refers to its capability to extend and retract by injecting compressed air into one of its inlets. A double acting pneumatic cylinder features two inlets where one is located in the lower part of the cylinder module while the second one is located in the upper part of the cylinder. If compressed air is injected through the lower inlet, the air forces the piston to extend until it reaches its maximum displacement. The air on the other side of the piston is pushed out through the upper inlet as the piston extends (push) to its maximum displacement. If compressed air is injected through the upper inlet, the system would work in the opposite direction, in other words, the piston would retract (pull) to its minimum displacement.

To dimension a pneumatic cylinder to make a specific task by either pushing or pulling, one would dimension the cylinder depending on a force needed be overcome. Therefore, the dimensioning is directly relying on the pressure of the compressed air as well as the piston area on which the air is pushing on. See the following formulas for both pulling and pushing cylinder.

$$F_{push} = p_{air} * A_{piston} = p_{air} * \frac{\pi * d_{piston}^{\ 2}}{4} \tag{2.1}$$

$$F_{pull} = p_{air} * (A_{piston} - A_{rod}) = p_{air} * (\frac{\pi * d_{piston}^{2}}{4} - \frac{\pi * d_{rod}^{2}}{4}) \tag{2.2}$$

The main difference between a pushing and a pulling cylinder is the area which the air is pushing on. If a cylinder is planned to push an object with a known needed push force, as well as the inlet pressure, the piston area can be easily calculated using the formula.

As the undercarriage for an operator seat is planned to be fully active controlled, all the pneumatic cylinders must be of the double acting type. This is to enable levelling of the seat so it is parallel to the ground. In order to find the position of the piston in the cylinders, the cylinders feature a magnetic piston.

2.4.2 Solenoid 5/3-valves

A 3/2-pneumatic valve can be seen as an air director. By the name “5/3-valve”, it basically refers to its five inlet/outlet ports as well as its positions. In the initial state, the valve is at its stand-by position. By switching it to either position and injecting compressed air through the input port, the air is delivered through another port depending on the valves position. For example, if the valve is switched to its left position and the air is injected through the input port, the air is delivered through the output port making a cylinder to extend. The other port of the cylinder is then connected to a port of the valve used as an exhaust port. If the valve is switched to its right position, the port that was used as exhaust port is now being fed with air causing the cylinder to retract. By switching the valve setting which can be done manually, the air is delivered according to its connection scheme. Keep in mind that the inlet port is always used as inlet port no matter the setting of the valve. Also, if the valve position is not switched, it usually returns to its stand-by position.

The setting between second and third port can be controlled electrically rather than manually. By applying electricity to the valve, the valve sets itself to the wanted position. As the undercarriage will adapt itself depending on the magnitude of the cabin vibrations, there is need of electric controlled 5/3 valves. These valves will be controlled by the use of a programmed control box.

2.4.3 Solenoid pressure regulators

Since the pressure of the air will vary in order to overcome the vibrations by making the system stiffer, there is need of pressure regulators. A pressure regulator features one inlet and one outlet port. In between these, there is a spring loaded mechanism which increases or decreases the air pressure in the outlet port. The pressure regulation can either be controlled manually or electrically. Since the components within the undercarriage will be controlled by the use of a control box, the pressure regulators have to be of electro pneumatic type.

2.4.4 Electronic control box

The electronic control box is the brain of the whole active damped undercarriage. It is from here electricity and sensor signals are fed to the responsible such as the electro pneumatic valves and the double acting cylinders. The main inputs to the control box are information about vibration levels and direction, cylinder position, as well as existing pressure of stored compressed air. From this vital information, the control box send signals and electricity feed to the corresponding valve and cylinder pair of the seat suspension module. By this, the corresponding valves are set to deliver compressed air to the pneumatic cylinders making them move in order to stiffen the system to overcome the vibration levels. As they do so, they send signals to the control box which then sends electricity to the pressure regulators which reduces the air pressure in the system.

2.4.5 Accelerometers

In order for the operator suspension to work actively, the system requires an active input of the vibrations found in the cabin. To collect the vibration levels, the system is in need of accelerometers. From these, data in form of accelerations in three directions (x-y-z) is actively collected and sent as an input to the electric control box. Depending on the level of vibrations, the control box would calculate the needed air pressure to the cylinders in order to overcome the vibrations and send electric feed and signals to the corresponding valves. A minimum of four accelerometers should do the job and should be placed in a smart way either the same way as in previous cabin vibration tests or in a better way to achieve as good measurements as possible near the operator seat.

2.4.6 Air reservoirs

The most important components are the air reservoirs which stores compressed air for the seat suspension module. After both consulting and meeting with Matti Leinonen and Kalle Einola at Ponsse, information was retrieved that the current forest machines feature one air reservoir which stores air up to maximum 8.5 bars of pressure. Since the operator seat will be active damped at all times as the forest machine is running, it will need to be fed with compressed air. It is unclear if one reservoir is enough, but however to be on the safe side, it is decided that the seat suspension module would require two air reservoirs to ensure a fully working prototype. When one of the air reservoirs is in use, the other is being refilled by the internal compressor of the forest machine. As soon as it is filled up, the control box will smoothly switch to it, while the other is being refilled.

2.4.7 Rubber dampers

To understand the vibration levels and their magnitude of force in the prototype, the forces in the concept are drawn in a force diagram in order to dimension the cylinders. From the force diagram, the needed cylinder force is calculated. Using the force and the known air pressure in the system, the piston area can be calculated. If the calculated piston diameters turn out to be too big for the prototype, then they would not fit in the prototype and major design changes are needed. To prevent this, the input vibration levels can be reduced in the system by including rubber dampers. The rubber dampers would be placed between the swivel base module of Vesa and the bottom steel plate with brackets. By this, the diameter of the cylinders would become smaller as the vibration levels are damped before the actual suspension of the seat suspension module.

2.4.8 Journal bearings with flange

At all rotating parts of the undercarriage, there is need of bearings. Since the rotational movement of the suspension modules is only in a range of ±10°, there is no need of roller bearings. All bearings will then be of journal bearing type featuring a flange for smooth translation along adjacent surfaces as well as smooth rotation along shafts. The journal bearings are ordered and bought from Igus AB (Igus AB, 2014).

2.5 Computer software

Here are the computer software described in short.

2.5.1 CAD-modelling using Solid Edge ST5

Solid Edge ST5 is a Computer Aided Design-software (CAD) by Siemens PLM ((Siemens, n.d.)) heavily used in this project to examine the previous design work of the undercarriage. Solid Edge ST5 is used for designing 2D/3D-systems and for geometric analyses.

In this project, the previous design work was examined and from it, a re-design of the concept was made in order to manufacture a prototype. As the concept re-design was finalized, the designed parts were imported into ANSYS Workbench 14.5 for further analysis using FEM. Figure 6 present how the work in Solid Edge ST5 looks like in Windows.

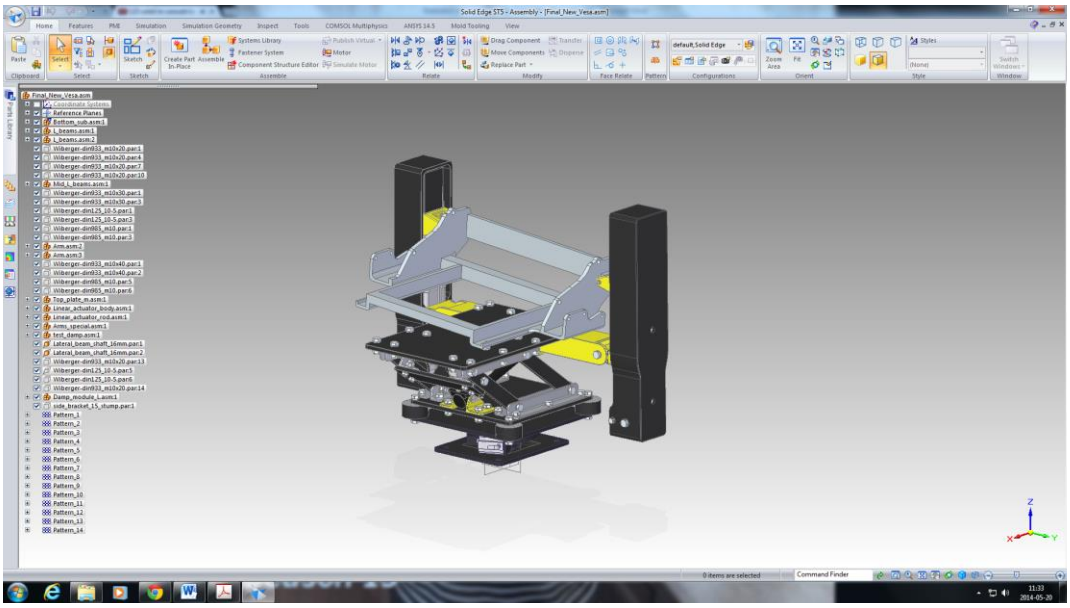

Figure 6: Example of Solid Edge ST5s functionality and features.

2.5.2 FEM-analysis using ANSYS Workbench 14.5

ANSYS Workbench 14.5 ((ANSYS, n.d.)) is an excellent FEM-analysis tool for analyzing systems in order to find deformations and stress levels. The program can beside static analyses and structural mechanics be used for fluid dynamics, multiphysics as well as for electromagnetics.

In this project, ANSYS Workbench 14.5 is lastly used to ensure that all the designed parts are fully compatible with the vibration levels in the cabin. The important factors to ensure are low are the maximum deformations and highest stress levels to ensure that they do not break or yield. If the parts do not pass the FEM-analyses, they are re-designed in Solid Edge ST5 so they get stronger. As soon as they pass the FEM-analysis, the manufacturing process can start resulting in a prototype.

Figure 7 presents an example of a total deformation analysis in ANSYS Workbench 14.5.

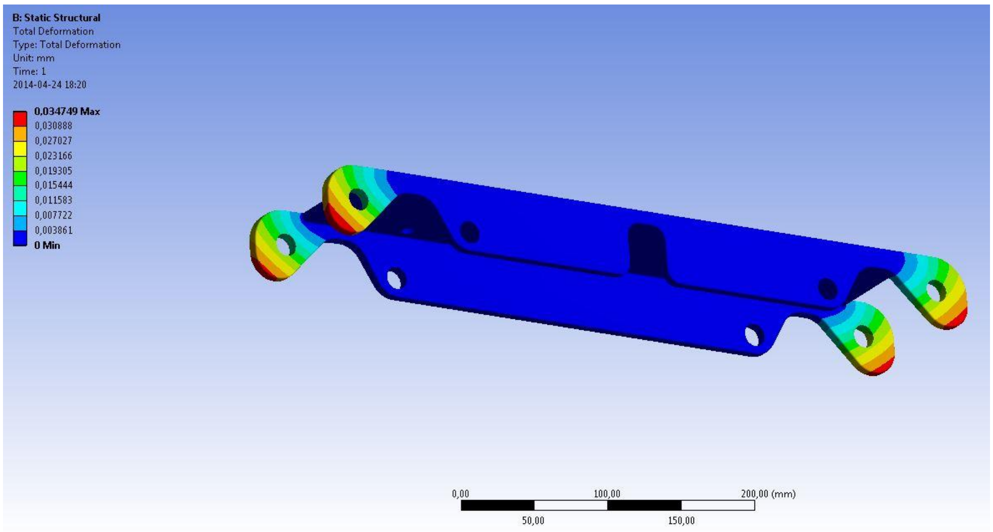

Figure 7: Example of total deformation of a beam in ANSYS Workbench 14.5.

3 THE PROCESS

In this chapter the Vesa-concept is examined and suggestions for re-design of the system are brought up. The pneumatic cylinders are dimensioned and the development of the concept is described. As for the development of the concept, it concerns the overall design changes and the implementing of the pneumatic cylinders. This chapter does also bring up the final design of the undercarriage and FEM-analysis of the ingoing parts of the undercarriage.

3.1 Examination of the Vesa-concept

As previously mentioned in this report, Vesa is the concept of choice for further development. What makes Vesa the choice of concept to work further with is its capability to be re-designed to be manufactured using standard elements. In this section, major parts of Vesa will be described and how these can be re-designed using standard elements without changing its function. All the design changes can be read about in *3.3 Re-design of Vesa*.

To start off, the top plate of Vesa features an integrated lateral beam on which both the suspension modules are attached. In figure 8 and figure 9, it is shown how the top plate has been developed over time for a stiffer and better design. The integrated bracket joints of the top plate are used both for the scissor mechanism as well as for fixing the hydraulic cylinder which controls the height-adjustment. All in all, the top plate features all these fixing joints and can be seen as a stiff system.

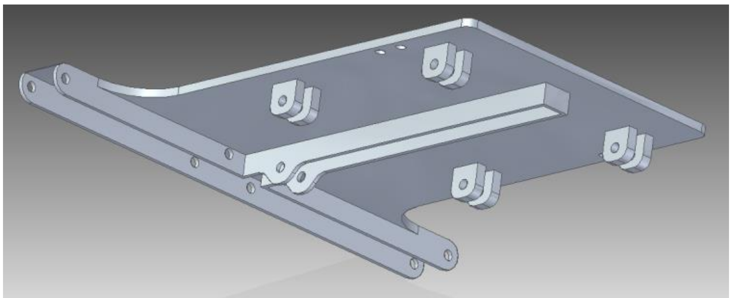

Figure 8: First design of the top plate

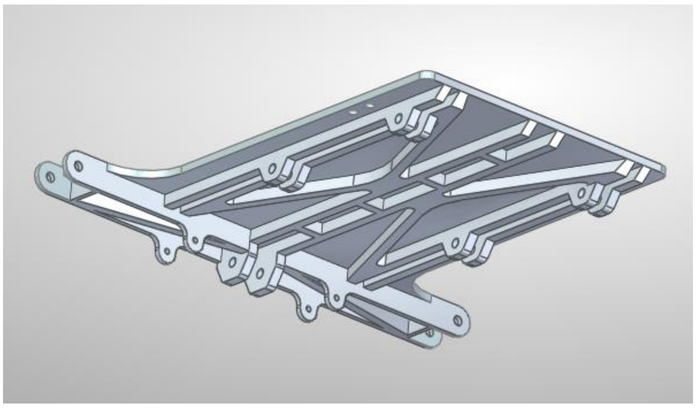

Figure 9: Final design of the top plate

The same goes for the bottom plate which also feature integrated bracket joints for fixing the scissor arms as well as fixing the hydraulic cylinder. As the top plate was developed further over time for a stiffer design, the bottom plate was to. The development process can be seen in figure 10 and figure 11.

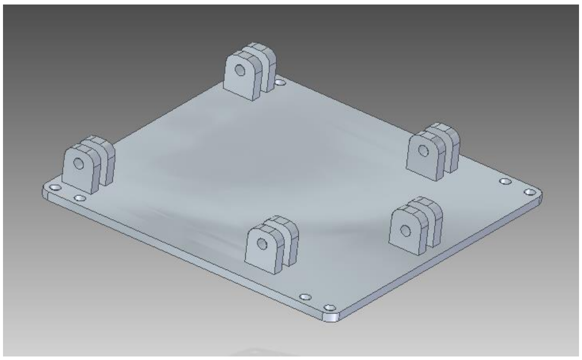

Figure 10: First design of the bottom plate

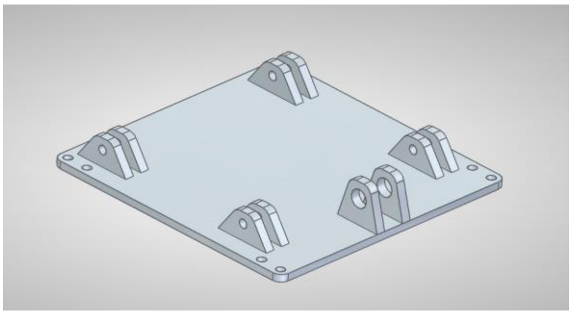

Figure 11: Final design of the bottom plate

Since both the top and bottom plate feature all these important and integrated parts, there are only few manufacturing methods which can be used to manufacture it. The plates can either be casted as whole pieces or milled in a 5-axis CNC-mill. As the solid parts has the possibility to be manufactured using these methods, there are downsides with each. By casting the parts into two units featuring all the integrated bracket joints, the manufacturing cost will be high since there is need to design a new cast form which will only be used for a prototype. Also, the manufacturing cost will be high for the part itself since there is only one part which is being manufactured and not in a series.

The second option would be to mill the parts out of solid blocks of steel, but not only will it take long time to do so, but also there will be a lot of material spill. Therefore, to avoid material spill and to keep the manufacturing cost as low as possible, this calls for a re-design of both the top and bottom plate.

The housings of the damp modules can be seen as blocks which are carved out for the internal components and mechanisms. To manufacture these, they can either be casted or milled same as the top and bottom plate. The cheaper alternative though is to choose the second one since there are no advanced features within each of the housings. The housings can also be manufactured in 3-axis CNC-mills and turned in between to reach all the corners if a 5-axis CNC-mill is not available.

3.2 Free force-diagram and cylinder dimensioning

In order to find suitable pneumatic components which can cope with the vibrations, the forces within the system must be defined. As discussed and agreed with Ponsse, the pneumatic system can feed the undercarriage with compressed air with a maximum pressure of 8.5 bars. By this, the input value of pressure is defined as maximum 8.5 bars.

In order to find input values of cabin vibrations (accelerations), Petrus Jönsson at Skogforsk was contacted. Petrus Jönson provided the project with cabin vibration data conducted during several test runs. As the cabin vibration where measured as a function of time, the results were harmonically distributed. To dimension the cylinders for the undercarriage, the system has to be defined as static or quasi-static. By quasi-static, it means that the system will be dimensioned as static but using dynamic load. By this, the accelerations in the equations below are the maximum measured during the test runs including safety factor to ensure that the cylinders will cope with the same accelerations as well as accelerations with a slightly higher magnitude. Figure 12 shows a simplified free force-diagram of the undercarriage.

$$\begin{aligned} \ddot{x}_G &= 0.44g \\ \ddot{x}_{G.Max} &= g = 9.81\, m/s^2 \end{aligned} \tag{3.1}$$

Where x_G is the lateral acceleration.

$$\begin{aligned} \ddot{y}_G &= 1.6g \\ \ddot{y}_{G.Max} &= 2\,\mathrm{g} = 19.62\, m/s^2 \end{aligned} \tag{3.2}$$

Where y_G is the vertical acceleration.

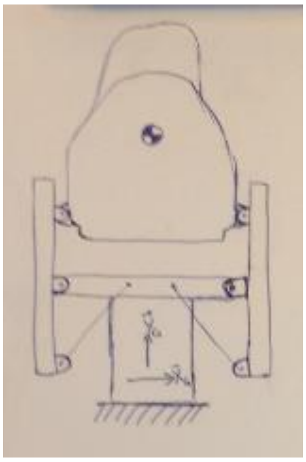

Figure 12: Simplified free force-diagram of the undercarriage.

To start off, the angled cylinders are dimensioned. Below are figures and equations which define the needed cylinder forces to operate the system. The lengths; L_1 and L_2, where measured from the CAD-models of previous thesis work by Doroftei.T and Osario O. The masses of the suspension modules, m_1 and m_2 were assumed to be 20 kg fully assembled, the mass m_3 stands for the weight of the operator seat (≈100 kg) and the operator weight (150 kg) giving 250 kg in total. For the left suspension module seen in figure 13, the equations are the following:

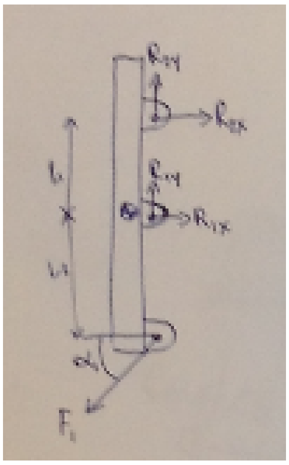

Figure 13: Free force-diagram of the left suspension module.

$$\begin{aligned} &\rightarrow: \mathrm{R}_{2X} + R_{1X} - F_1 \cos(\alpha_1) = m_1 \ddot{x}_G \\ &R_{2X} + R_{1X} - F_1 \cos(45^o) = 20g \end{aligned} \tag{3.3}$$

$$\begin{aligned} &\uparrow: R_{2Y} + R_{1Y} - F_1 \sin(\alpha_1) = m_1 \ddot{y}_G \\ &R_{2Y} + R_{1Y} - F_1 \sin(45^o) = 0 \end{aligned} \tag{3.4}$$

$$\begin{aligned} &A: R_{2X} L_1 - F_1 \cos(\alpha_1) \mathrm{L}_2 = J_x \ddot{\gamma} \\ &R_{2X}\, 0.210 - F_1 \cos(45^o) 0.147 = 0.44 \ddot{\gamma} \\ &Where: \ddot{\gamma} = \frac{\ddot{x}_G}{L_1} \end{aligned} \tag{3.5}$$

For the right suspension module seen in figure 14, the equations are the following:

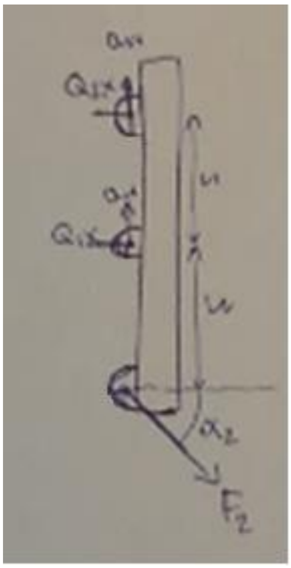

Figure 14: Free force-diagram of the right suspension module.

$$\begin{aligned} &\leftarrow: Q_{2X} + Q_{1X} - F_2\cos(\alpha_2) = m_2\ddot{x}_G \\ &Q_{2X} + Q_{1X} - F_2\cos(45) = 20\,\mathrm{g} \end{aligned} \tag{3.6}$$

$$\begin{aligned} &\uparrow: Q_{2Y} + Q_{1Y} - F\sin(\alpha_2) = \mathrm{m}_2\,\ddot{y}_G \\ &Q_{2Y} + Q_{1Y} - F\sin(45) = 0 \end{aligned} \tag{3.7}$$

$$\begin{aligned} &\mathrm{B}: Q_{2X}L_1 - F_2\cos(\alpha_2)\mathrm{L}_2 = J_x\ddot{\gamma} \\ &R_{2X}\,0.210 - F_2\cos(45^o)0.147 = 0.44\ddot{\gamma} \\ &Where: \ddot{\gamma} = \frac{\ddot{x}_G}{L_1} \end{aligned} \tag{3.8}$$

For the seat seen in figure 15, the equations are the following:

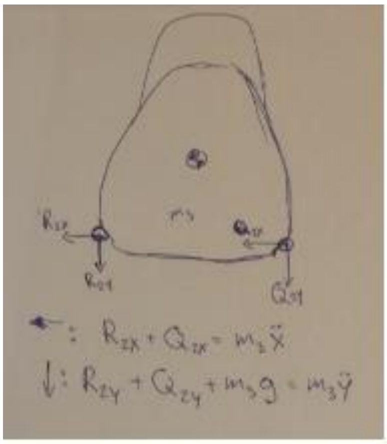

Figure 15: Free force-diagram of the operator seat.

$$\begin{aligned} &\leftarrow: \mathrm{R}_{2X} + Q_{2X} = m_3\ddot{x} \\ &R_{2X} + Q_{2x} = 250g \end{aligned} \tag{3.9}$$

$$\begin{aligned} &\downarrow: \mathrm{R}_{2Y} + Q_{2Y} + m_3 g = m_3\ddot{y} \\ &\mathrm{R}_{2Y} + Q_{2Y} + 250g = 0 \end{aligned} \tag{3.10}$$

$$R_{2Y} = Q_{2Y} \tag{3.11}$$

Substituting gives the force $F_1 \approx 3.1$kN

From this, the cylinder diameter can be calculated as:

$$d = \sqrt{\frac{4F_1}{\pi p}} = \sqrt{\frac{4*3100}{\pi*8.5*10^5}} = 0.0682m \tag{3.12}$$

The diameter is calculated in way that one of the angled cylinder pair takes up all the forces. Since there is a pair of cylinders, one of the cylinder pushes back while the other cylinder pulls the operator seat back to its neutral position.

