TAMPERE UNIVERSITY
OF APPLIED SCIENCES

PREPARING FOR FORWARDER ROPS - TEST

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ABSTRACT

Tampere University of Applied Sciences
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Preparing for forwarder ROPS -test

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This thesis is based on the preparation of John Deere forwarder for new 33 tonne ROPS –testing. The starting point is to analyze the current 29 tonne ROPS against 33 tonne requirements. The structure is analyzed with finite element method to see if it can survive the requirements as it is. Since this is not very probable the second phase is to make design changes to the structure and/or changes to materials to strengthen the structure for the 33 tonne test. This thesis presents the preparation for the analysis and the design changes need to be made for the forwarder to pass the test. The challenges during process and reliability of the results are also presented.

Forest work is considered to be one of the most dangerous professions in the world in spite the fact that mechanized harvesting has reduced the risk of injuries. Safety is one of the most important aspects in design of new machines. By legislation most developed countries monitor the safety of machines operating in their soil. Machines in Europe are regulated by machinery directive 2006/42/EC. This is the general regulation concerning all machines but there are also other standards, laws and guidelines that need to be fulfilled. Another important standard for forest machines is ISO 8082-1 and ISO 8082-2. These two standards are applied to self-propelled machinery for forestry and they define the performance requirements for rollover protective structures (ROPS).

In case of rollover no part can enter the safe zone of the operator. This is done by adding protective structures to the machine that take the impact and absorb the energy during rollover. The protective structures need to pass ROPS –test that will be done to make sure the structure is adequate. In ROPS –test a load is applied to the protective structure and deflection of the structure is measured. Machine passes the test if the operator safe zone stays intact and specified energy and force levels are reached.

In ROPS –test the structures are deformed permanently since loads during test need to simulate the forces during actual rollover situation. In most cases the weight of the machine needs to be as light as possible due to energy efficiency and fuel consumption requirements and for these reasons the compromise of strong yet light structures is crucial. For this reason careful preparation for the testing is important as the safety factors cannot be exaggerated.

Key words: forwarder, rollover protective structure (ROPS), finite element analysis

TIIVISTELMÄ

Tampereen ammattikorkeakoulu
Konetekniikan koulutus
Tuotekehitys

VAIRINEN NOORA:
Valmistautuminen kuormakoneen ROPS -testiin

Opinnäytetyö 53 sivua, joista liitteitä 3 sivua
Lokakuu 2016

Metsätyötä pidetään yhtenä maailman vaarallisimmista ammateista, vaikka koneellistumisen myötä loukkaantumisien riskit ovatkin vähentyneet. Turvallisuus on tärkeä lähtökohta koneita suunniteltaessa. Lait ja asetukset määräävät rajat, jotka jokaisen koneen ja laitteen tulee täyttää. Erilaiset lait ja asetukset takaavat kunkin maan konekannan turvallisuuden suurimmassa osassa kehittyneitä maita. Euroopassa kaikkia koneita ja laitteita, myös metsäkoneita koskee Konedirektiivi 2006/42/EC. Tämän yleisen koneita koskevan säännöksen lisäksi on muita täydentäviä lakeja ja asetuksia, joiden mukaan toimitaan. Yksi tärkeä turvallisuusstandardi on kaksiosainen ISO 8082 -standardi (osat 1 ja 2). Tämä standardi määrittää itsekulkevien metsäkoneiden turvarakenteiden (ROPS-rakenne) kestävyysvaatimukset.

Jos metsäkone kaatuu, yksikään osa ei saa tunkeutua kuljettajan turva-alueelle. Tämä varmistetaan suunnittelemalla turvarakenteet metsäkoneen ohjaamolle, joiden tarkoitus on ottaa vastaan kaatumisesta ja ympäripyörähtämisestä aiheutuvat iskut ja absorboida pyörimisen aiheuttama energia. Turvarakenteiden tulee läpäistä ROPS-testit, jotka varmistavat rakenteiden keston todellisessa kaatumistapauksessa. ROPS-testissä standardissa määritellyn suuruinen voima kohdistetaan turvarakenteisiin ja rakenteiden siirtymää seurataan. Kone läpäisee testit, kun operaattorin turva-alueelle pysyy koskemattomana ja kun rakenne pystyy absorboimaan riittävän määrän energiaa itseensä.