The closest standard cylinder piston diameter from AirTec is 0.063m (AirTec, 2014). By reducing the pressure of the compressed air to feed the cylinders, the needed diameter increases. However, since the input accelerations for the equations were adjusted for the cylinders to cope with higher forces.

Having a piston diameter of 0.063m and an input pressure of 8.5 bars, the maximum force is calculated to be:

$$F = pA = 8.5*10^5*\left(\frac{\pi 0.063^2}{4}\right) = 2.65kN \tag{3.13}$$

Using the maximum measured acceleration from the test runs ($\ddot{x}_G = 0.44g$, $\ddot{y}_G = 1.6g$) as input for the equations, the maximum force is calculated to be only 1.35kN. This proves that the cylinder can cope with the accelerations from the test runs as well as higher accelerations.

Next is to dimension the vertically mounted cylinders. Below are figures and equations which define the needed cylinder force to cope with the dynamic loads. From the results of the equations, closest standard piston diameter is found and analyzed.

$$\begin{aligned}
&\uparrow: \mathrm{R}_{2Y} - m_3\ddot{y}_G + Q_{2Y} = R_{2Y} - (250*2*9.81) + Q_{2Y} = 0\\
&R_{2Y} + Q_{2Y} = 250*2*9.81\\
&R_{2Y} + Q_{2Y} = 4905N\\
&R_{2Y} = Q_{2Y}\\
&R_{2Y} = 2452.5N\\
&Q_{2Y} = 2452.5N
\end{aligned} \tag{3.14}$$

The diameter of the cylinders can be calculated as:

$$d = \sqrt{\frac{4R_{2Y}}{\pi p}} = \sqrt{\frac{4*2452.5}{\pi*8.5*10^5}} = 0.06062m \tag{3.15}$$

Closest standard cylinder diameter from AirTec is found to be 0.063mm (AirTec, 2014) which gives a maximum force of:

$$R_{2Y} = pA = 8.5*10^5*\left(\frac{\pi 0.063^2}{4}\right) = 2.65kN \tag{3.16}$$

Since the acceleration would not necessarily be as high as in the calculations, it is known that the cylinders would cope with the forces up to the calculated forces.

Now that the cylinder diameter is found, the next step is to find the suitable stroke length for each cylinder. This is primarily done by examining both the previous CAD-model as well as checking the standard cylinder lengths from AirTec. For the angled cylinder pair, the important parameters are the angles of the cylinders α_1 and α_2 as well as the lengths from the rotational joint to the fastening joints L_2. These values should not be changed in order to fulfill the force equilibrium.

For the vertically mounted cylinders, the important parameter is the maximum length between the rotational joint and the upper bracket, $L_{1,}$ has to be same as been used in the force-study.

By investigating through Solid Components (Solid, 2014), different models of AirTec showed up. From here, CAD-models and drawings be generated after ones requests and can be downloaded. By searching for the proper cylinder model, the best models to work with were found to be the XL-series (AirTec, 2014) and the NXD-series (AirTec, 2014) which can handle up to 10 bars of pressure. Both cylinder series can be seen in figure 16.

Since there is not much space to work with between the fixing joint of the suspension modules and the fixing joint of the lateral beams, the NXD-series were found to be optimal to be used as angled cylinders. This was decided to be optimal due to their compactness.

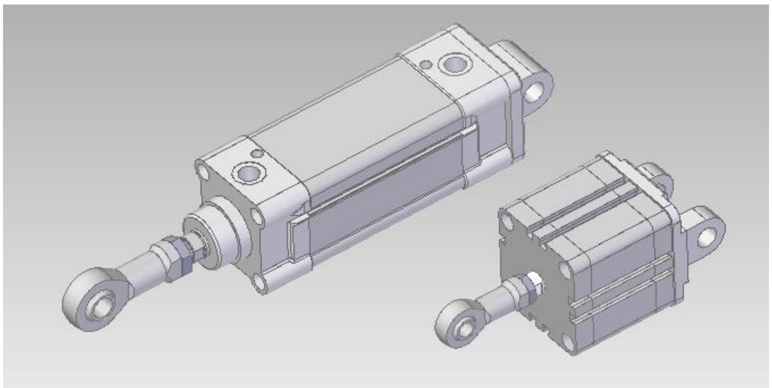

Figure 16: Left cylinder, XL-063-80 with rod eye and swivel base. Right cylinder, NXD-063-30 with rod eye and double mounting ears.

The XL-series cylinders where decided to be used as vertical dampers because of their long shape and flatness. From here, the suspension module housing can be re-designed after the cylinder for a stiff solution.

In order to find the needed stroke length of the angled cylinders, the suspension modules were tilted ± 10^{o} and trace curves where drawn of the fixing joints for the cylinders. By this, the line between the outermost trace point and the innermost was measured and gave an indication of the needed stroke length. This turned out to be close to 30mm which is close to standard stroke length. From here, the angled cylinders were decided to be NXD-063-30.

To find the stroke length of the vertically mounted cylinders within the suspension modules, a CAD-model of an XL-cylinder with a diameter of 63mm and a stroke of 80mm was downloaded. By importing the CAD-model into Solid Edge ST5, the XL-cylinder was placed next to a suspension module seen in figure 17, in order to see if the stroke length was sufficient.

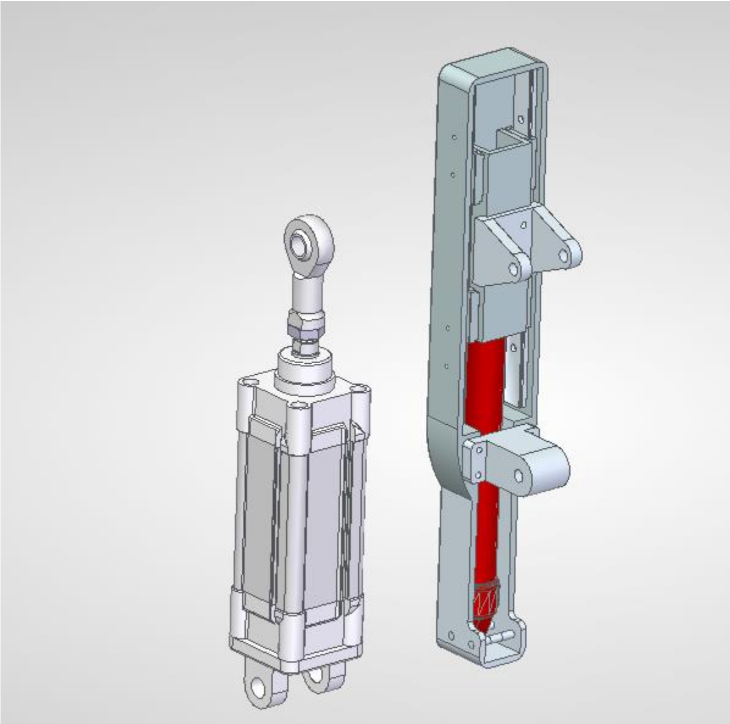

Figure 17: XL-063-80 pneumatic cylinder next to the old suspension module.

This turned out to be good, however the suspension module housing needed to be re-designed for the cylinder to fit within it and to ensure that the length L_1 is not changed. The vertically mounted cylinders were decided from here to be XL-063-80 and the re-design process can be introduced.

3.3 Re-design of the Vesa-concept for prototype manufacturing

In each of the following sections, the re-design process of each part and sub-assembly is described in detail with words and images. The last section, *3.3.9 Manufacturing methods* bring up how each part will be manufactured.

3.3.1 Bottom plate with separated bracket joints

The re-design process started basically with the bottom plate where the focus was laid on separating the integrated brackets from the actual plate. To achieve this and to keep the functionality of the brackets, the idea was to design a thick steel plate and to replace the brackets with L-profiled angle steel which is fastened onto the plate. First, the base protrusion of the bottom plate was measured in order to design a new bottom plate. From here, the new bottom plate started to get shaped with its base protrusion. An important part of the bottom plate is the holes found in the corners which are needed to keep when re-designing the plate since they are mounting holes for the swivel base-160 mm that the existing operator seat uses from Förarmiljö AB (Förarmiljö, n.d.). Therefore, the holes pattern is kept and applied to the new bottom plate.

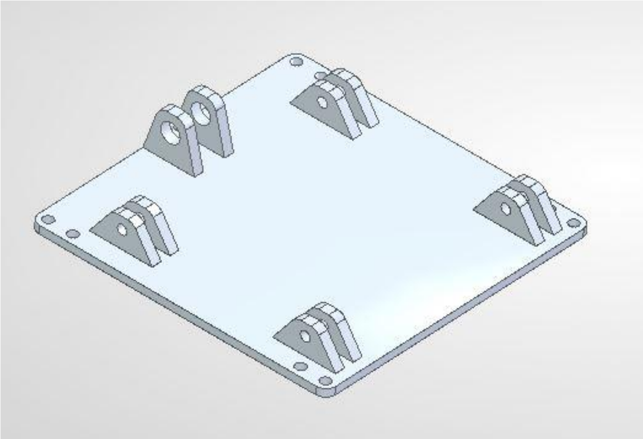

Figure 18: Old design of the bottom plate with integrated bracket joints

The next step was to find the angle steel to replace the internal brackets, seen in figure 18, and in order to do so, measurements of the bracket joints were taken in form of positions and dimensions of the holes. As the measurements were made, steel angles were investigated where the minimum size of the profile to keep the holes positions was found to be 30x30mm with a wall thickness of 5mm.

The last part of the re-design was to find the length which the steel angles were to be cut into. Since the scissor arms follow the same direction of motion when the seat is adjusted to the proper height, the idea was to integrate two bracket joints in one steel angle. Therefore, the distance between the holes were measured and marked out in the steel angle part seen in figure 19.

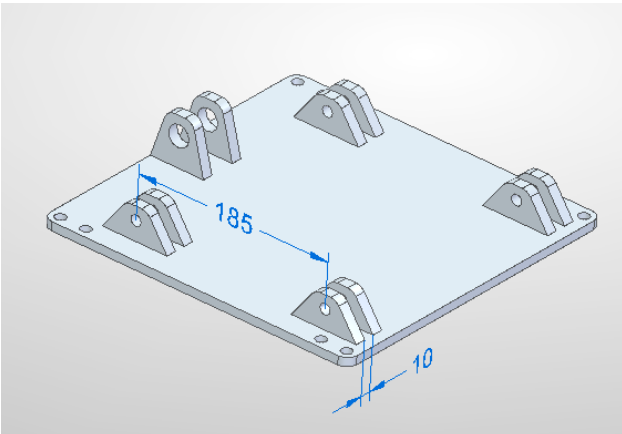

Figure 19: Measurements taken for the new steel angle bracket joints.

As the holes were drilled out, the steel angles were mounted onto the bottom plate in order to verify the positions of the holes for the scissor arms. From here, the design was finalized with drilling out holes for the steel angles to be screwed onto the bottom plate. For a stiff screw joint, the screws were decided to be standard M8 machine screws with hexagon head. The results of the design changes can be seen in figure 20 where both the old and the new bottom plate are presented.

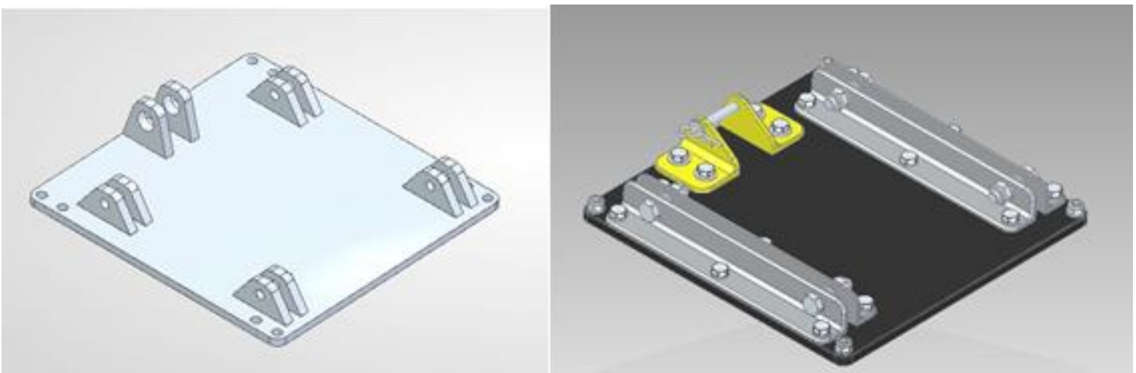

Figure 20: Left, old bottom plate. Right, new bottom plate

3.3.2 Top plate with separated bracket joints and lateral beam

As the top plate was a little more complex than the bottom plate since it featured both integrated bracket joints as well as a lateral beam, the re-design process was divided into three stages:

- Designing a 10 mm thick steel plate
- Find a standard steel UNP-beam and replace the lateral beam
- Cut and assemble steel angles as bracket joints

First, the old top plate seen in figure 21 was examined to find measurement of the base protrusion. This was to design a new steel plate on which the UNP-beam and the steel angles were to be mounted.

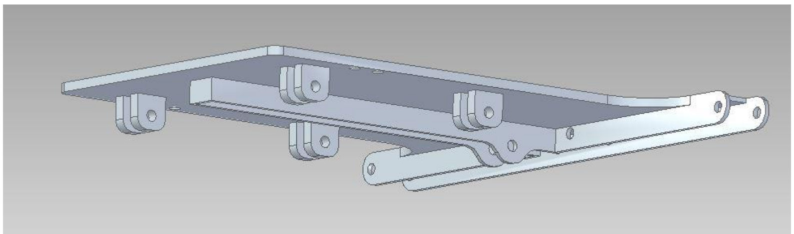

Figure 21: Old design of the top plate.

As the new steel plate started to get shaped, the lateral beam was examined in order to find a suitable UNP-beam to replace it with. The choice of using a standard UNP-beam was not only to fulfill the function of the lateral beam but also to ensure that the system would be stiff using standard elements. In order to find a UNP-beam of suitable profile size, the following requirements of it had to be made and fulfilled:

- Maintain the function of the lateral beam
- Be able to be assembled onto the top plate without interfering with the scissor arms or steel angles
- Be able to fit the angled cylinders within its gap and to keep them fixed
- Design of the UNP-beam must consider the movements of the angled cylinders.

To know the minimum gap size of the UNP-beam, first the angled cylinders had to be examined to see the size of the mounting ears (AirTec, 2014). From this value, the profile of the UNP-beam was decided to be 100x50mm with wall thickness d=6.0mm and t=8.5mm (SBI, n.d.) as it featured a gap which would work with the cylinders. Now that the profile of the UNP-beam was decided, it was time to cut it into the same length as the lateral beam to maintain its function. By copying the distance between mounting holes for the suspension modules as well as positioning holes for fixing shaft for the angled cylinders, the UNP-beam was cut and shaped to the correct size seen in figure 22.

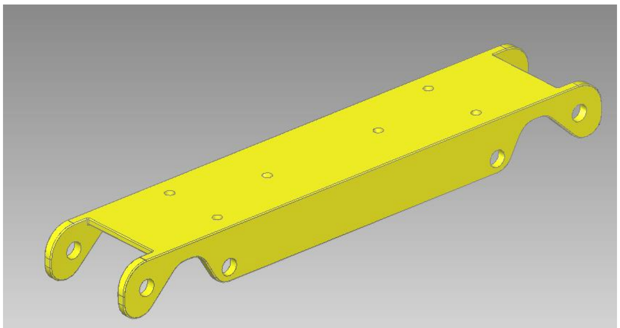

Figure 22: New design of the UNP-beam with mounting holes

As the UNP-beam started to get its final shape, a pattern of 6 holes were marked and drilled in it as well as in the steel plate with a diameter of 9mm to fit 6 standard M8-bolts with hexagon heads. Lastly, the UNP-beam was mounted in place onto the steel plate.

The last step of the re-design of the top plate was to design the separated bracket joints. This was simply made by test-assembling the already designed steel angles for the bottom plate and as it turned out, the steel angles fit perfectly in the top plate without interfering with the UNP-beam seen in figure 23 below.

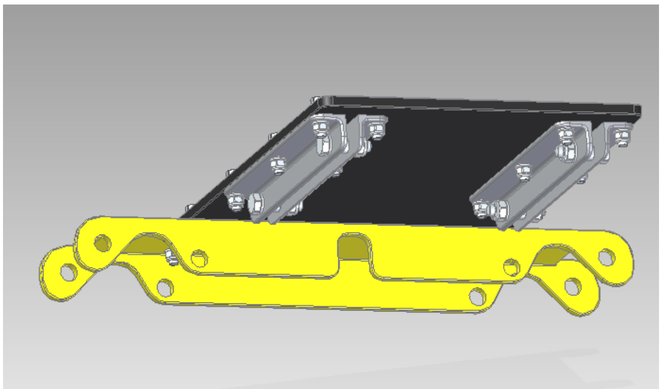

Figure 23: Top plate with mounted steel angle bracket joints and UNP-beam.

3.3.3 New scissor arms, bearings and bushings

For smooth rotation of the scissor arms to adjust the height of the seat, each arm was designed to feature journal bearings with flange from Igus AB (Igus AB, 2014). These bearings were decided not only to enable smooth rotation around each's fixing shaft, but also to maintain smooth translation along the surfaces of the steel angles. These arms however had to be resized for the linear actuator to fit in-between the top and bottom plate without interfering with any. To solve this problem, the scissor arms had to be longer than the previous ones. Due to this change, the minimum and maximum height of the operator seat changed as well. However, after consulting about this issue with Matti Leinonen, a CAD-file of a new swivel base module was received by e-mail, only this was 80mm high instead of the old one which was 160mm. With the new swivel base module, the issue with the height was not only solved but it also enabled the new design concept to feature a set of four rubber dampers.

As the new scissor arms had to be longer than the old ones, suspicion thoughts about their stiffness were brought up. Therefore, a reinforcement plate was designed to be mounted in the front pair of the scissor arms, fixed in a pair of steel angles seen in figure 24. By this, the issue of the systems stiffness was solved as the reinforcement plate increases the prototypes lateral stiffness.

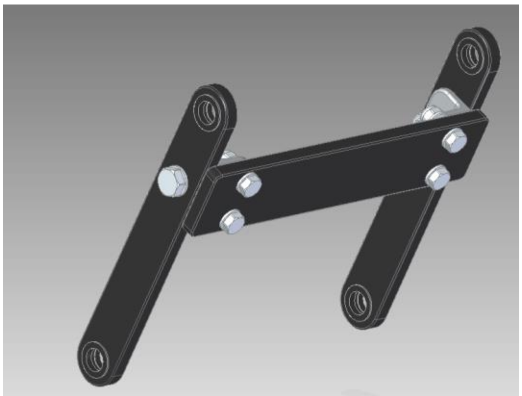

Figure 24: Front scissor arms featuring reinforcement plate.

As the arms got their final shape, the fixing shafts were designed to be threaded bushings which were fixed in-between the steel angle brackets. After searching for journal bearings for the scissor arms at Igus AB, a proper size of the bearings were found to be 15x17x4 mm ($Ød_i$ $*Ød_o*w$) seen mounted in figure 24 as well as in figure 25 below. By this, the bushings were designed to be of Ø14.9mm and 12mm wide to feature an M10 internal thread for fixing the arms in-between the steel angles. In left of figure 25, the bushing design is presented and in the right when mounted inside the scissor arm with journal bearings.

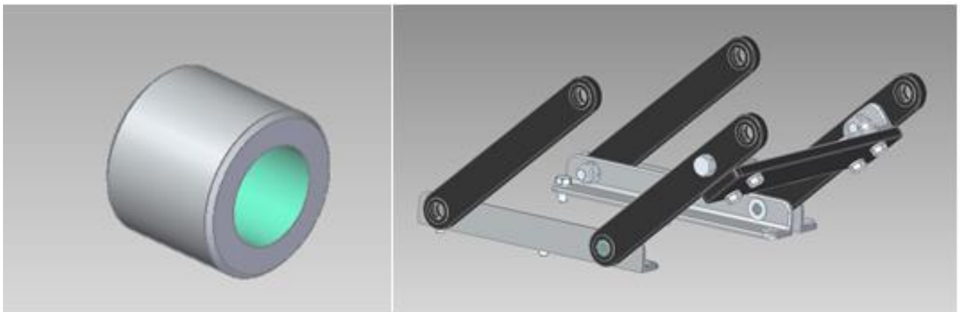

Figure 25: Left, M10-threaded bushing for scissor arms. Right, bushing mounted inside front scissor arms.

3.3.4 New height adjustment solution and fixing elements: Linear actuator

During the study-visit at Ponsse, it became clear that all hydraulic components within the undercarriage would be switched to pneumatic components and also to separate the suspension solution from the height-adjustment. This is for the undercarriage to dampen the operator seat no matter of what position of the height the seat is in. However, by simply replacing the hydraulic cylinder for a pneumatic cylinder could cause the height position if the pressure is not enough. Another problem that would occur is the diameter of the piston within the pneumatic cylinder which needs to be big in order to cope with the forces that the undercarriage would withstand and thus causing the pneumatic cylinder not to fit between the scissor arms. The other problem is that pneumatic cylinders do have an ability to work as dampers which is not wished since the seat should be locked into a specific position.

With all these problems stated using a pneumatic cylinder for the height-adjustment, a new solution for the height-adjustment had to be found. Using components from LEGO TECHNIC-series (LEGO, n.d.), a quick mockup of the undercarriage was built presenting the top and bottom plate, the scissor arms, rubber dampers as well as a linear actuator to control the height position seen in figure 26.

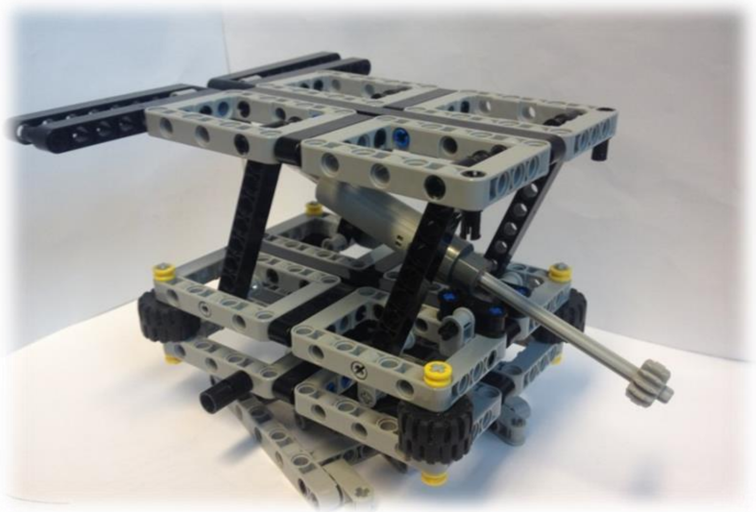

Figure 26: Mockup of the height-adjustable scissor arms using a linear actuator, completely built using components from LEGO TECHNIC.

From this inspiration, the new solution had to be an electric driven linear actuator. By searching through Solid Components (Solid, 2014) for a linear actuator which could cope with the forces within the undercarriage, a linear actuator of RE-35-series was found from Reac AB (Reac, 2014). This type of actuator is powered by either 12 or 24VDC and can cope with forces up to 6 kN. A CAD-model of the actuator was downloaded to verify its size in comparison with the top and bottom plate. However for it to be used in the undercarriage, the scissor arms had to be modified for it to fit properly without interfering with the top plate. This process is presented in *3.3.3 New scissor arms with journal bearings*.

To fix the lower end of the actuator in the bottom plate, bracket joints were designed using a standard steel angle of profile size 45x45mm with a wall thickness of 5mm. As the actuator was fixed in place and moved around the axis of its mounting holes in the steel angles, mounting holes in both the bottom plate and the bottom part of the steel angles were drilled. Then, the steel angles were bolted in place on the bottom plate.

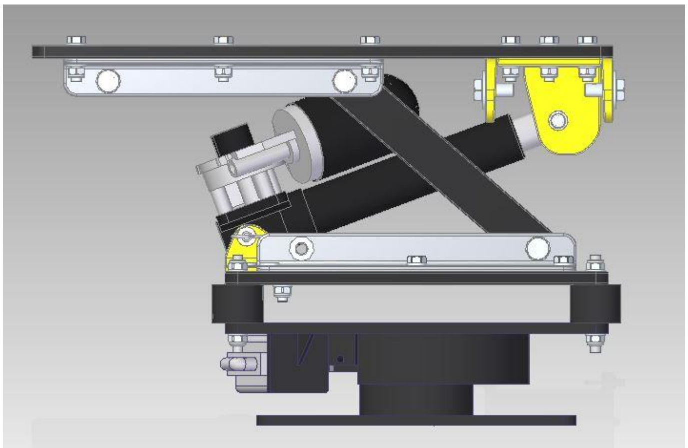

Figure 27: Positioning of the linear actuator to find points for mounting.

To find the position of the mounting holes for the other end of the actuator, the distance between the floor and the top plate of the previous version of Vesa was measured when the operator seat was in its lowest position. By simply placing the top plate at the same distance from the floor as with the previous version, the linear actuator was rotated until it could be mounted in the UNP-beam of the top plate. From here, two mounting plates of 10mm thick steel plate were designed to be welded into the gap of the UNP-beam in order to fix the actuator end, seen in figure 27. To fasten the linear actuator in the bracket joints and mounting plates, bushings and HCB-bolts were used secured with scissor pins, all from Wiberger AB (Wiberger, n.d.). The result can be seen in figure 28.

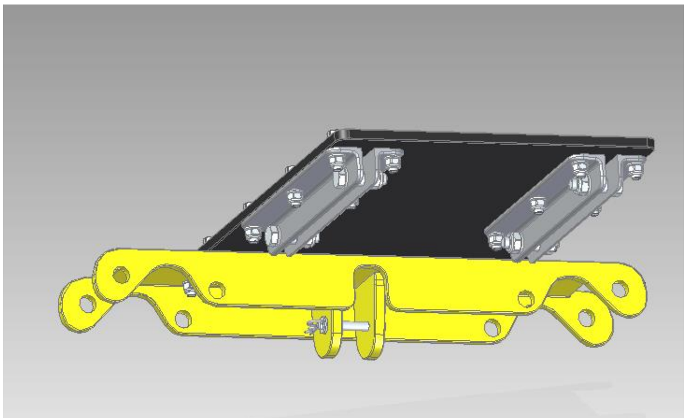

Figure 28: Top plate with mounted steel angle bracket joints and UNP-beam with mounting plates.

3.3.5 Rubber dampers

For the suspension to work properly without air pressure drop to overcome the vibrational forces, rubber dampers from Wiberger AB (Wiberger, n.d.) were introduced to reduce the vibration levels before the actual suspension system. Using the hole pattern of both the swivel base and the bottom plate (described in *3.3.1 Bottom plate with separated bracket joints*.) as reference, a set of four rubber dampers were found featuring internal M8-threaded shafts. These rubber dampers have dimensions 40x30mm (Ød*h) and can handle up to 1.5kN each according to Bo Glembring at Wiberger AB. Since the forces within the operator seat will be distributed over the rubber dampers, they seemed to be fully working.

A CAD-file of a rubber damper of this size was downloaded and mounted in-between the swivel base and the bottom plate, secured with both washers and nuts seen in figure 29.

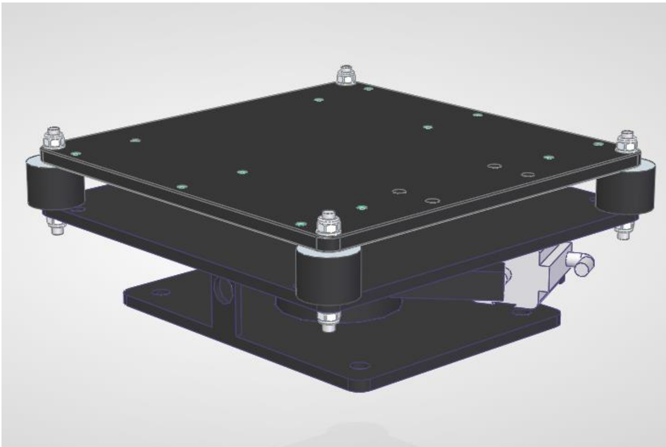

Figure 29: Rubber dampers fixed in-between the bottom plate and the swivel base-80mm.

3.3.6 Re-design of the suspension module housings for the pneumatic cylinders

By examining the old suspension module, important factors had to be taken in consideration when re-designing the new housing. As the housing of the suspension module has to fit in all the necessary components such as the pneumatic cylinder, linear rail guide as well as a sliding bracket, the following important factors had to be considered:

- The housing and mounting holes for the cylinders must be designed to fulfill the lengths L_1 and L_2 described in *3.2 Free force-diagram and cylinder dimensioning.*
- The housing must be designed after the standard components
- The housing must be able to fix the lower bracket joint at the same position using the lengths L_1 and L_2.

By knowing that the largest component within the suspension module is the pneumatic cylinder XL-063-80, measurements of the cylinders length and width was taken of it in fully extended state. From this value, a larger model of the old housing was quickly extruded in CAD featuring mounting holes for the cylinders lower mounting holes. As the housing was designed to fit the cylinder, the next step was to make mounting holes for the lower bracket joint to fulfill the length L_2. The process of the re-design worked side-by-side with section

3.3.7 Re-design of the sliding bracket and upper and lower bracket joints. The result of the suspension moduel housing can be seen in figure 30.

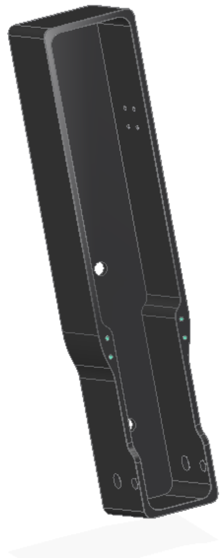

Figure 30: Final design of the housing.

3.3.7 Re-design of the sliding bracket and upper and lower bracket joints

As the base shape of the suspension housing was designed, the lower bracket joint was re-designed for it to be mounted on the housing and fixed in the UNP-beam of the top plate. Since the UNP-beam was wider than the integrated beam of the old top plate, the lower bracket joints had to be adjusted. To achieve this, the base extrusion of the lower bracket joint was first adjusted to fit inside the UNP-beam and later designed to be mounted on the housing using 4 M6-machine screws. Working side-by-side with the lower bracket joint and the suspension housing, the parts started to reach their final design. In figure 31, the old vs the new lower bracket joint are presented. The reason why the mounting holes for the UNP-beam had to be bigger that the old ones was for the bracket joints to have built in journal bearings, more described in the next section *3.3.8 New fastening elements and more bearings*.

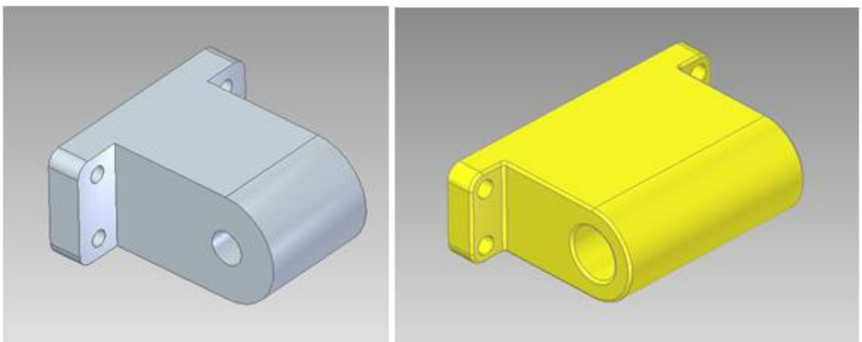

Figure 31: Left, old design of the lower bracket joint. Right, new design of the lower bracket joint.

When designing the upper bracket joint, the idea was first to design a sliding bracket for it to be mounted on. The idea behind the design of the sliding bracket was to use standard UNP-beam of a proper size to fit inside the suspension housing. By examining the design of the housing, a UNP-beam of 65x42mm was found to be perfect to use. The sliding bracket was simply drawn using the profile of the UNP-beam and was drawn to be 145 mm long and to fulfill the length L_1 when the pneumatic cylinder was in fully extended state, a mounting hole was for it was drilled. Meanwhile, the old upper bracket joint was re-designed to fit on the sliding bracket and be fixed with 4 M5-machine screws. In figure 32, the old vs the new upper bracket joint are presented.

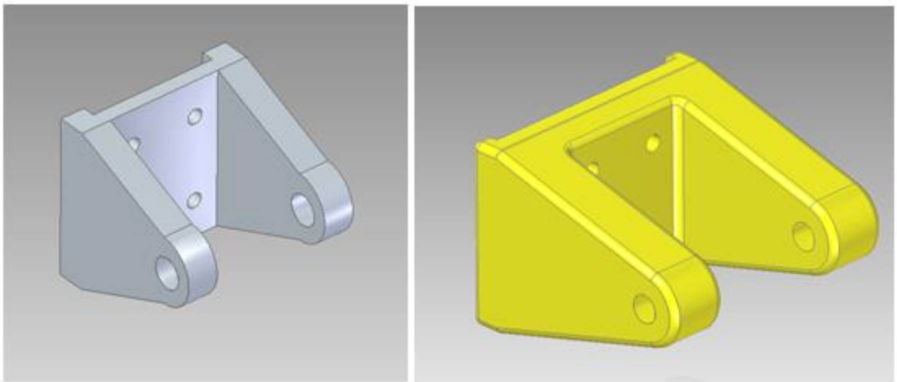

Figure 32: Left, old upper bracket joint. Right, new bracket joint.

As the sliding bracket and the bracket joints were reaching their final design, the next step was to find a suitable linear rail guide for the sliding bracket to ensure stable sliding. This was done by searching at Schneeberger (Schneeberger, n.d.). This resulted in a linear rail guide of model MNNL-12-145 featuring a 145mm long rail and a guide block of a long size. As the rail guide assembly was imported to the assembly model of the suspension module, there were some fine tuning of the sliding bracket and the suspension housing to mount the rail guide.

3.3.8 New fastening elements and more bearings

Now all that was left to design was fastening element to fix all the moving parts in place and only allow rotation. This was solved by designing shafts which were designed to be fixed using machine screws not only increase the prototype's stiffness but also to ease the maintenance of the final product. The first set of shafts that were designed was the shafts to fix the suspension modules and the angled pneumatic cylinders. Since the forces will most likely be the highest at the rotational points of the suspension modules, it was decided to design four shafts of Ø16mm and almost as long as the width of the UNP-beam. On the sides of the shaft, there are internal M10-threaded holes for simple fixing using machine screws.

On the shafts which are used to support the suspension modules, the bracket joints need journal bearings for smooth rotation along the shafts. By searching for proper sized journal bearings at Igus AB (Igus AB, 2014), the bracket joints were modified for the bearings to fit inside their mounting holes as the bearings were found to be 16x18x17($Ød_i$ *$Ød_o$*w). To fasten the shafts, they are first inserted through the holes in the UNP-beam, then through the journal bearings and lastly through the other hole of the beam. Then, M10-machine screws and washers fix the shaft in place on both sides seen in figure 33.

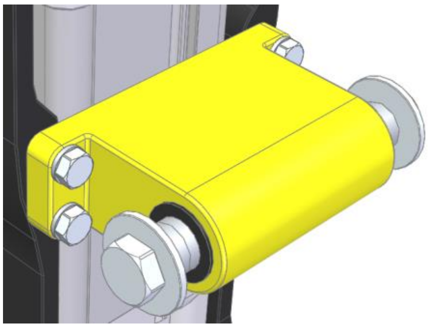

Figure 33: Shaft inserted through the lower bracket joint and fixed to support the suspension module in the UNP-beam.

The next set of shafts that were designed, were the shafts for fixing the pneumatic cylinders within the suspension housing. These shaft were designed in the same way as the ones used for the UNP-beam, however they were slightly smaller due to the diameter of the rod eyes of the angled cylinders. The shafts for fixing rod eye of the angled cylinders had a diameter of 12mm and featured M8-threaded holes, while the other shafts for fixing the double ear mounts of the vertical cylinders had a diameter of 16mm and M10-threaded holes, seen in figure 34.

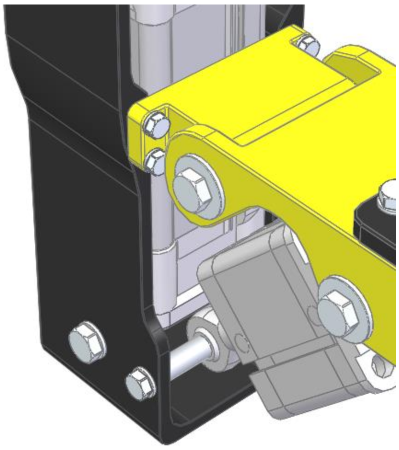

Figure 34: Fastened shafts supporting the angled cylinder and the vertical cylinder

For the upper bracket joints, a set of shafts of Ø 8mm were designed with internal threaded holes on each side. The reason why the shafts had a diameter of 8mm was for the operator seat to be supported and allowed to translate smoothly with journal bearings from Igus AB with dimensions of 8x10x5mm(Ød_i *Ød_o*w). To ensure a good and firm fixing of the shaft, the shafts feature M4-threads only to lock it from moving sideways when inserted in the bracket joints seen in figure 35.

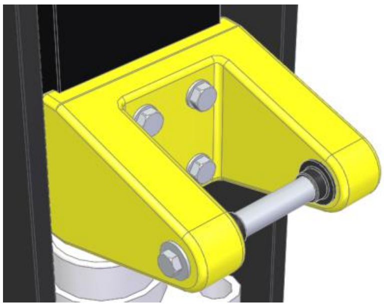

Figure 35: Shaft inserted and fastened in the upper bracket joint.

The design for the last shafts to make was totally different from the other shafts. Not only that they could not be threaded on both sides due to the linear rail guide but also because of the space within the suspension housing. To solve this, a compact design of a shaft of Ø16mm was made featuring an integrated M10-thread on a part of it which sits in the side of the sliding bracket. To fasten the rod eye of the vertical cylinders, the shaft is inserted through the first hole and later screwed into the threaded hole of the sliding bracket with the help of a screwdriver. Figure 36 show the shaft fixed in the sliding bracket.

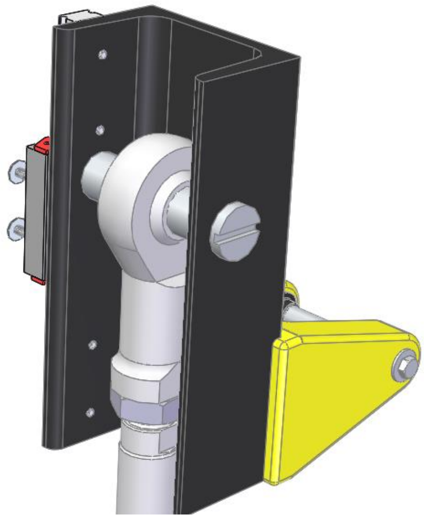

Figure 36: Shaft inserted and screwed through the rod eye into the sliding bracket.

3.3.9 Manufacturing methods for the parts

Table 1 below presents the manufacturing methods for each part.

Table 1: Manufacturing methods for every part within the undercarriage.

PART NAME	MANUFACTURING METHODS
Bottom plate	Waterjet-cutting, grinding, drilling and threading
Top plate	Waterjet-cutting, grinding and drilling
UNP-beam	CNC-milling, grinding and drilling
Steel angles (brackets)	Cold saw-cutting, grinding and drilling

Scissor arms	Waterjet-cutting, grinding and drilling
Suspension module housings	CNC-milling, grinding and drilling
Mounting brackets (upper and lower)	CNC-milling, grinding and drilling
Sliding brackets	Cold saw-cutting, grinding, drilling and threading
Fixing shafts and scissor arm bushings	Lathing, drilling, grinding and threading. Extra for sliding bracket shaft: saw-cutting for screwdriver

3.4 FEM-analysis

Now that the undercarriage reached it final design, it was time to make sure that all the major components within it could withstand the forces that evolves in the operator seat in the cabin. Full-scale images of the analyzed parts can be found in *9. APPENDIX*. For the analyses of all the parts, structural steel was used as material since they will be made in steel during the manufacturing process.

The FEM-analysis started by analyzing the most crucial parts, the bushings for the scissor arm. The CAD-model of the bushing was simply imported to ANSYS where it was set up according to figure 37.

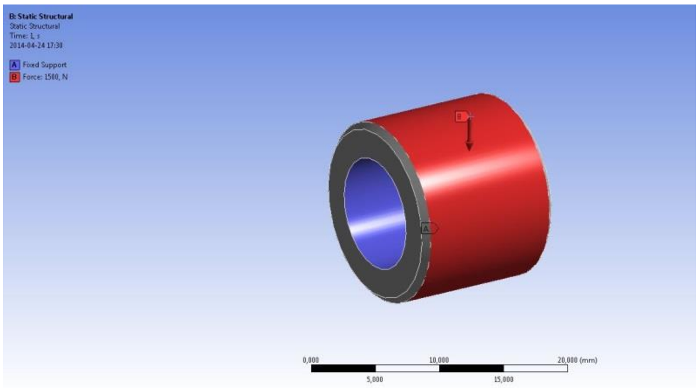

Figure 37: Setup of the scissor arm bushing in ANSYS.

The bushing was fixed in its internal M10 thread since it is held in place by an M10 bolt between the steel angle brackets. As the maximum load within the operator seat was previously calculated to 250kg (roughly 2.5kN), with a dynamic load reaching 5kN, the force acting on it vertically was decided to be with a magnitude of 1.5kN. This was done by increasing the dynamic load with 20% from 5kN to 6kN, and later divides it over a set of four arm bushings which gives 1.5kN each. By setting the material to structural steel, the analysis was started giving out the results presented in figure 38.

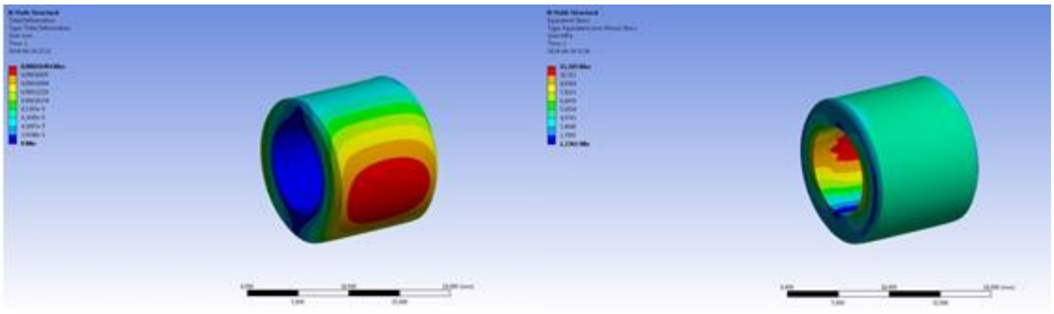

Figure 38: Left, deformation distribution. Right, distributed stress levels.

The results of this analysis show the maximum of deformation calculated to be 0.18µm and a maximum stress level of 11.27MPa. By these values, the bushing should work to support the scissor arms.

The next parts that were analyzed using FEM were the steel angle bracket joints since they were connected with the scissor arm bushings. One of these steel angles was imported to ANSYS were it was set up according to figure 39 below.

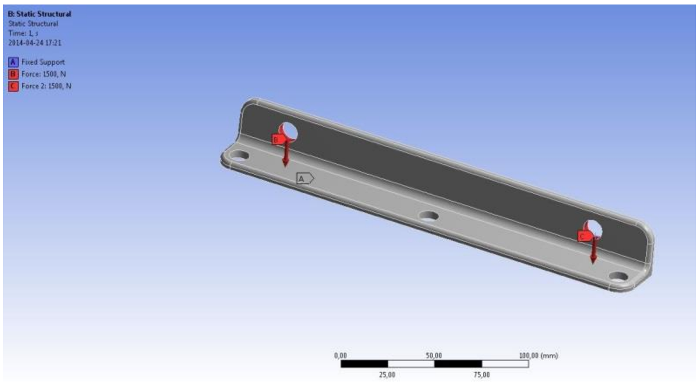

Figure 39: Steel angle bracket setup in ANSYS.

By fixing the bottom surface and applying vertical load of 1.5kN in each of the holes, same load as for the bushings, the analysis could be performed. The choice of material was decided to be structural steel for the analysis and the results showed both maximum stress levels as well as maximum deformation in figure 40. For the deformation, the peak levels reached 2.1µm and the stress levels peaked at 21.6MPa found inside the holes.

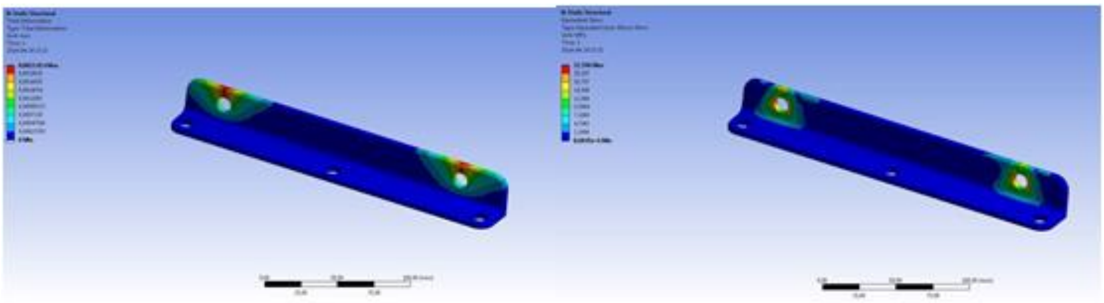

Figure 40: Left, deformation distribution. Right, distributed stress levels.