Opinnäytetyössä tutkittiin John Deeren kuormakoneen kestävyyttä 33 tonnin ROPS –testissä. Työssä lähdettiin liikkeelle nykyisestä kuormakoneesta, joka on hyväksytty 29 tonnin vaatimusten mukaan. Tätä rakennetta analysoitiin elementtimenetelmällä. Koska oli hyvin oletettavaa, ettei nykyinen rakenne sinällään kestä uusia vaatimuksia, varauduttiin myös siihen, että rakennetta joudutaan muokkaamaan. Työn toisessa vaiheessa rakennetta vahvistettiin ja materiaalivalintoja muutettiin, jotta rakenne saataisiin kestämään 33 tonnin testin vaatimukset. Opinnäytetyössä esitetään analysointiprosessi ja uudelleen suunnitellut rakenteet. Suunnittelun haasteet ja analysoinnin ongelmakohdat käydään myös läpi tuloksien ohella.

ROPS-testissä rakenteet muovautuvat pysyvästi, sillä kaatumisen ja pyörimisen aiheuttamat voimat ovat suuria ja testissä käytettyjen voimien tulee simuloida näitä voimia. Koska liikkuvien koneiden massa on kriittinen suure energiatehokkuutta ja polttoaineenkulutusta mitattaessa, ei rakenteita voida ylimitoittaa. Turvarakenteiden suunnittelu on tasapainoilua tarpeeksi kestävien ja kevyiden rakenteiden saavuttamiseksi.

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kuormakone, ROPS, elementtimenetelmä

CONTENTS

ABBREVIATIONS AND TERMS

CTL	Cut-to-length –method
delimbing	removing branches from trees or parts of trees
FEA	Finite element analysis
FEM	Finite element method
forwarder	a self-propelled machine used for transporting logs off forest
TL	Tree length –method; tree is felled and delimbed at the stump and then transported off forest
harvester	A self-propelled machine that falls and processes trees at the stump
ROPS	Rollover protective structure

1 INTRODUCTION

The objective of this thesis is to analyse forwarder 1910G (picture 1) levelling structure to see if it fulfils the 33 tonne ROPS –test requirements. If the structure does not meet the requirements in the current form suggestions to the new materials are given and/or modifications to part geometries are made. In this thesis the structure is analysed using finite element method (FEM). The ROPS –test is done later so the results of the test are not part of this thesis.

The current version ROPS is approved against 29 tonnes but the need is to get forwarder ROPS for 33 tonnes in the future. The idea is to analyse the current structure to find out what needs to be changed in order for the future demands to be met. The assumption is that the current structure cannot withstand the loads and some modifications will need to be done. In this thesis these modifications are also designed and analysed.

PICTURE 1. Deere 1910G forwarder.

In the ROPS test heavy loads are applied to forest machine to simulate the rollover situation of the machine. As the test is destructive and expensive it is important to analyse the structures thoroughly beforehand to find the weak parts and to make corrective actions. This analysis is a continuation to the forest harvester ROPS analysis that is currently on going at John Deere. The analyses of other parts of ROPS (cabin etc.) are not part of this thesis.

2 DEERE & COMPANY

2.1 History and product lines

Since its early days John Deere's mission has been to provide machinery and services "to those who are linked to the land – farmers and ranchers, landowners, builders, and loggers". (Deere, 2016) Deere is the world's leading manufacturer of agricultural and forest machinery (picture 2). With the machines Deere also provides precision Ag technology to help operators and land owners to more efficiently plan the jobs from start to finish. The equipment information, production data, etc. can be followed online to assure machines work with the highest accuracy and receive the best maintenance possible.

PICTURE 2. Deere has long history of manufacturing agricultural machinery (Deere, 2016)

In addition to agricultural and forestry machinery, Deere also manufactures earthmoving and landscaping machinery and engines. From small green lawn mowers to supersize yellow articulated dump trucks Deere offers vast variety of machines to make life easier for operators all over the world.

John Deere was a pioneer blacksmith who developed the first commercially successful self-cleaning steel plow. In 1837, he founded the company that still bears his name.

Deere & Company is one of the oldest industrial companies in the United States. Today, John Deere does business around the world and employs nearly 60,000 people in over 30 countries. The headquarters is located in the United States, in Moline Illinoi. For the fiscal year 2015 the net sales was $1,94billion and revenue $28,9billion. (Deere, 2016)

Deere's four core values handed down by founder John Deere are integrity, quality, commitment, and innovation. These values are adapted to the customer relationships, products and services, and the safety of all Deere employees.