As the steel angles were analyzed, the scissor arms were analyzed as well. As the linear actuator would take up most of the load rather than the back pair of scissor arms, the front pair were analyzed and ensured that they would cope with the forces. By simply the assembly of the reinforced front scissor arms into ANSYS, they were set up as seen in figure 41.

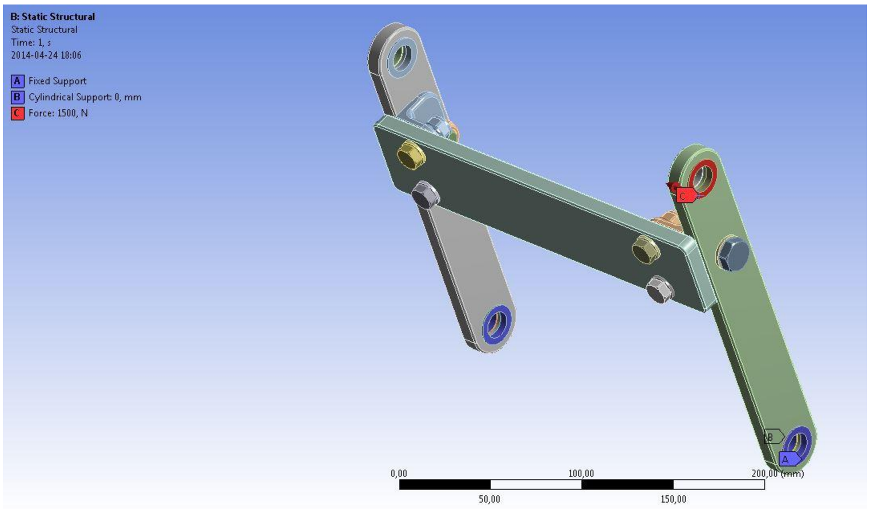

Figure 41: Reinforced front scissor arms setup in ANSYS.

The scissor arms were loaded with the same magnitude as the bushings, 1.5kN of force and fixed in the flanges of the bearings so they would moves sideways as they are mounted in the complete assembly. Then they are fixed using cylindrical support as they are free to rotate along the bushings. Performing the analysis of the setup gave a maximum deformation of 0.37mm found in the top hole, and a maximum stress of roughly 166MPa found inside the lower hole seen in figure 42. As the load was increased than what it would have been in real life, the scissor arm should cope with the dynamic load resulting in a far less deformation and stress level.

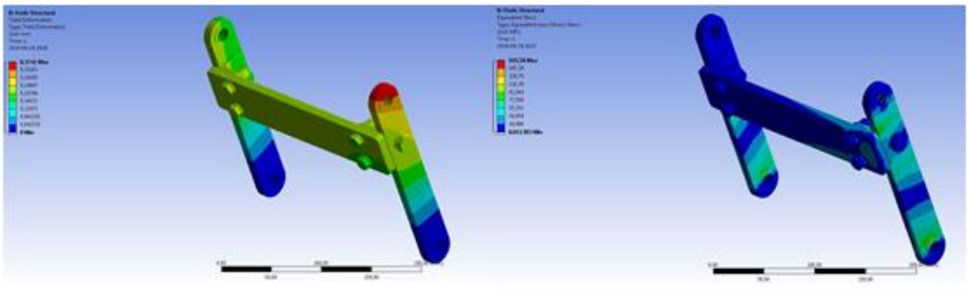

Figure 42: Left, deformation distribution. Right, distributed stress levels.

The lateral UNP-beam which basically is the most important part since it supports the suspension modules which support the operator seat must be able to cope with the dynamic loads within the undercarriage. The assembly of the beam with its mounting plates for the actuator was imported into ANSYS where it was set up as seen in figure 43.

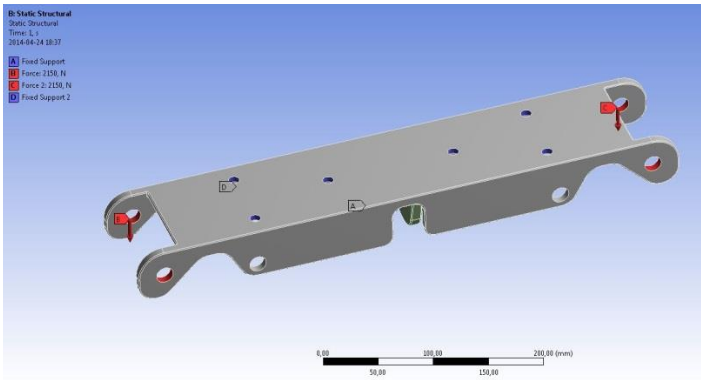

Figure 43: Lateral UNP-beam setup in ANSYS

By fixing it like it would be in real life by its holes for mounting screws and the HCB-bolt for the linear actuator, the load can be applied as two forces acting on the holes for the suspension modules, each of a magnitude of 2.15kN. The force 2.15kN was decided as sum of masses: 250kg for the seat (100kg) and an operator (150kg), 20kg for each suspension module giving 40kgs in total. The sum so far is 290kg, and by applying a safety factor of 1.5 to ensure that the system would work, the resultant load is calculated to be roughly 4.3 kN. By dividing the sum of load over the two shafts, this gives 2.15kN each. Performing the analysis of the setup gave a maximum deformation of 0.26mm found at the tip of the sides due to bending, and a maximum stress of 161.6MPa seen in figure 44.

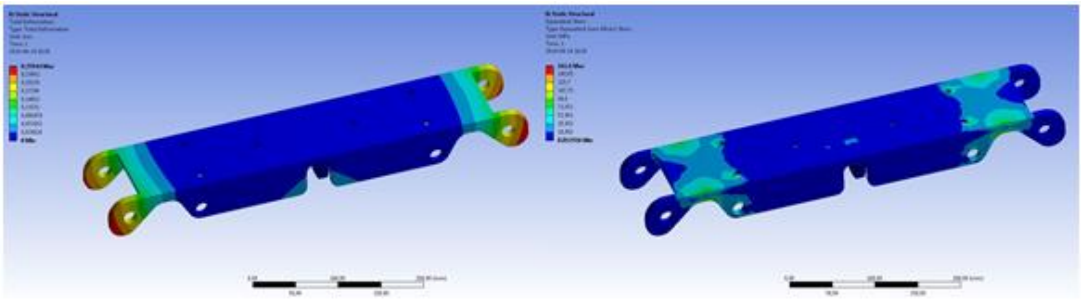

Figure 44: Left, deformation distribution. Right, distributed stress levels.

As the undercarriage would not necessarily be situated in the loads used in the analysis, the UNP-beam is considered to be fully capable to work in the prototype.

The lower bracket joint was the next part to analyze since it is connected to the UNP-beam. The part model was imported into ANSYS were it was set up as seen in figure 45. The part was fixed in the shaft hole and load of 2kN was distributed over the mounting holes to see how the deformations and stress levels would occur. The reason for using 2kN as a load was by summing up the loads and dividing them over two since there are two lower bracket joints. The sum of loads was based on: 250kg for the operator seat (100kg) and an operator (150kg) and an additional 40 kg for the two suspension modules, giving 290kg in total. To this number, a safety factor was added bringing the total load on two bracket joints as 4kN (roughly 400kg). Dividing the number gives 2kN on each bracket joint.

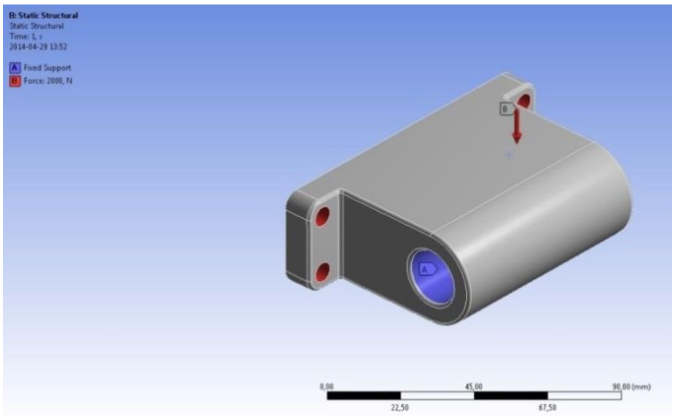

Figure 45: Lower bracket joint setup in ANSYS.

Performing the analysis gave a maximum deformation of 0.4µm found in the shaft hole, and maximum stress level of 19MPa found in the grooves of the bracket joint, both results shown in figure 46.

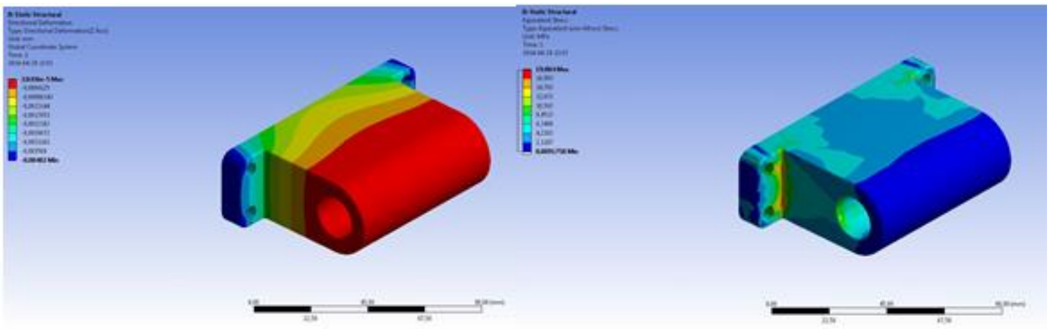

Figure 46: Left, deformation distribution. Right, distributed stress levels.

Dealing with these deformations and stress levels, the part can be seen as unharmed and therefore fully compatible with the prototype.

As for the upper bracket joints, they had to be analyzed in order to see if the re-design is stiff enough to withstand the forces within the cabin environment. The part was simply imported into ANSYS where it was set up as seen in figure 47.

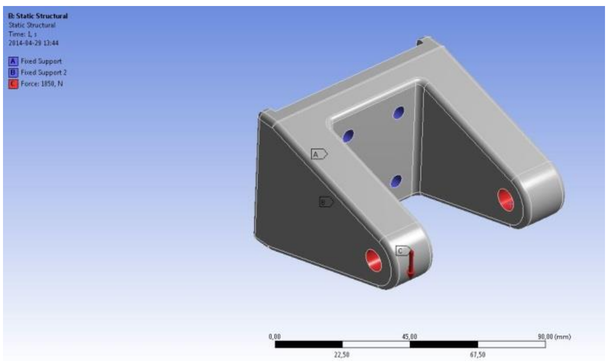

Figure 47: Upper bracket joint setup in ANSYS.

The part was fixed in its four mounting holes and the back surface which is mated with the sliding bracket in the complete model. For the load, a vertical force with a magnitude of 1.85kN was applied over the two holes for the shaft supporting the operator seat. The reason for applying such load was due to the following data: 250kg for the operator seat (100kg) and an operator (150kg), including a safety factor of 1.5 giving roughly 3.68kN. Dividing it over two bracket joints gives 1.84kN each which is rounded up to 1.85kN. As the analysis was performed, it gave a maximum deformation of 4.4μm found by the holes for the shaft and a maximum stress peaking at 11.78MPa in the grooves of the bracket joints, both results shown in figure 48.

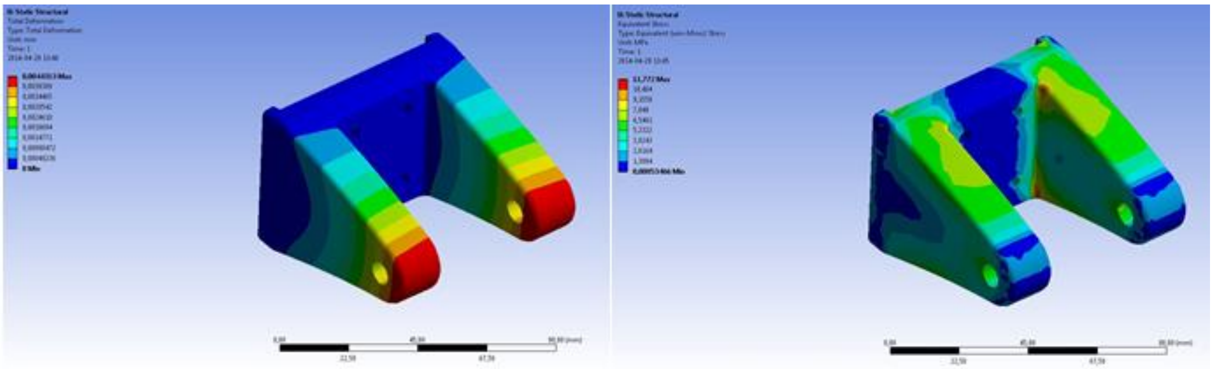

Figure 48: Left, deformation distribution. Right, distributed stress levels.

By these results, the part can with ease withstand the forces in the undercarriage.

For the shafts, the only crucial part in need of analyzing was the shaft supporting the operator seat, mounted in the upper bracket joint. The analysis was made due to the shafts small diameter. The part was imported into ANSYS where it was set up according to the figure 49 below.

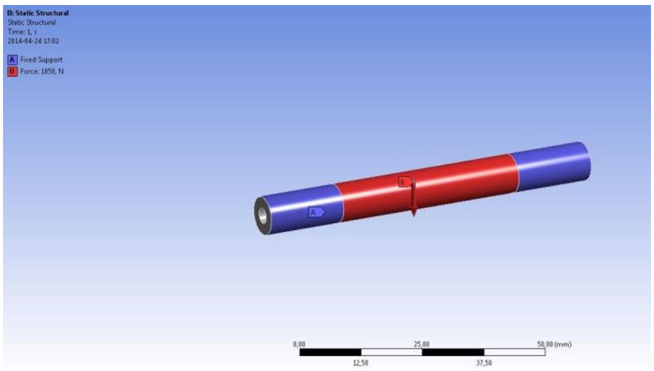

Figure 49: Bracket joint shaft Ø 8mm setup in ANSYS

The shaft was fixed in the surfaces taken up by the upper bracket joint's holes and a load of 1.85kN was applied same as for the bracket joint. As the analysis was performed, the results displayed a maximum deformation of 10 μm in the middle of the shaft and stress levels peaking at 69.3MPa, also found in the middle of the shaft and the sections close to the holes of the bracket joint. The results can be seen in figure 50 below.

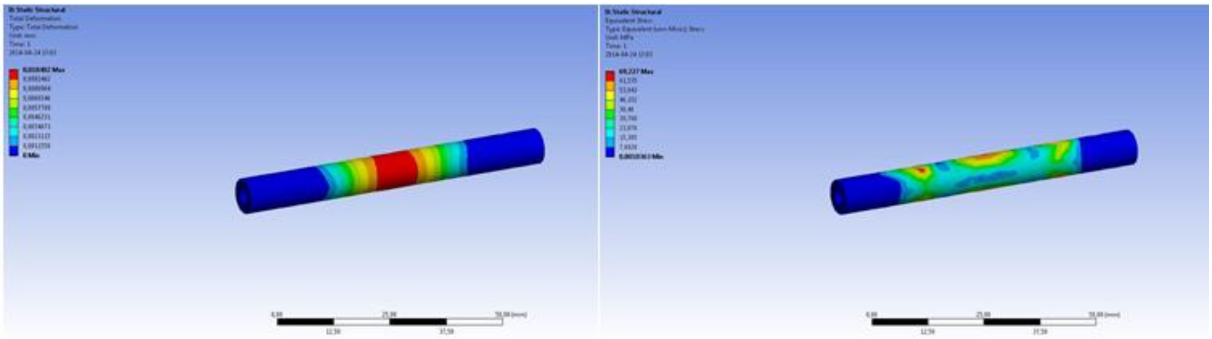

Figure 50: Left, deformation distribution. Right, distributed stress levels.

From these results, the part can be seen as fully working.

3.5 Ordering of standard components and material

As the CAD-model reached its final design, it was time to order all the necessary standard components in order to build the full-scale prototype. In the following sections are the needed components and material described in short.

3.5.1 Material and standard components for the prototype

The bullet points below describe the needed materials and from where the parts were ordered and bought.

- The material for the top and bottom plate, scissor arms and the reinforcement plate as well as the fixing plates of the linear actuator were decided to be made in 10mm thick steel plate. This material was found available at KTH.
- Standard UNP-beam of 100x50mm for the lateral beam was found available at KTH.
- Standard steel angle 30x30x5mm for the steel angle brackets, and the UNP-beam 65x42mm for the sliding brackets had to be ordered from a material supplier to KTH.
- Standard steel angle 45x45mm for the bracket joints for fixing the linear actuator was found available at KTH.
- Steel rods for the shafts had to be ordered from a material supplier to KTH.
- Steel block of 110x110x1500mm had to be ordered for all the upper and lower bracket joints as well as for the housings of the suspension modules. Also from material supplier to KTH.

The standard components needed for the prototype were the following:

- All fastening elements (bolts, washers, nuts etc.) as well as the rubber dampers were ordered and bought from Wiberger (Wiberger, n.d.)
- Pneumatic cylinders XL-063-80 and NXD-063-30, all with rod eyes and double ear mounting plates, were ordered and bought from AirTec (AirTec, 2014)
- All journal bearings were bought from Igus AB (Igus AB, 2014)
- The linear rail guides for the suspension modules were bought from Schneeberger (Schneeberger, n.d.)
- The linear actuator was ordered and bought from Reac AB (Reac, 2014)

3.5.2 Standard components for demonstration of the prototype

When the prototype is manufactured and assembled, a demonstration of it is planned to be held at Skogforsk headquarters in Uppsala during the final thesis presentation 4^{th} of June. Since the prototype will not feature active suspension till that date, it will instead be manually controlled using manual 5/3-valves with control levers. During the final presentation, the height adjustment as well as the motions of the suspension module are planned to be demonstrated. For them to be fully working, the prototype must be geared with the following equipment from AirTec:

- Three manually controlled 5/3-valves with levers.
- Attachable quick-fit ports
- Angled quick-fit ports
- 10m plastic hose

The reason for using only three valves is due to the angled cylinders working in different directions. As the left angled cylinder is extending to overcome the forces, the right cylinder is retracting and helping the left cylinder to overcome the forces by pulling. This can be done using only one valve. The other two valves are used to control the suspension modules separately.

4 RESULTS

In this chapter, the final design of the CAD-model of the undercarriage is presented and the manufacturing process of the prototype is described. Lastly, the chapter presents the full-scale prototype ready for demonstration.

4.1 Final CAD-model

Before the actual manufacturing of the prototype, the design had to be verified. After presenting the final design to both Björn Löfgren and Ulf Sellgren, it was decided to start manufacturing the prototype. The final CAD-models can be seen in the figures 51-53.

Figure 51: CAD-model of an operator seat mounted on the final undercarriage design.

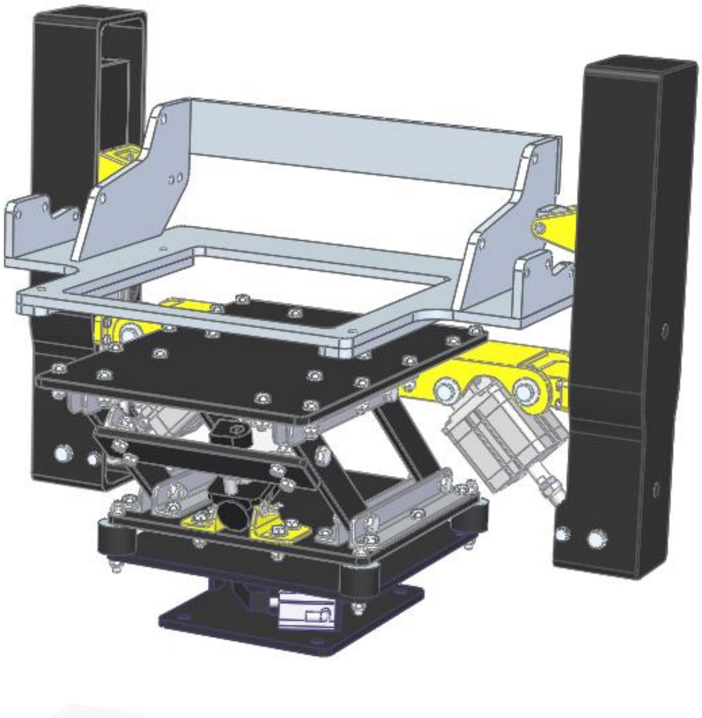

Figure 52: Complete final design of the undercarriage.

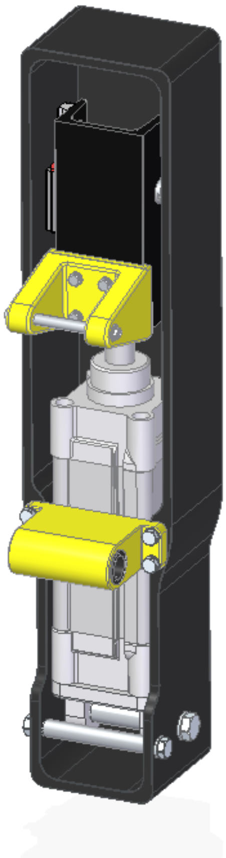

Figure 53: Complete final design of the suspension module.

The only this to keep in mind though is to ensure that there would not be any clamping risk of fingers which is solved by attaching protective curtains. After consulting with Björn Löfgren, the swivel base-160mm as well as a protective rubber below from an existing undercarriage was received from Skogforsk to be used in the prototype.

From this point and forth, the manufacturing process is described.

4.2 Prototype manufacturing

The following sections describe the manufacturing process for each part of the complete prototype assembly. The last section describes how the prototype is being assembled.

4.2.1 Suspension module housings, upper and lower bracket joints and UNP-lateral beam

As the design of the suspension modules was finalized, STEP-files of the housings, the upper and lower bracket joints were exported from Solid Edge ST4 and sent to Industrial Production facilities at KTH where CAM-programs for each part were generated. The plan was in an early state to manufacture all the parts in a five-axis CNC-mill called HERMLE. However, due to machine malfunction and overbooked machine schedule, the plan had to be changed. Instead, a 3-axis CNC-mill called MASAK was used to mill the parts, resulting in somewhat uneven surfaces. For the mill, the steel block that was ordered and received was cut into proper lengths for each part.

Unfortunately, the suspension module housings were not manufactured in time due to delay in the production facility due to machine failure. However, the rest of the parts were manufactured.

4.2.2 Top and bottom plate, scissor arms and reinforcement plate

The 10mm steel plate for the top and bottom plate as well as for the scissor arms and reinforcement plate was placed on the water-jet bed to be cut into the different parts. The water-jet cut out all the parts from the steel plate and marked out the positions for all the holes in the parts to be later on drilled manually. As the parts came out from the water-jet, the edges were grinded down and holes were drilled according to the CAD-models. The holes for the M8-screws in the bottom plate were threaded (seen in figure 54) to be later used for fastening the steel angle brackets.

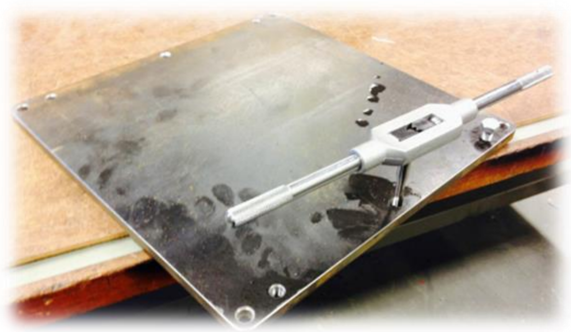

Figure 54: Bottom plate being threaded.

4.2.3 Bushings and shafts

As the steel rods for the bushings and all the shafts was ordered and received, 2D-drawings were made and handed to Thomas Östberg to be manufactured. The rods were cut into proper lengths and fixed in a lathe to be lathed. When the rods were lathed down to the correct diameter, they were cut into the dimensioned lengths and got holes drilled into them to be later on threaded according to the drawing. The shafts for the sliding brackets however were threaded externally at the ends.

For the bushings, a steel rod was cut and installed in the lathe. As soon as the wished diameter was reached, the rod was cut into smaller pieces to be formed as bushings. These parts were then drilled and threaded with M10-thread.

4.2.4 Steel angle brackets and sliding brackets

To prepare the steel angle to be cut into lengths for the steel angle brackets, 2D-drawings were made and printed. The steel angle was installed in a cold-saw were it was cut into lengths for all the steel angle brackets. As the parts were cut and the edges grinded down, the holes for the fastening elements were marked out and drilled. When the parts were all made, they were aligned in both the top and bottom plate to ensure that the holes were properly drilled seen in figure 55.

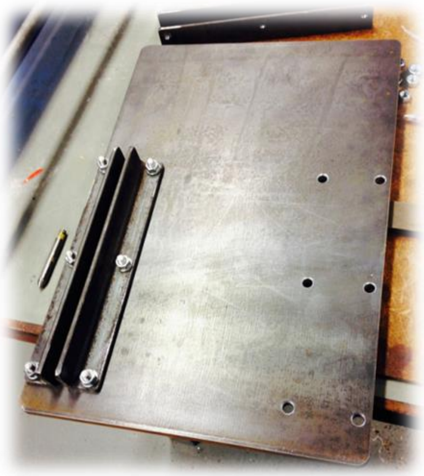

Figure 55: Steel angle brackets mounted on the top plate.

For the sliding brackets, the UNP-beam 65x42mm was cut using the cold-saw into the lengths used in the CAD-model.

4.2.5 Seat fixing bracket

By contacting the previous thesis worker Omar Osorio, it turned out that the bracket which is hanged in the upper bracket joints was not a standard part and was only designed to be used to fix the seat in the suspension modules. Therefore, the part was re-designed in 10mm thick steel plate by taking measurements from the old bracket. As the design was finalized seen in figure 56, it was analyzed in ANSYS to see if the design was stiff enough to overcome the forces.

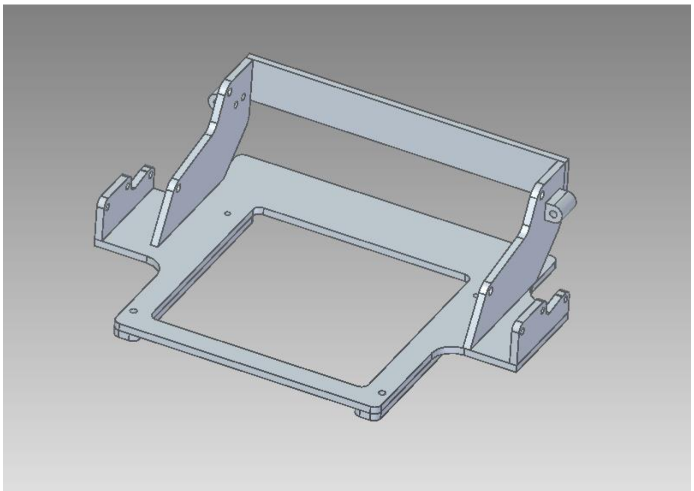

Figure 56: Final design of the seat bracket

The bracket was imported into ANSYS where it was set up as seen in figure 57.

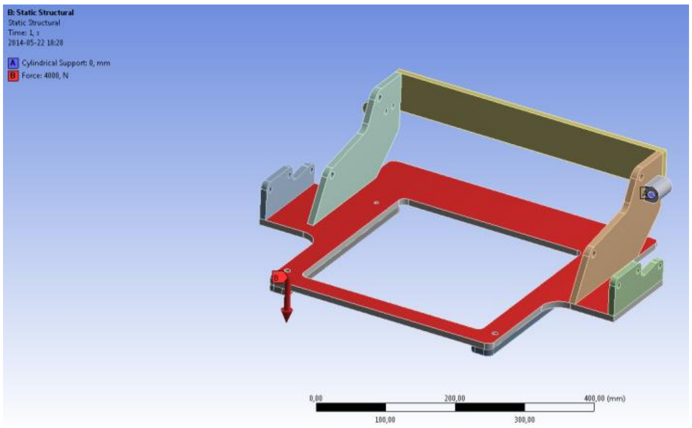

Figure 57: Seat bracket setup in ANSYS

It was fixed using cylindrical support in its mounting holes and a force with a magnitude of 6kN was applied on its flat surface. Running the analysis, the results showed a maximum stress level of 137MPa found close to the mounting holes and a maximum deformation of roughly 7mm found in the front of the bracket seen in figure 58.

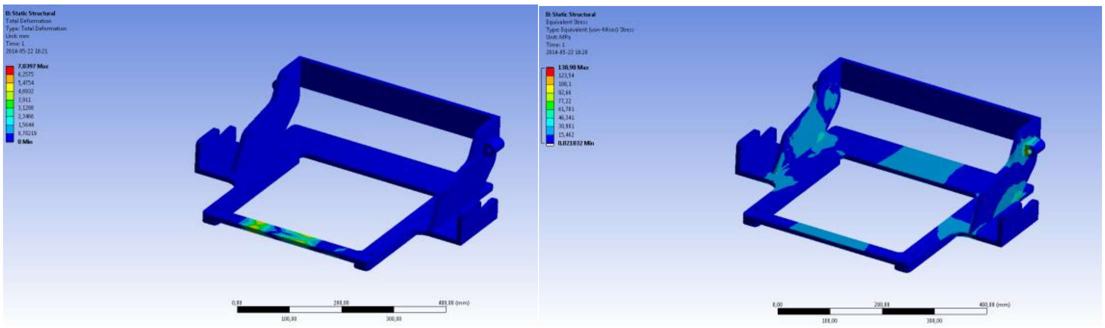

Figure 58: Left, total deformation distribution. Right, distributed stress levels.

By these values, it was assumed that the seat bracket would cope with forces since the vertical cylinders would take up most of the forces.

From here, dwg-drawings were made and sent to Björn Magnuson, a student at KTH who water-jet cut the parts out a 10mm thick steel plate. As the parts were cut, their edges were grinded down and the parts were welded together forming the bracket. Lastly, the mounting brackets were made by rounding the edges of two blocks of steel and mounting holes were drilled.

4.2.6 Prototype painting and assembly

As the manufacturing of the parts was heavily delayed, pictures of the assembly process were not taken to cover each step of the assembly. However, some pictures are presented in this section. Also due to this delay, the suspension modules were not assembled. See chapter *5 DISCUSSION AND CONCLUSIONS* for more information.

Before the prototype was assembled, all the manufactured parts excluding the shafts and bushings were spray-painted according to the colors used in the CAD-model. The parts can be seen painted in figure 59 below:

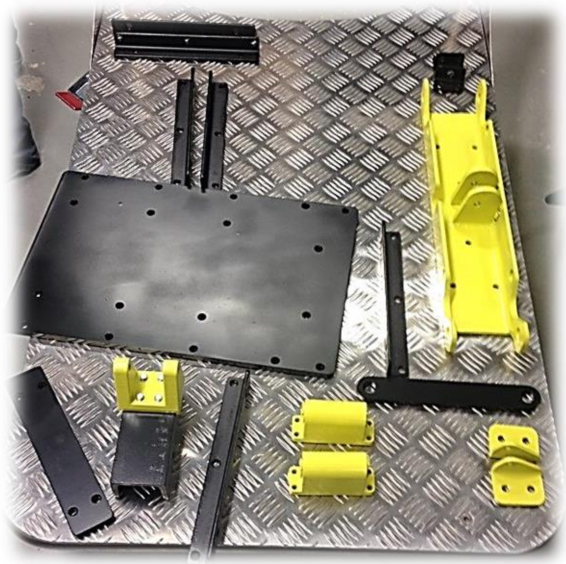

Figure 59: Painted parts of the prototype.

Since Skogforsk provided the prototype with a swivel base-160mm, it was first fixed on a EUR-half pallet as the prototype would be assembled onto it for easy transportation. As the swivel base was mounted, the rubber dampers, the bottom plate as well as the protective rubber bellow were fastened on it using washers and M8-nuts, seen in figure 60.

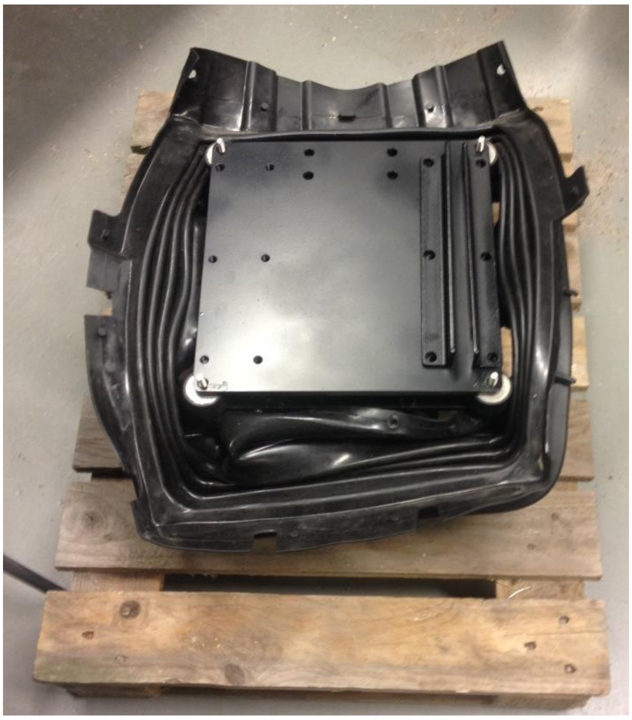

Figure 60: Bottom section of the prototype mounted on EUR-pallet.

To this assembly, the bushings for the scissor arms as well as the arms were placed between the steel angle brackets, secured with M10-bolts, nuts and washers. Before placing the arms into place, they had journal bearings fitted into them and for the front arm pair, the reinforcement plate and brackets were fastened. As the scissor arms and bushings were secured, the steel angle brackets were fastened into the bottom plate using M8-screws.

Same goes for fixing the linear actuator, the steel angle brackets for it were aligned with their holes in the bottom plate. Journal bearings were fitted into the linear actuator and later on, it was secured in-between the steel angle brackets with HCB-bolt, washers and scissor pins. Lastly, the brackets were fully fastened in the bottom plate with M8-bolts, nut and washer.

The steel angle brackets, bushings and scissor arms were secured with M10-bolts, nuts and washers. As they were, the steel angle brackets were fastened into the top plate using M8-bolts, nuts and washers.

The sliding brackets for the suspension modules were prepared by mounting the guide rails into them with M3-machine screws. As the rails were tightly fastened, the rod eyes of the XL-cylinders were fixed inside the brackets with the threaded shaft.

4.3 The final product for demonstration

In this section, photos of the half-assembled prototype for demonstration are presented in the figures 61-62.

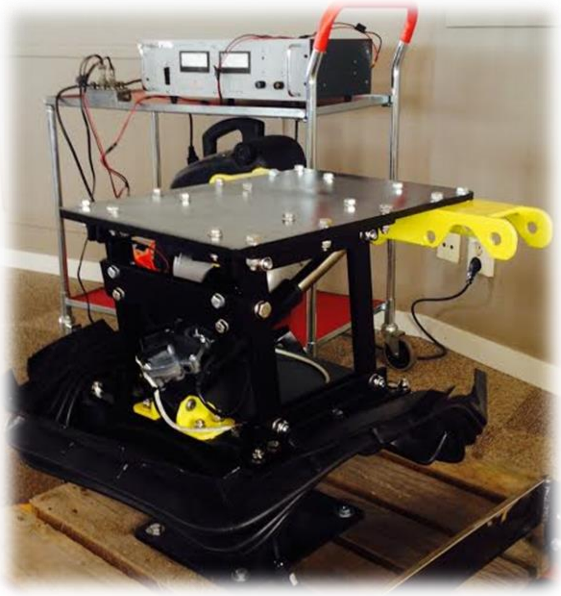

Figure 61: Half-assembled prototype connected to a 24VDC electric box for demonstration.

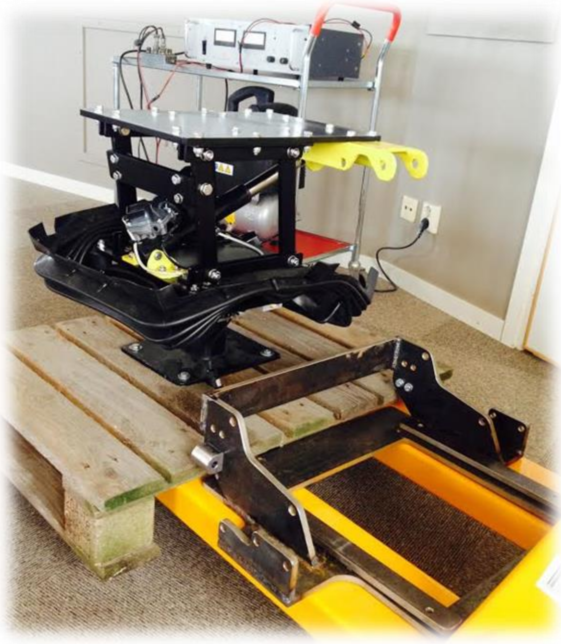

Figure 62: Half-assembled prototype next to the seat bracket.

5 DISCUSSION AND CONCLUSIONS

In this chapter, discussions of the overall work and issues obtained during the workflow are held. From this, conclusions are drawn in direct response to the goals presented in chapter 1 INTRODUCTION.

5.1 Discussion

In the beginning of the project, the expectations were set as very high such as to examine and re-design the previous generated concepts for the undercarriage, conduct both static and dynamic analyses as well as manufacture and test a full-scale prototype. As the prototype testing and function verification would take roughly 10 labor weeks to fulfill, the testing was out of the question. For the dynamic analyses, they were replaced with a quasi-static force-study where the load was set to be dynamic rather than static. However, as the goal was set from the beginning to realize and manufacture a prototype, it was fulfilled only that the system would only be prepared to be active controlled. In chapter *6 FUTURE WORK*, the work to continue this project is described.

The re-design of the complete undercarriage was only done by one person, namely the author of this report, where the aim was to design it to be easy to manufacture yet to fulfill its functions and be stiff and durable using as many standard components and manufacturing methods as possible. By this, the knowledge and the physical results were limited to what the person could manage. However, with good collaboration with employees at Ponsse, Skogforsk, as well as at KTH, the end result proved to be of good quality.

As the manufacturing of the parts was heavily delayed due to machine failure and other issues, so was the assembly of the prototype. By this, the suspension modules were not assembled at all. Also due to this delay, photos for this report were almost not taken as the prototype was being assembled due to short amount of time. The results showed a fully working half-assembled prototype missing the suspension modules. By this, the prototype was at that state capable of being adjusted in height. However, there is not much work left in order to fully assemble the prototype. The only work left is to manufacture the suspension module housings, paint them, and install the internal parts of each. As the suspension modules are fully assembled, they can be fixed in the top plate.

As only one person performed this work by his own, the testing of the prototype was not made due to time limitations and knowledge. By this, it is not said that the results were bad, but the planned activities were too much to done during 20 labor weeks. However, if this work was made by combining workers with background in machine design and mechatronics, the prototype would have been manufactured and tested during the scheduled time.

5.2 Conclusions

The conclusions of the project are the following:

- Two generated concepts by Teodor Doroftei and Osorio Omar were examined and described in detail.
- One concept was chosen to be developed further and re-designed.
- The re-design process resulted in a full CAD-model which in turn was verified to cope with the forces in the operator seat.
- The new design resulted in a half-assembled undercarriage featuring height-adjustment.
- The manufactured prototype is prepared to be active controlled. By this, to measure a minimum of 50% vibration dampening, the prototype must first be programmed and tested on a test platform at Skogforsk. This goal can be achieved in the future.

6 FUTURE WORK

In this chapter, the future work is described in order to achieve a fully active suspension for an operator seat in a forest machine.

6.1 Future work

The following work has to be made in order to achieve a fully working, active controlled seat undercarriage:

- The suspension module housings must be manufactured, painted and the internal parts installed in each. As this is done, the modules can be fixed in the top plate assembly.
- If the minimum position of the undercarriage is too high for short people, it can be adjusted by simply flipping the steel angle brackets of the actuator vertically and design a new bottom plate which features a cutout for the steel angle brackets and the linear actuator.
- The protective rubber bellow must be adjusted to be fastened into the top plate in order to cover the scissor arms and the linear actuator.
- Valves and air pressure regulators must be bought and controlled with a control box.
- To control the undercarriage, one must understand the movement of the undercarriage in order to design a control box. The control box should use accelerometers as input vibration collectors in order to control the needed air pressure which the cylinders are in need of to overcome the forces within the operator seat to keep it in place. From the vibration data, the control box should be programmed to control the needed input air pressure and valve settings to each cylinder.
- As the control box is programmed, the undercarriage has to be tested several times on a test platform to ensure that it works before testing it in a real forest machine.
- As soon as the prototype is fully active and working, weight optimization can be achieved in order to keep the weight down yet keeping the stiffness high.

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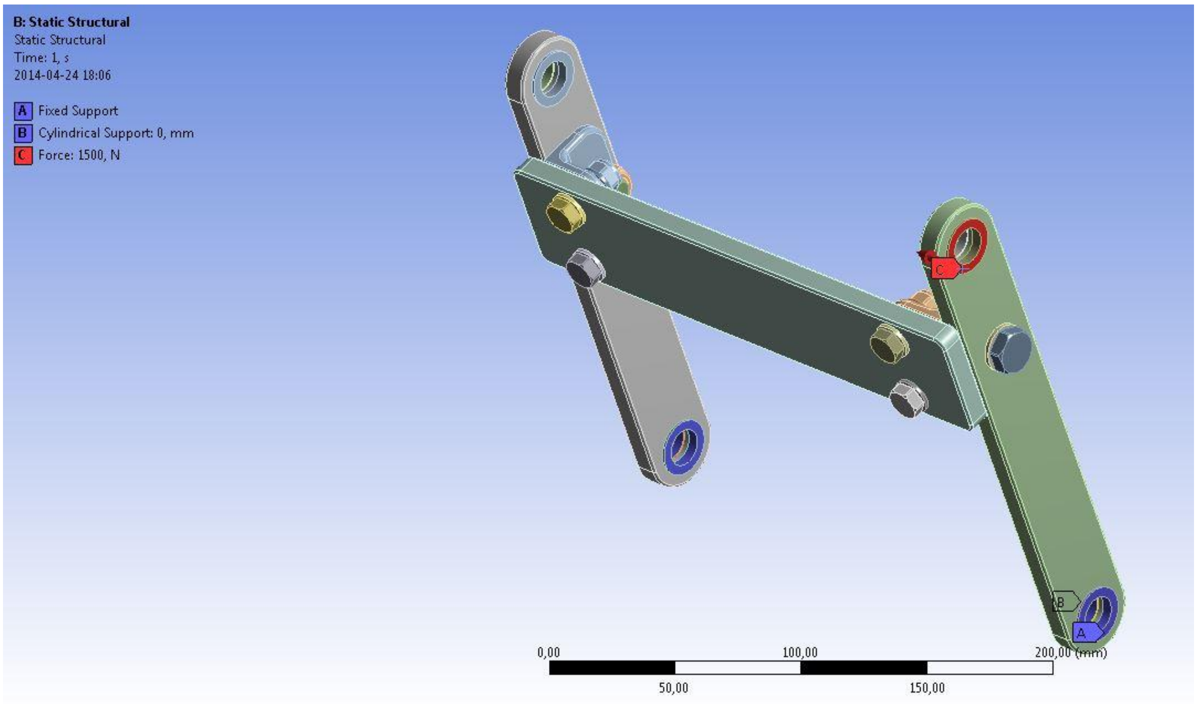
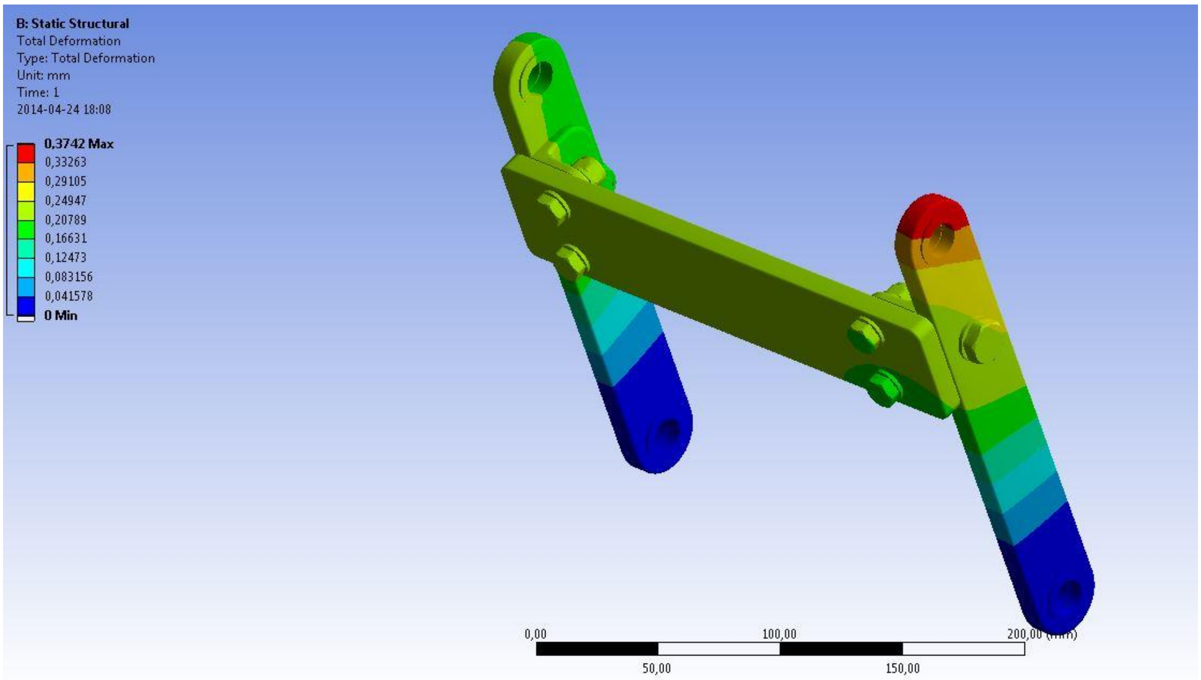
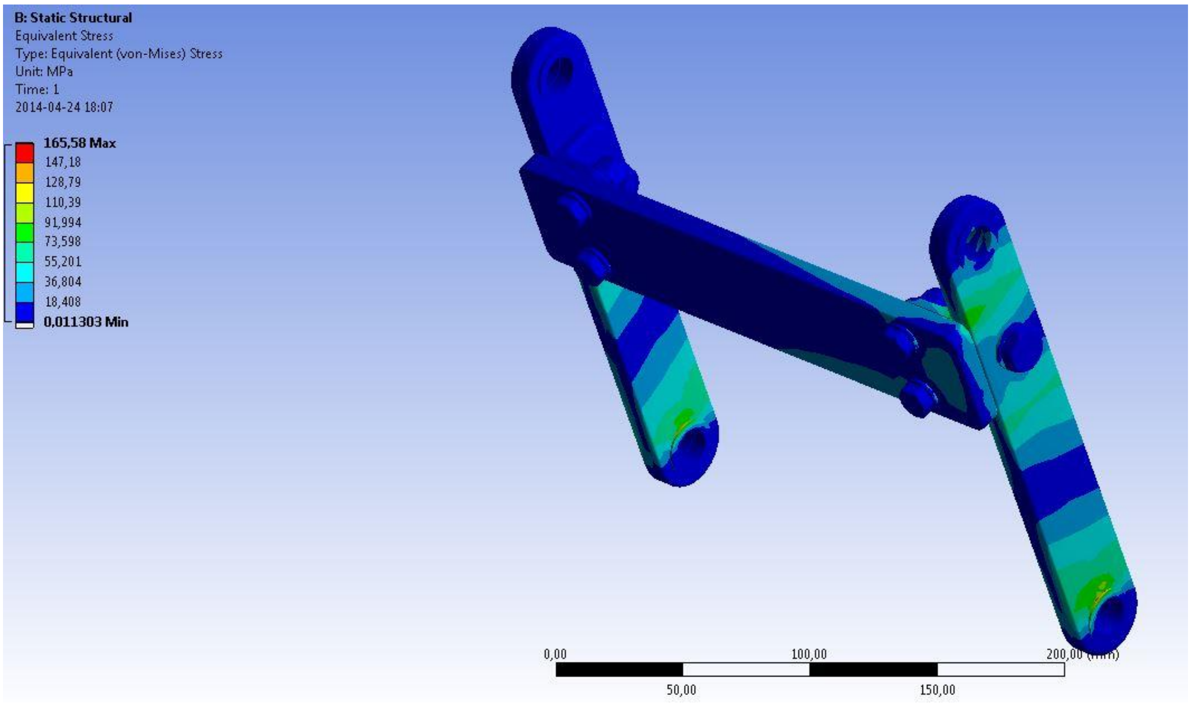
8 APPENDIX

In this chapter, the main results obtained from the FEM-analyses are presented.



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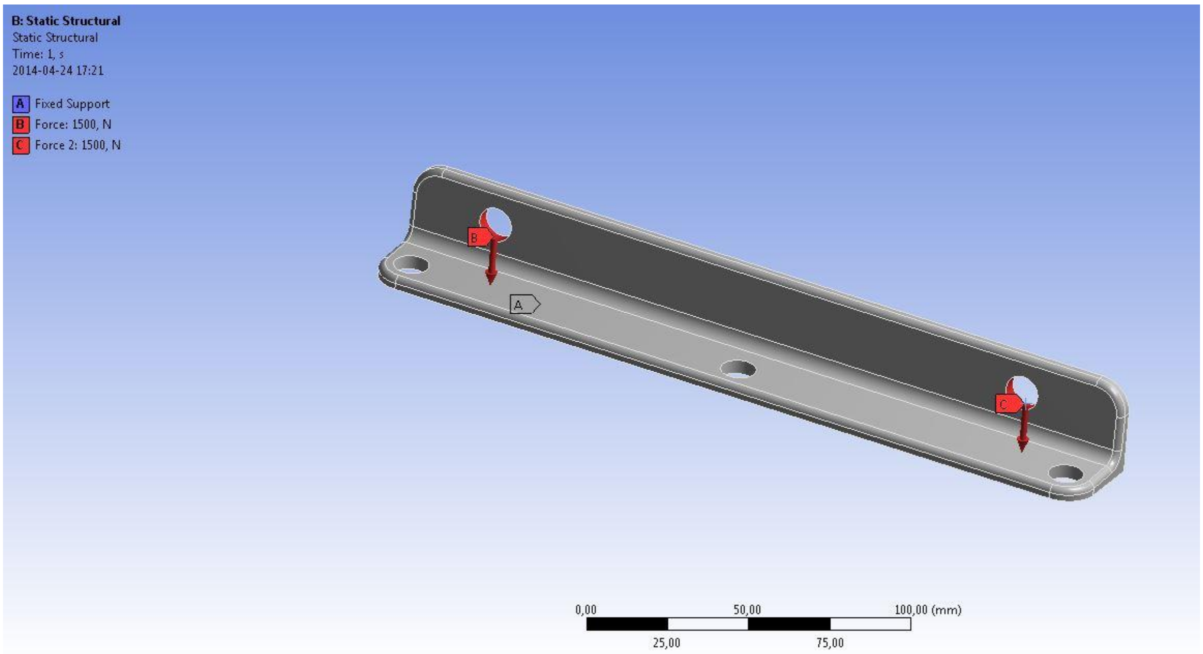
150,00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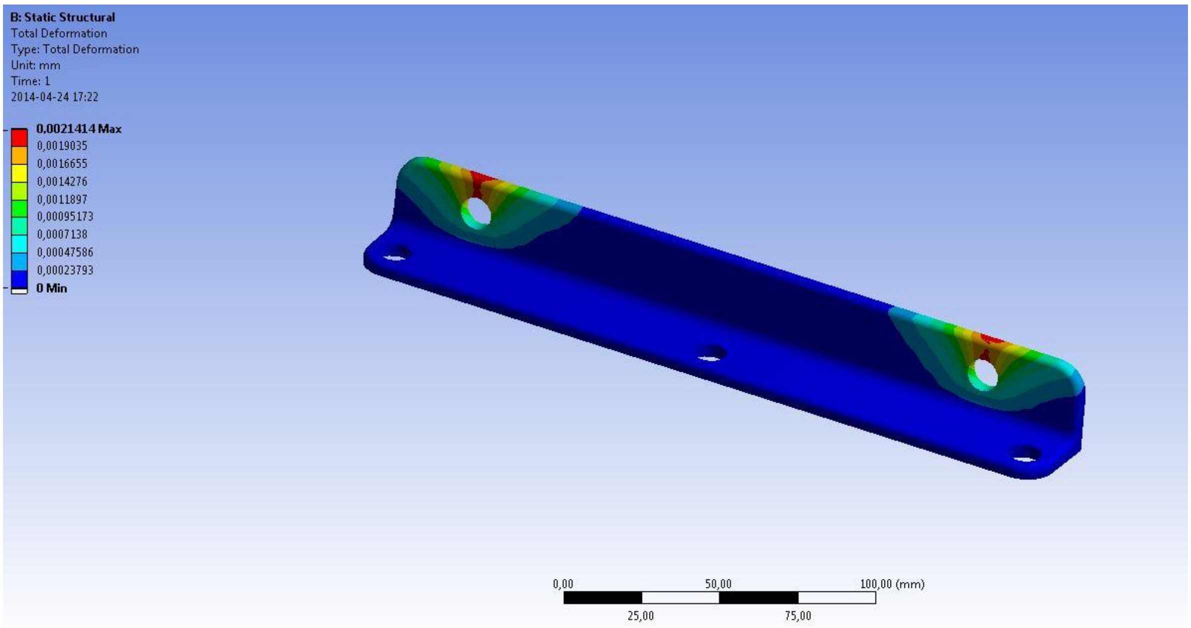
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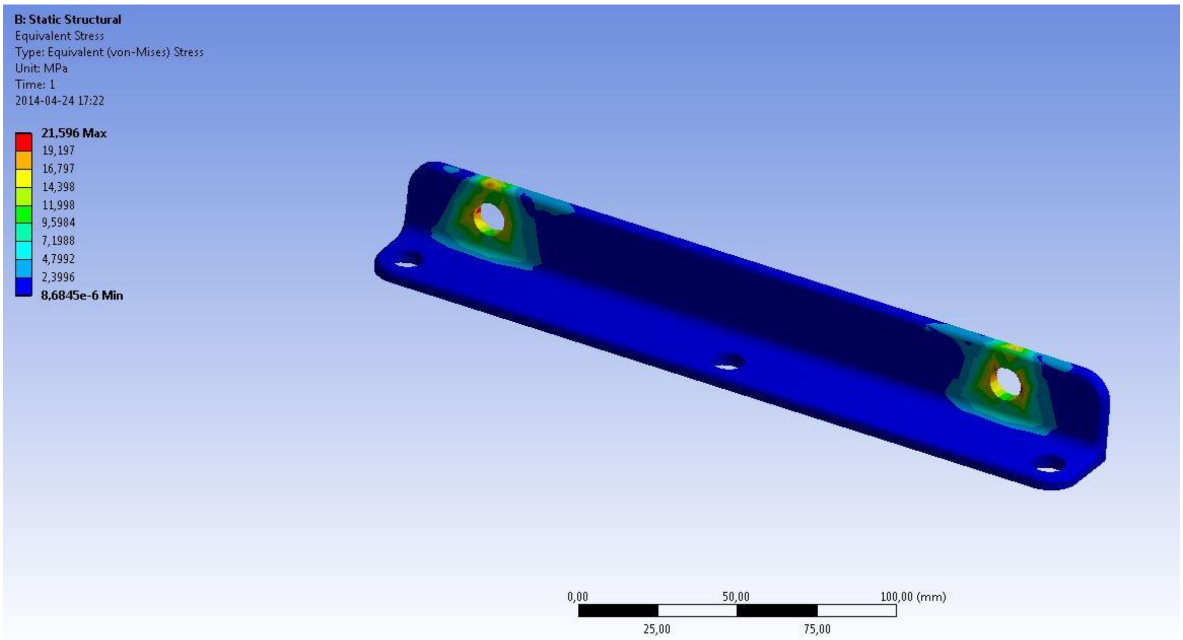
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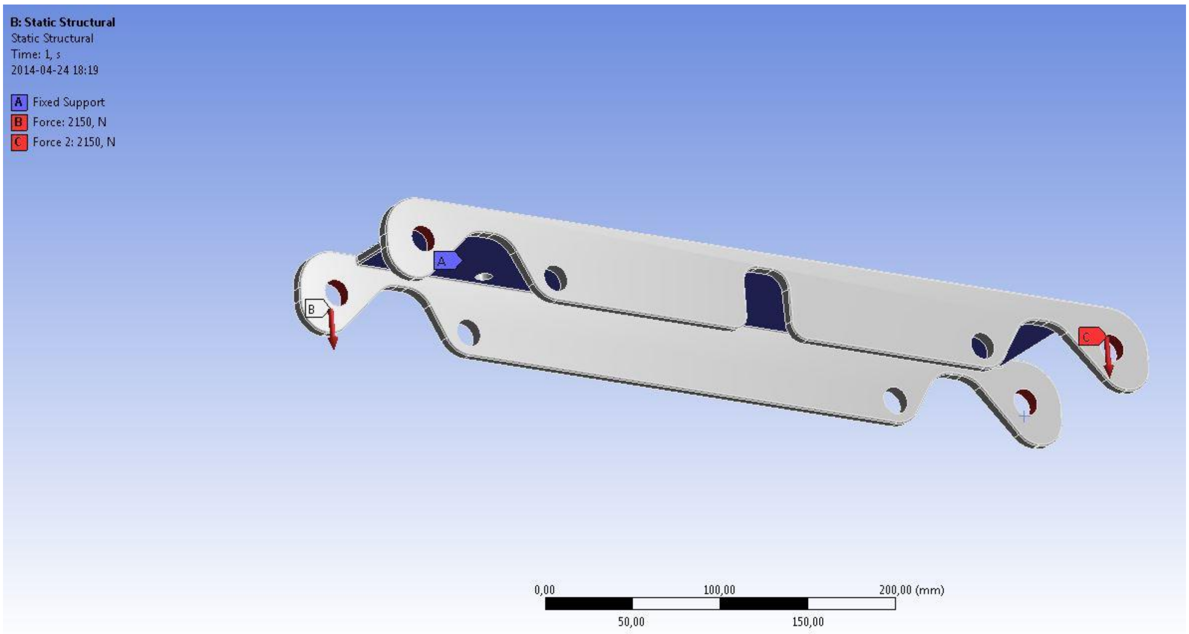
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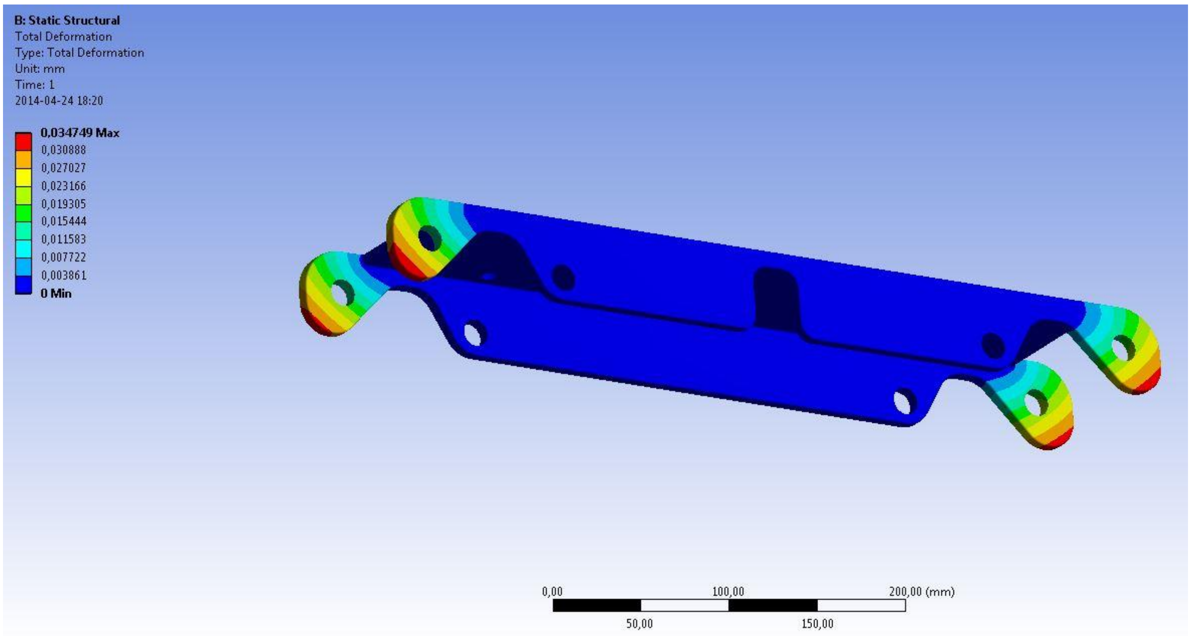
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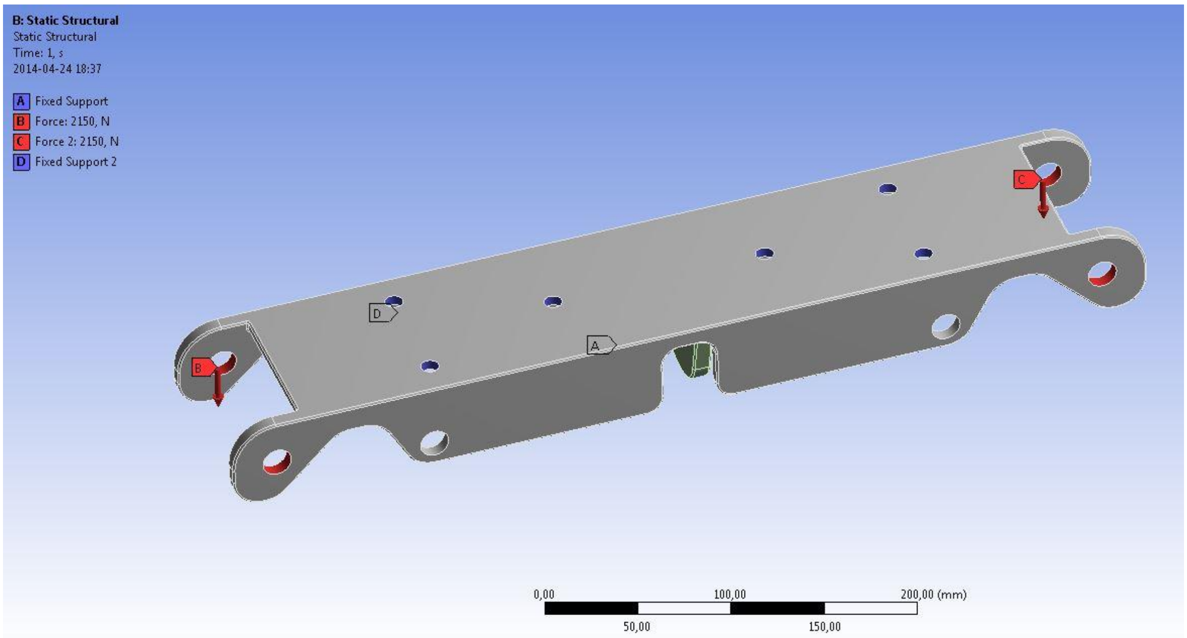
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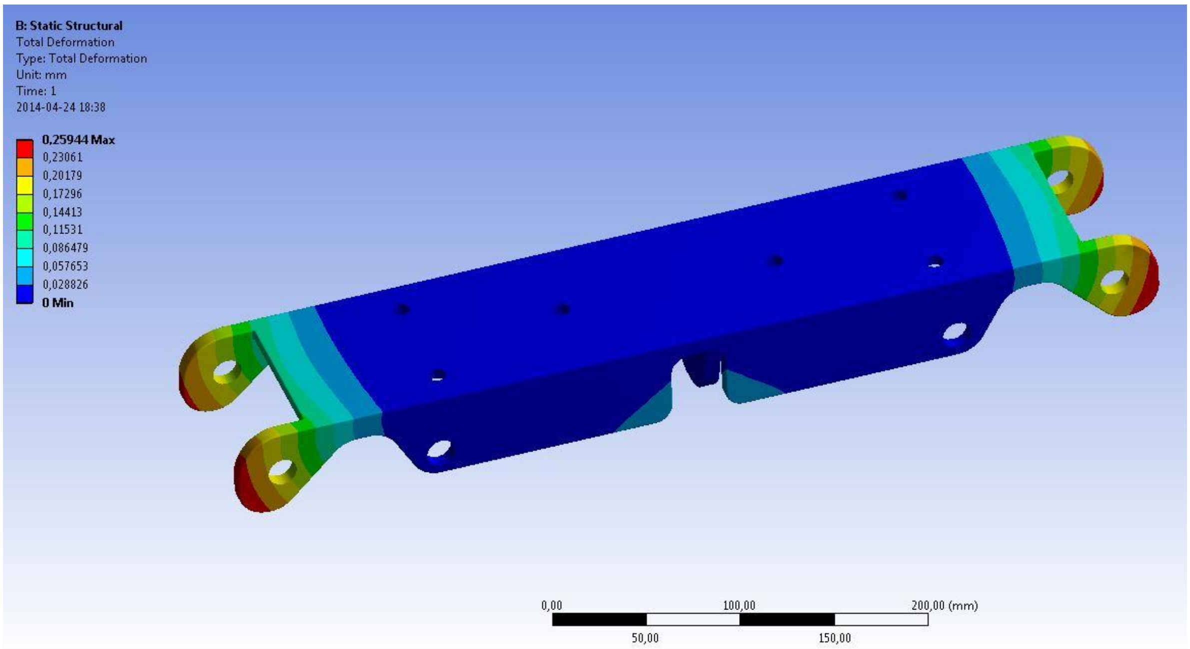
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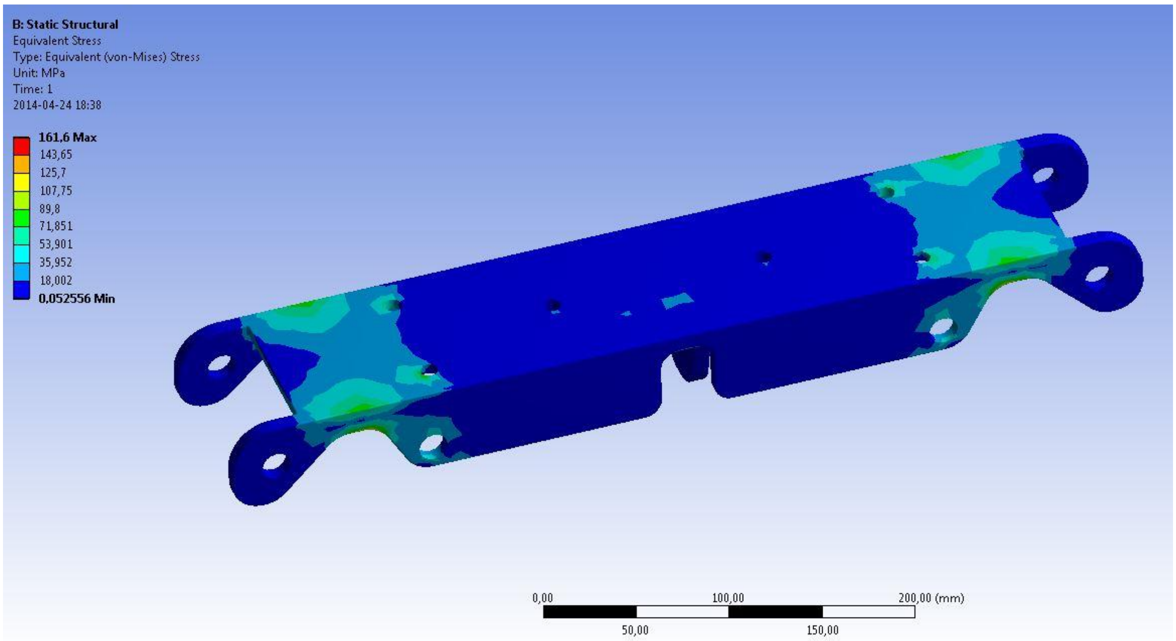
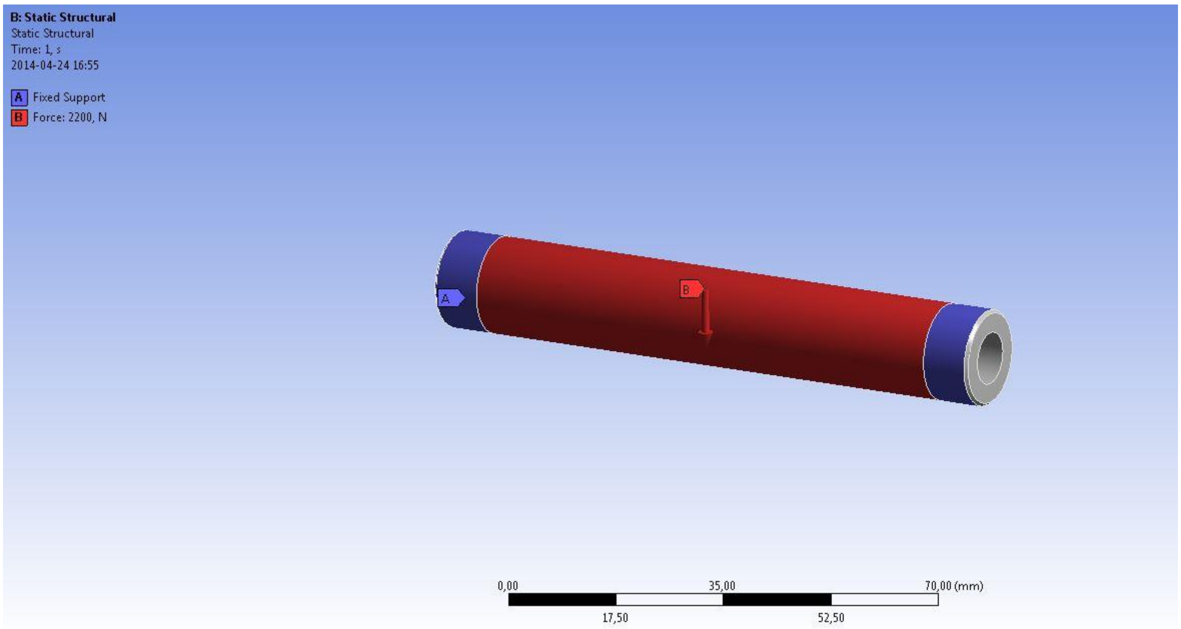
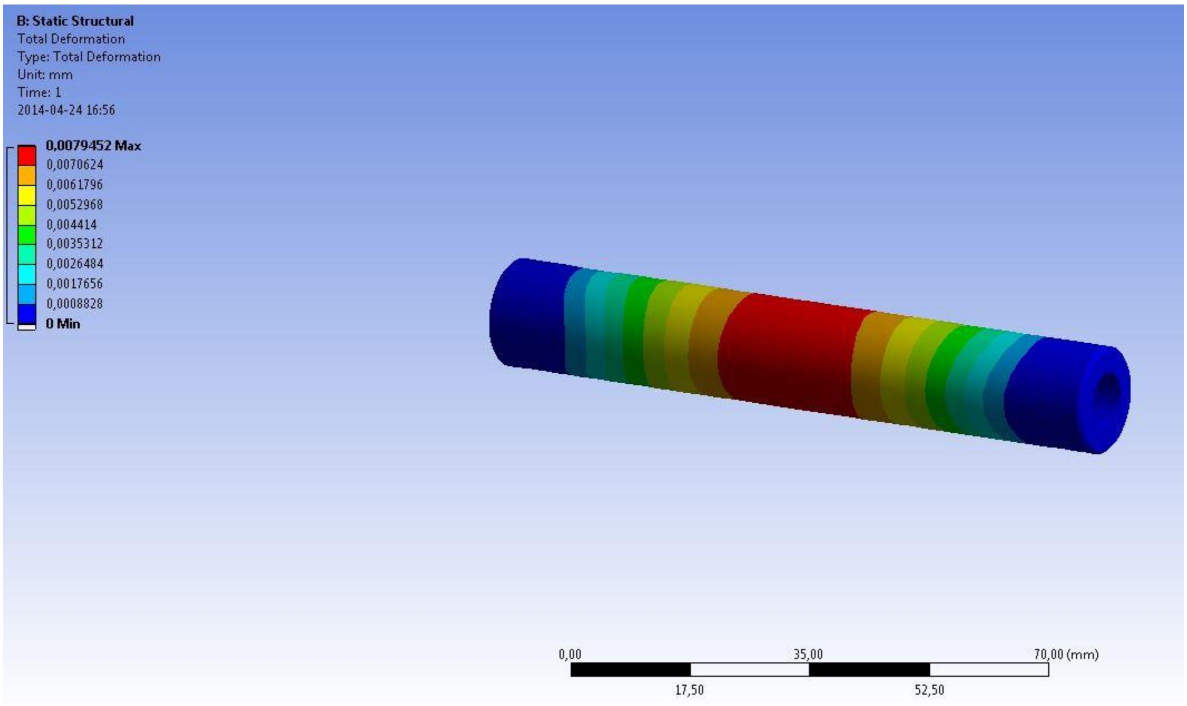
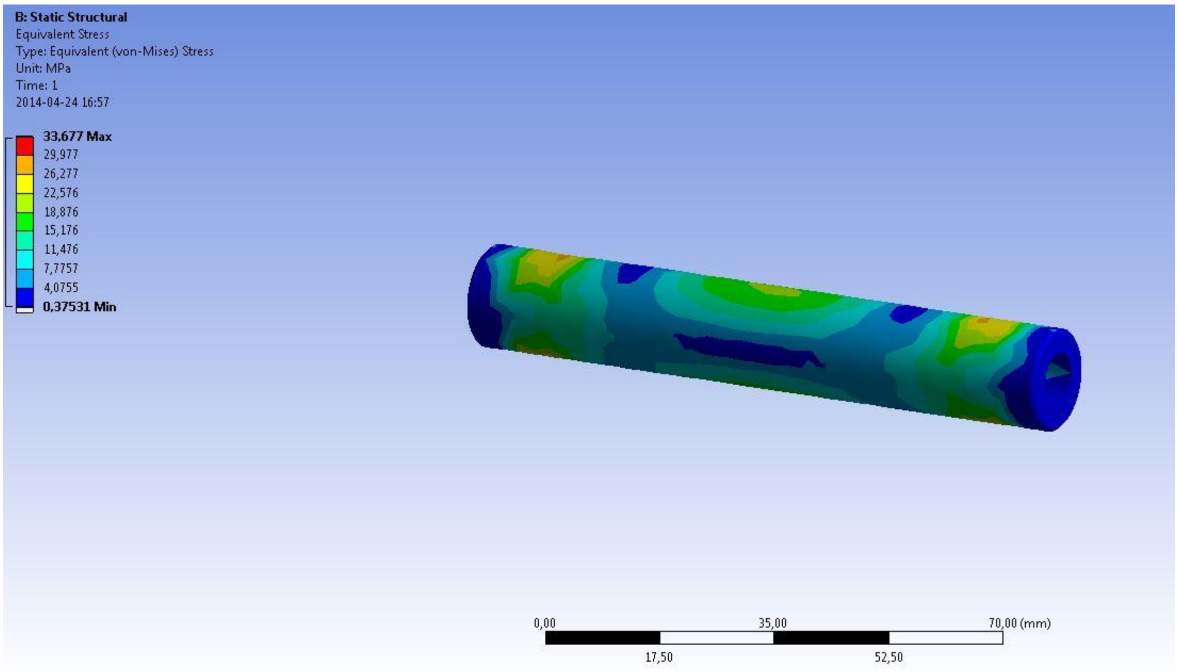
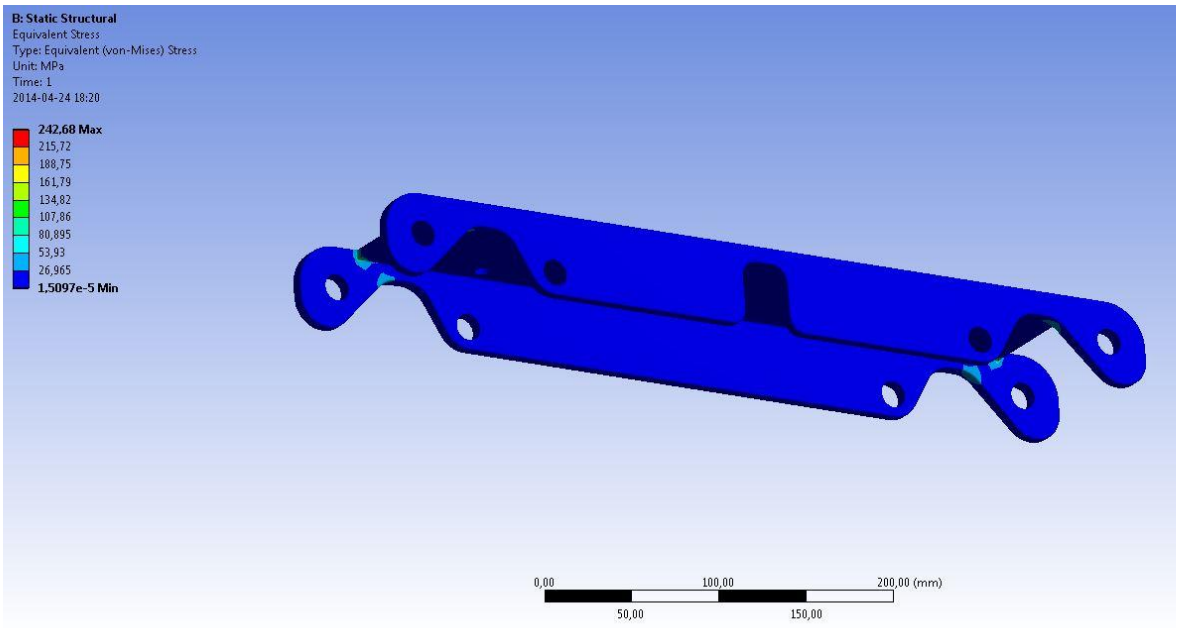
150,00

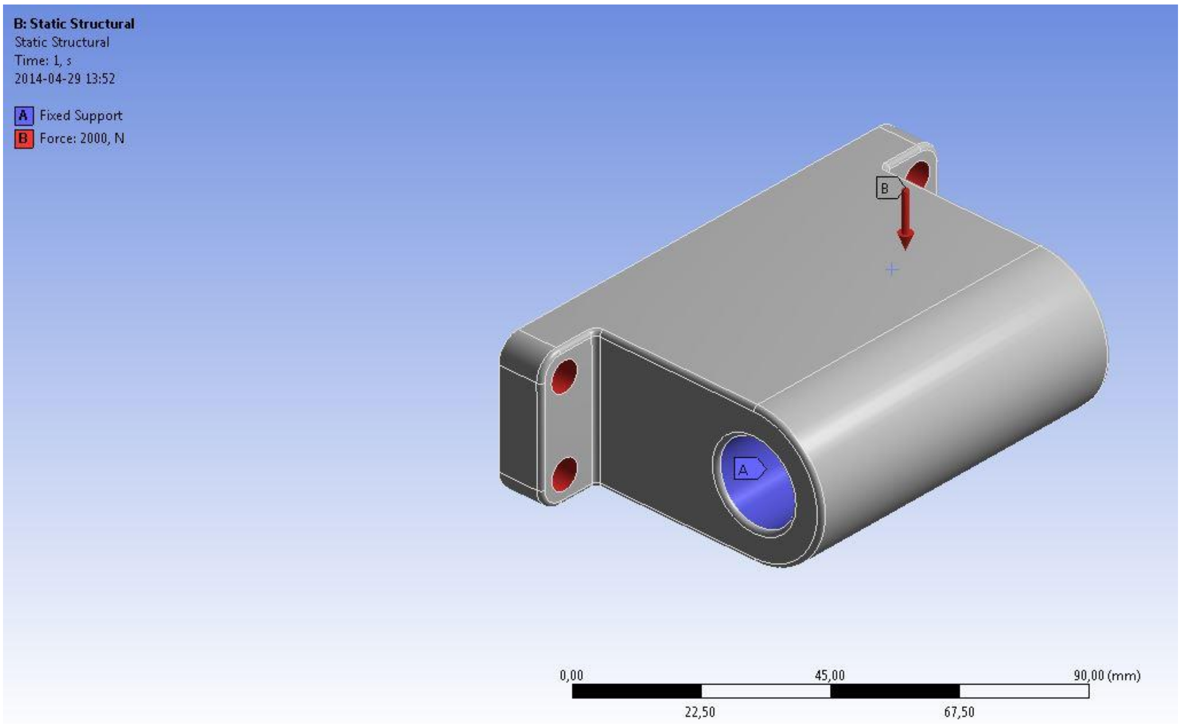

B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2014-04-24 18:38
161,6 Max
143,65
125,7
107,75
89,8
71,851
53,901
35,952
18,002
0,052556 Min
0,00
100,00
200,00 (mm)
50,00
150,00

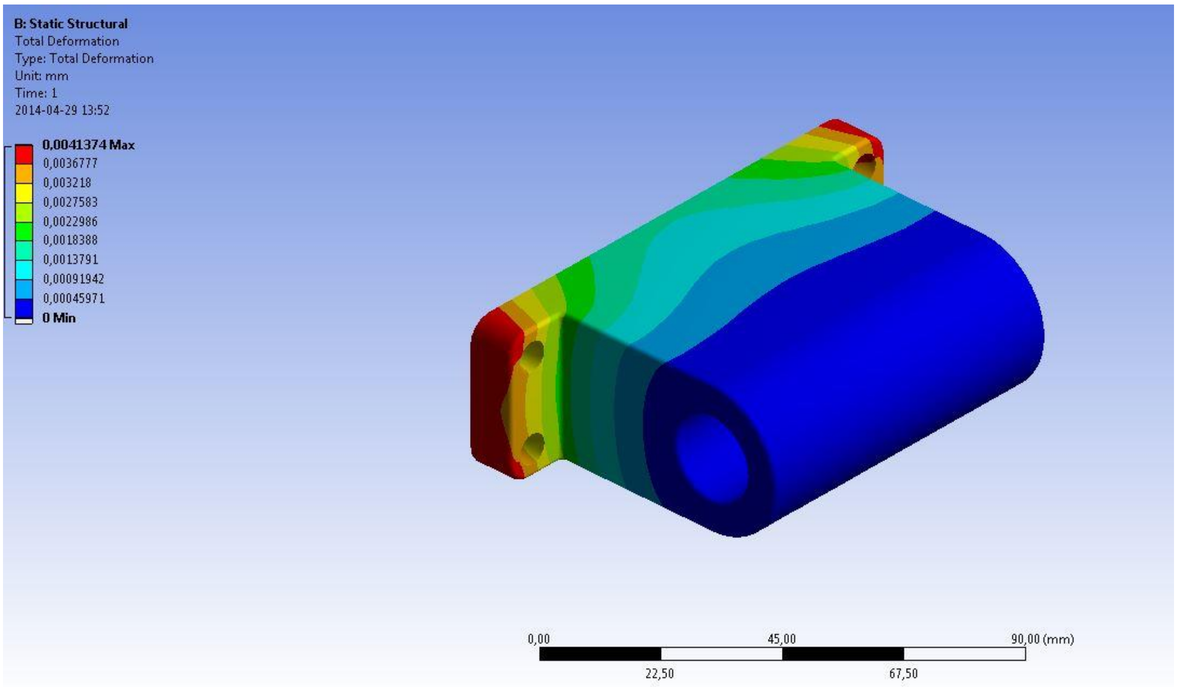

B: Static Structural
Static Structural
Time: 1, s
2014-04-24 16:55
A Fixed Support
B Force: 2200, N
A
B
0,00
35,00
70,00 (mm)
17,50
52,50

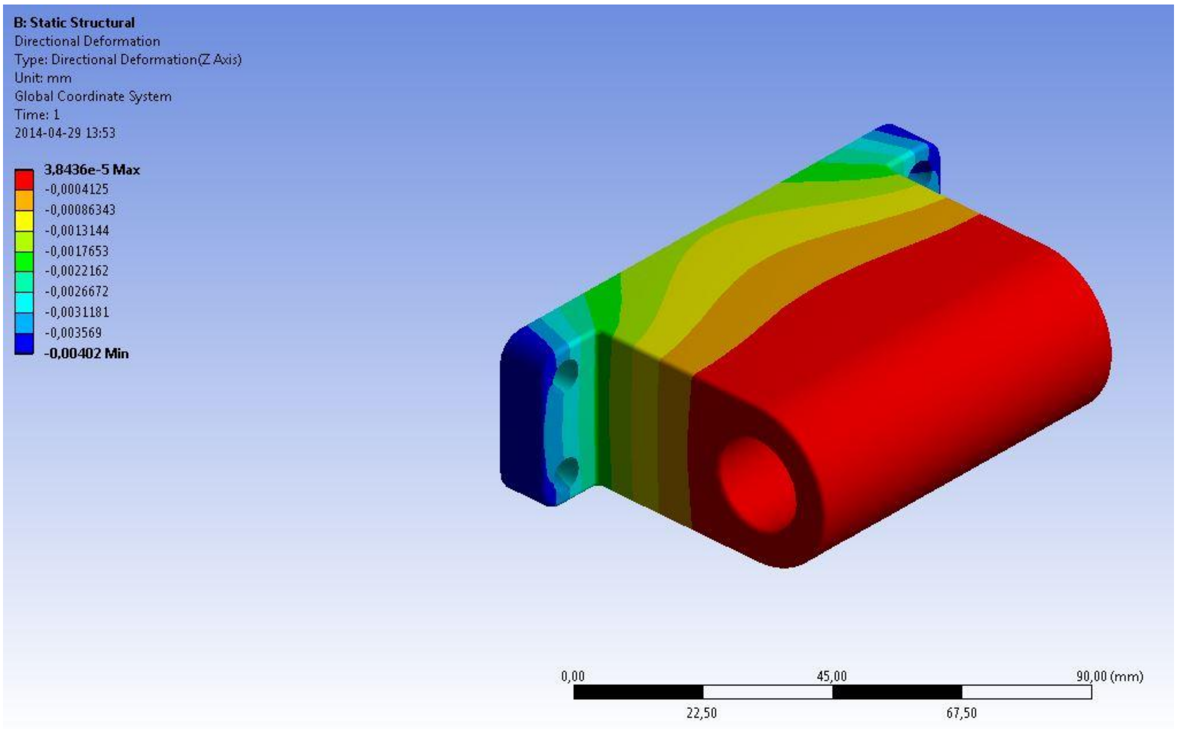

B: Static Structural
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
2014-04-24 16:56
0,0079452 Max
0,0070624
0,0061796
0,0052968
0,004414
0,0035312
0,0026484
0,0017656
0,0008828
0 Min
0,00
35,00
70,00 (mm)
17,50
52,50

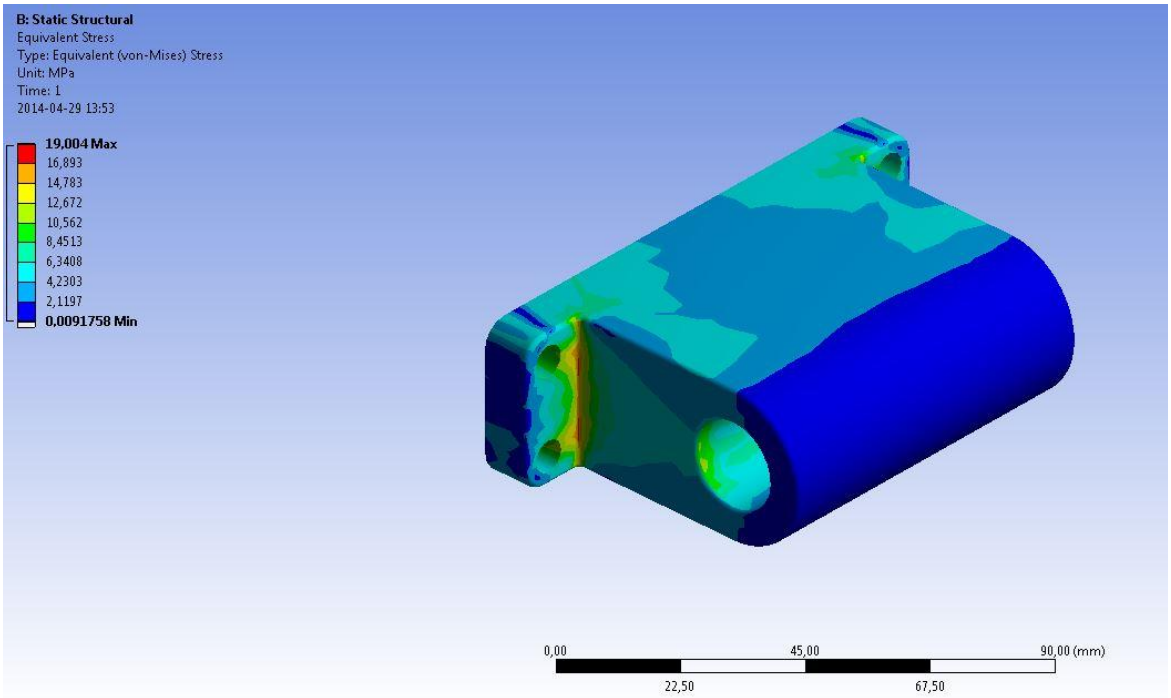

B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2014-04-24 16:57
33,677 Max
29,977
26,277
22,576
18,876
15,176
11,476
7,7757
4,0755
0,37531 Min
0,00
35,00
70,00 (mm)
17,50
52,50

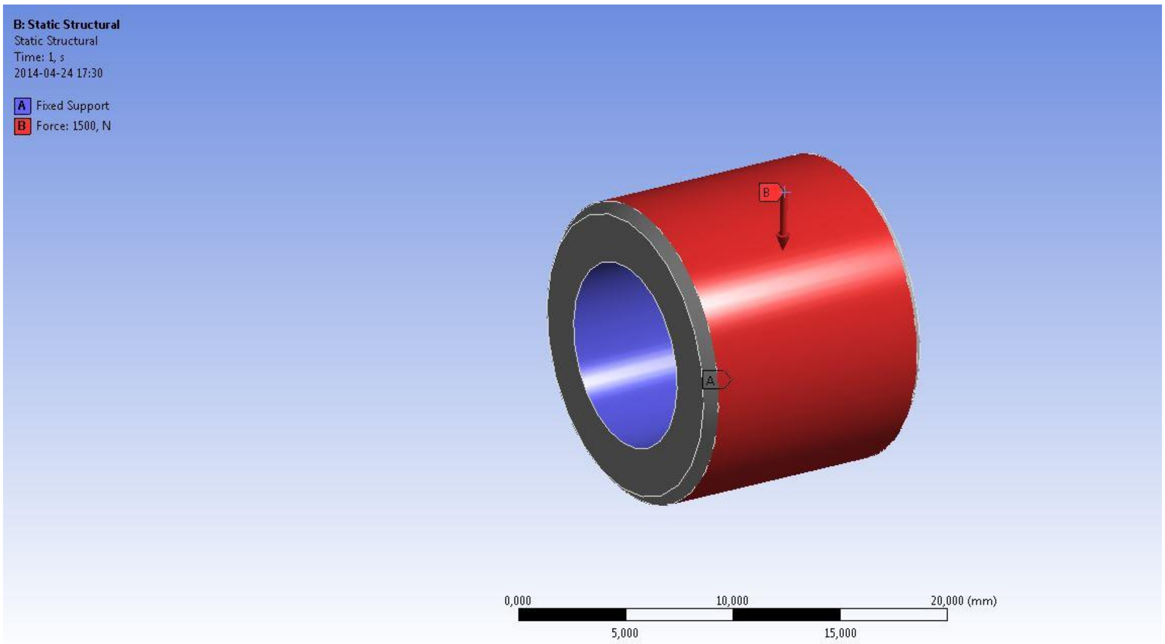

B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2014-04-24 18:20
242,68 Max
215,72
188,75
161,79
134,82
107,86
80,895
53,93
26,965
1,5097e-5 Min
0,00
100,00
200,00 (mm)
50,00
150,00

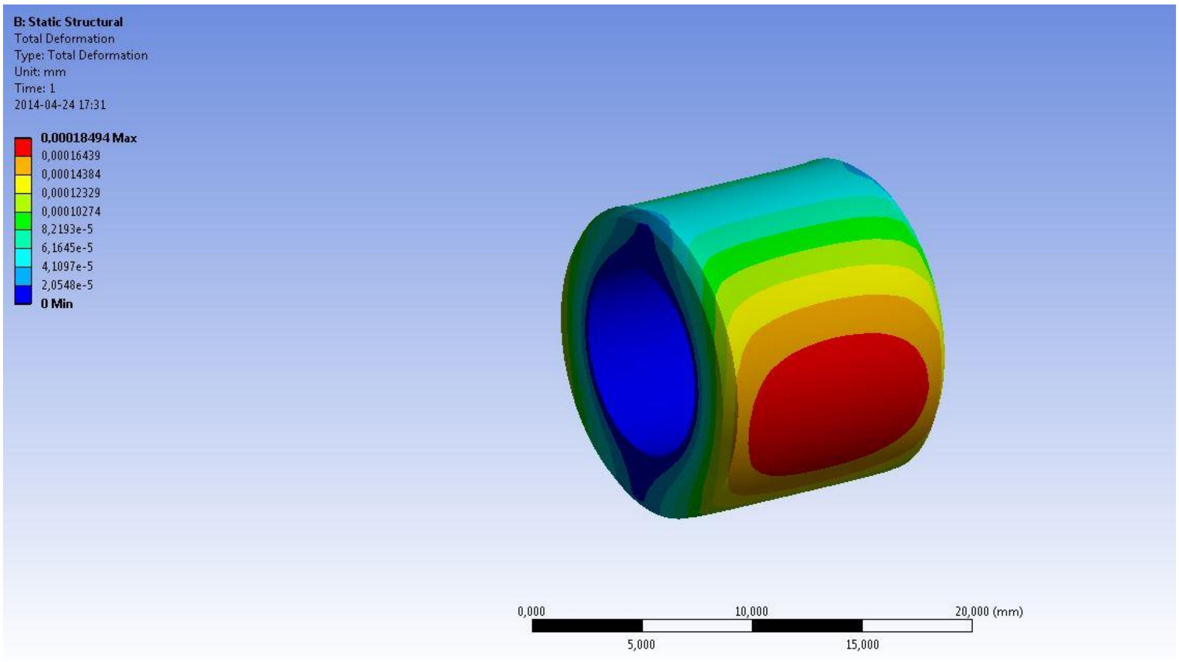

B: Static Structural
Static Structural
Time: 1, s
2014-04-29 13:52
A Fixed Support
B Force: 2000, N
B
A
0,00
45,00
90,00 (mm)
22,50
67,50

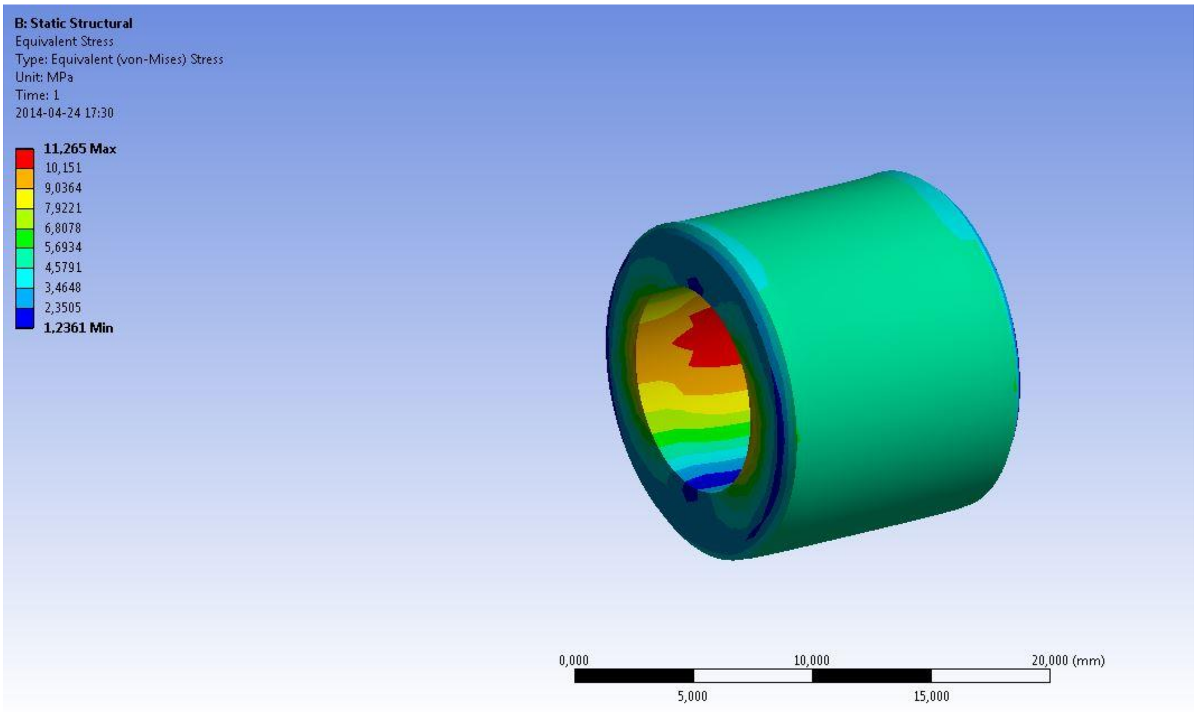

B: Static Structural
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
2014-04-29 13:52
0,0041374 Max
0,0036777
0,003218
0,0027583
0,0022986
0,0018388
0,0013791
0,00091942
0,00045971
0 Min
0,00
45,00
90,00 (mm)
22,50
67,50

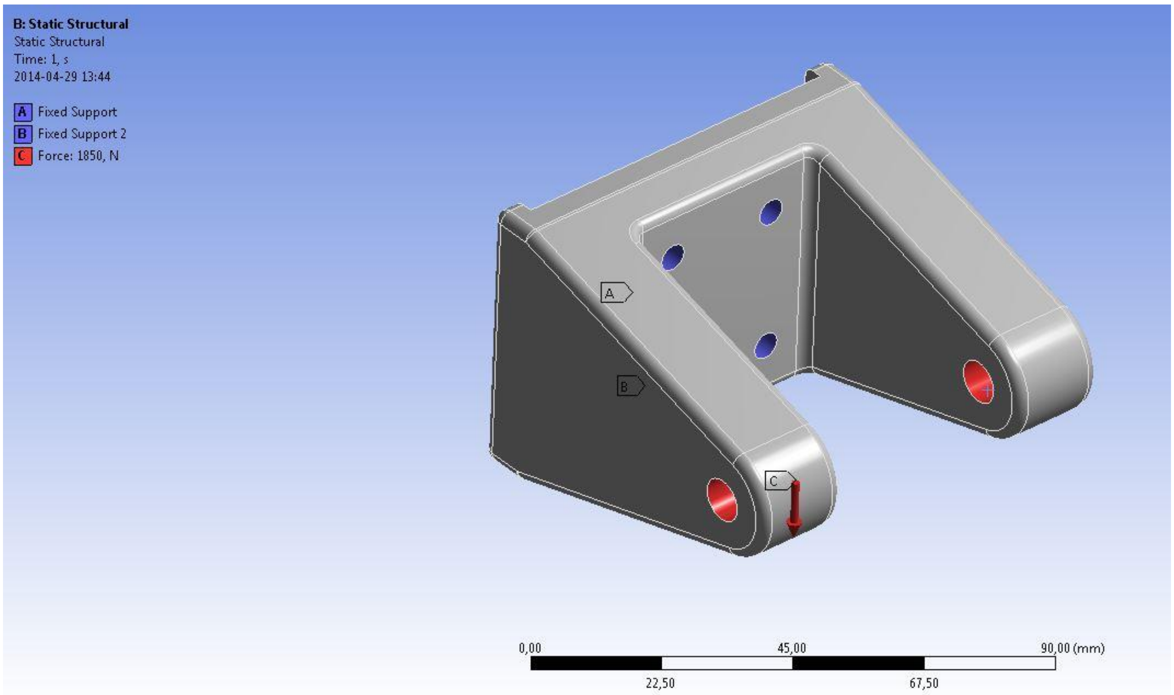

B: Static Structural
Directional Deformation
Type: Directional Deformation(Z Axis)
Unit: mm
Global Coordinate System
Time: 1
2014-04-29 13:53
3,8436e-5 Max
-0,0004125
-0,00086343
-0,0013144
-0,0017653
-0,0022162
-0,0026672
-0,0031181
-0,003569
-0,00402 Min
0,00
45,00
90,00 (mm)
22,50
67,50

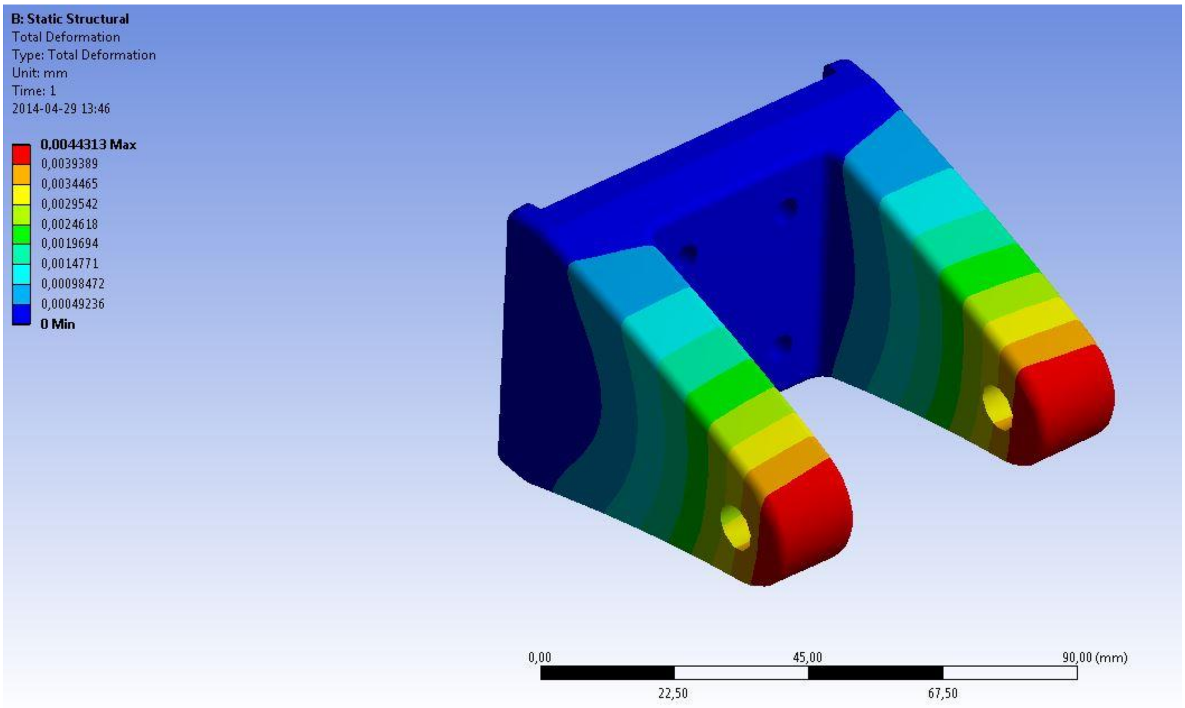

B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2014-04-29 13:53
19,004 Max
16,893
14,783
12,672
10,562
8,4513
6,3408
4,2303
2,1197
0,0091758 Min
0,00
45,00
90,00 (mm)
22,50
67,50

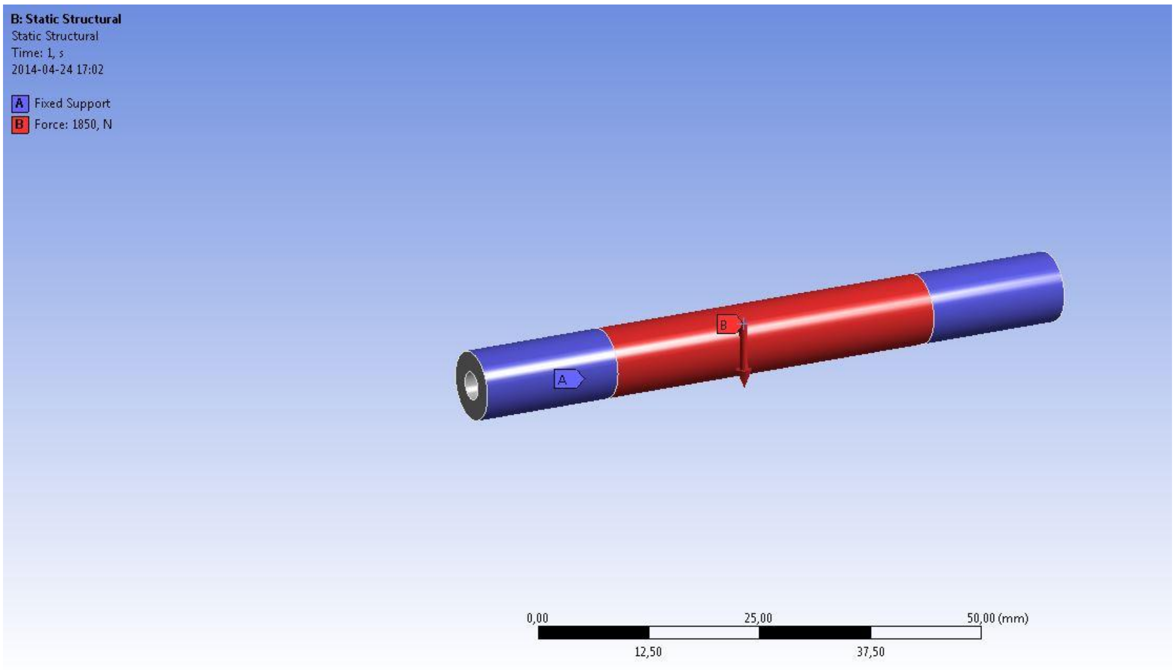

B: Static Structural
Static Structural
Time: 1, s
2014-04-24 17:30
A Fixed Support
B Force: 1500, N
B
A
0,000
10,000
20,000 (mm)
5,000
15,000

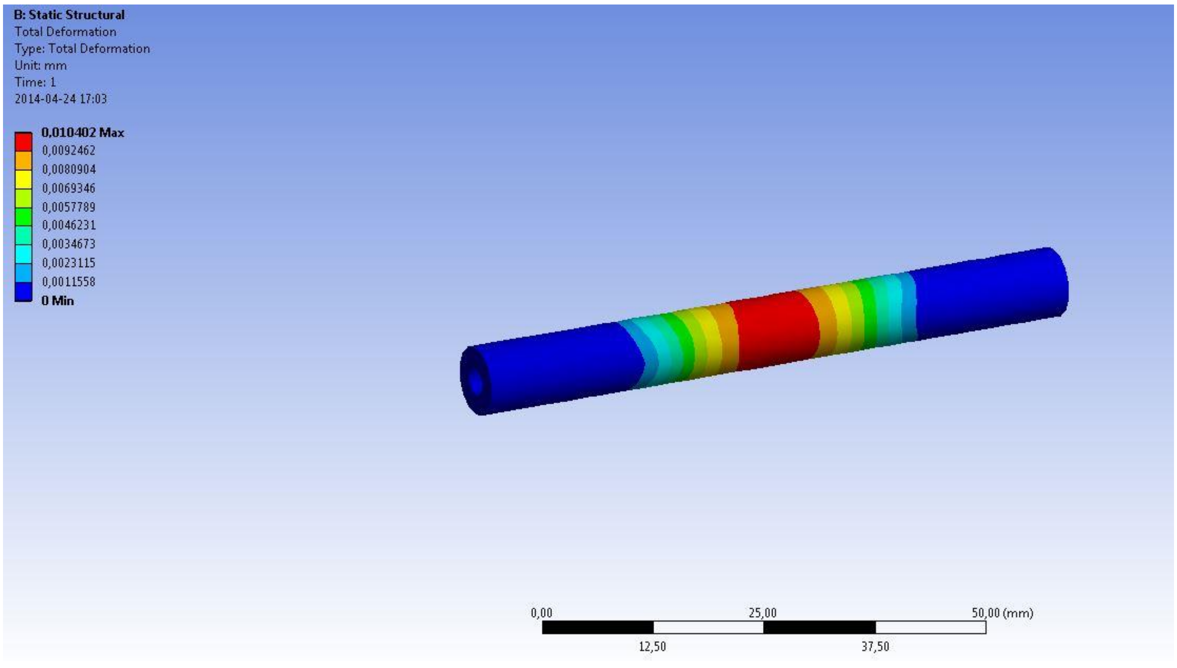

B: Static Structural
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
2014-04-24 17:31
0,00018494 Max
0,00016439
0,00014384
0,00012329
0,00010274
8,2193e-5
6,1645e-5
4,1097e-5
2,0548e-5
0 Min
0,000
10,000
20,000 (mm)
5,000
15,000

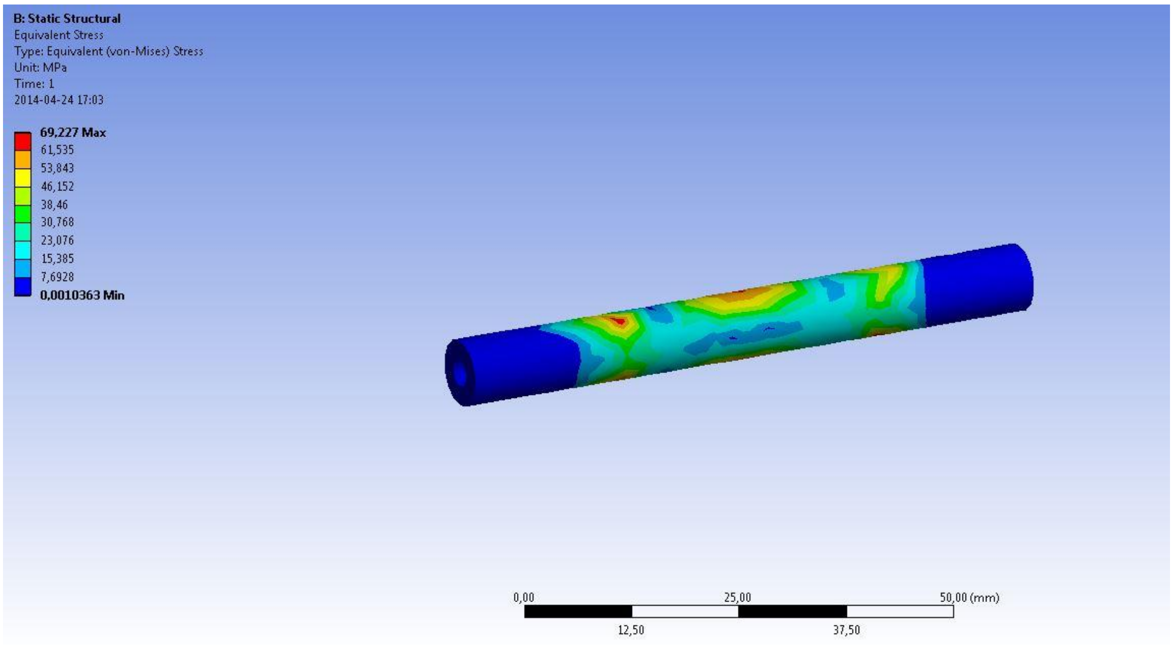

B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2014-04-24 17:30
11,265 Max
10,151
9,0364
7,9221
6,8078
5,6934
4,5791
3,4648
2,3505
1,2361 Min
0,000
10,000
20,000 (mm)
5,000
15,000

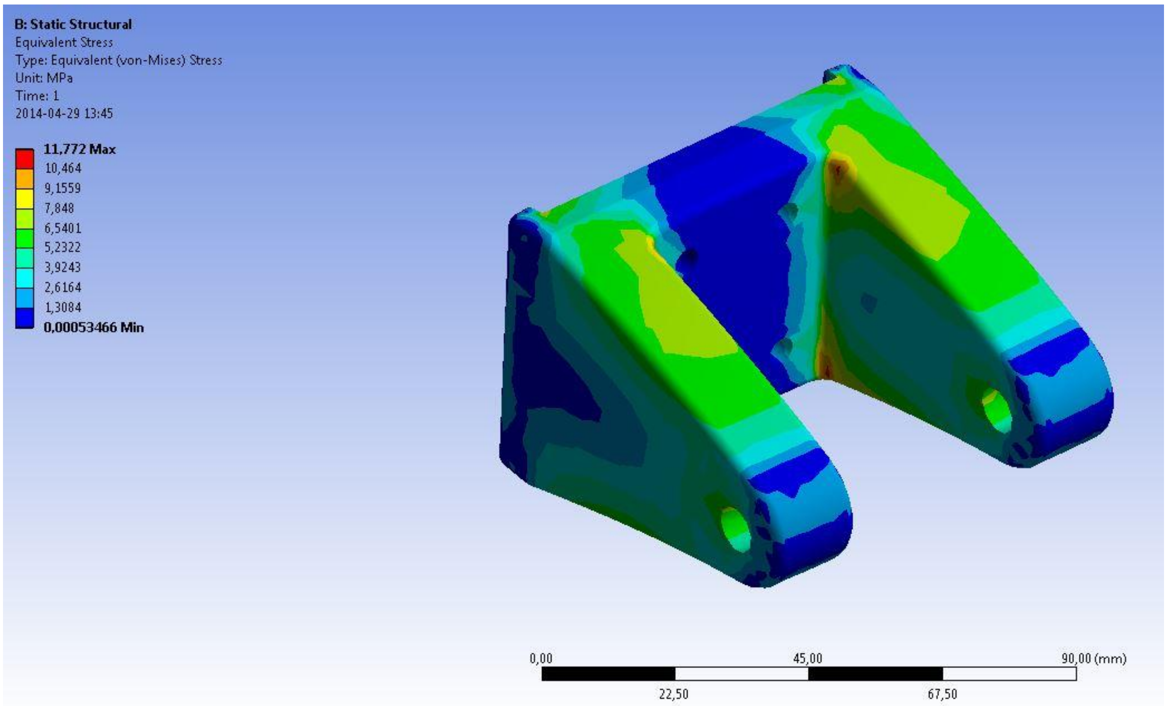

B: Static Structural
Static Structural
Time: 1, s
2014-04-29 13:44
A Fixed Support
B Fixed Support 2
C Force: 1850, N
A
B
C
0,00
45,00
90,00 (mm)
22,50
67,50


B: Static Structural
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
2014-04-29 13:46
0,0044313 Max
0,0039389
0,0034465
0,0029542
0,0024618
0,0019694
0,0014771
0,00098472
0,00049236
0 Min
0,00
45,00
90,00 (mm)
22,50
67,50


B: Static Structural
Static Structural
Time: 1, s
2014-04-24 17:02
A Fixed Support
B Force: 1850, N
A
B
0,00
25,00
50,00 (mm)
12,50
37,50


B: Static Structural
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
2014-04-24 17:03
0,010402 Max
0,0092462
0,0080904
0,0069346
0,0057789
0,0046231
0,0034673
0,0023115
0,0011558
0 Min
0,00
25,00
50,00 (mm)
12,50
37,50


B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2014-04-24 17:03
69,227 Max
61,535
53,843
46,152
38,46
30,768
23,076
15,385
7,6928
0,0010363 Min
0,00
25,00
50,00 (mm)
12,50
37,50


B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
2014-04-29 13:45
11,772 Max
10,464
9,1559
7,848
6,5401
5,2322
3,9243
2,6164
1,3084
0,00053466 Min
0,00
45,00
90,00 (mm)
22,50
67,50