2.2 John Deere Forestry Oy

John Deere Forestry Oy develops and manufactures cut-to-length forest harvesters (picture 3) and forwarders in Joensuu factory. In 1972 Rauma-Repola founded the Joensuu factory that started making forwarders in 1973. (Konttinen, 1997) Over the years the name of the forest machines and factory have changed several times but in 1990 they became Timberjack's. John Deere purchased the factory and the production in 2000 and changed the name to John Deere Forestry Oy five years later. (Eskola, 2016)

PICTURE 3. Deere 1470G cut-to-length harvester (Deere, 2016)

Manufacturing of John Deere harvesters is all done in Joensuu, but the head quarter, research and development and finance is located in Tampere (picture 4).

PICTURE 4. John Deere Forestry Oy Tampere factory. Picture was taken by Kuusela P. (Google, 2016)

John Deere has wide range of cut-to-length harvesters and forwarders sizing from small forwarders and thinning harvesters to large final felling forest machines.

3 FOREST MACHINES

3.1 Wood harvesting

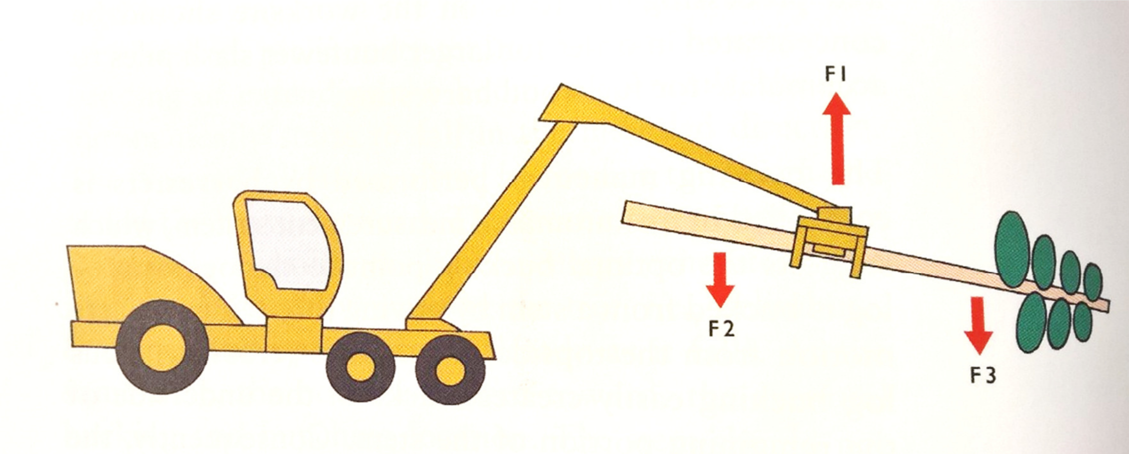
Wood can be harvested in many ways. Commonly the methods are divided into three main categories; full tree harvesting, tree length harvesting (TL) and cut-to-length harvesting (CTL). In full tree harvesting tree is felled and then dragged out from forest as it is. In the tree length method the tree is cut and delimbed (ISO 6814) at site and then dragged out from the forest. In CTL method the three is felled, delimbed and cut to logs at the stump area. The processing can be made manually with chainsaw or mechanically with harvester. The cut logs are then transported out from the forest using forwarder. In the final stage of supply chain trucks take logs to sawmill. (Picture 5) Cut logs are often also measured during cutting to optimize the ratio of different type of logs and pulps and to discard rot at site.

PICTURE 5. Example of supply chain from forest to sawmill. (Kokkarinen, 2012)

TL and full tree methods are still main methods in US since all sawmills are built to take in full trunks but in Europe and especially in Scandinavia the CTL method is dominant. CTL method is more environmentally friendly, versatile and safe method that provides end products of more consistent and higher quality than mechanized TL method. (LeDoux, 2001) John Deere manufactures full tree forest machines in United States and CTL forest machines in Finland.

3.2 Cut-to-length harvesters

Harvester is a self-propelled machine on wheels or tracks. Harvesters have cutting head attached to a boom which is attached to the harvester (picture 6).



PICTURE 6. Harvester with boom and harvester head (Uusitalo, 2010)

Harvester head is equipped with feeding and cutting systems. Feeding system can be roller based in which rotating feeding rollers transport the tree or it can be stroke based in which the harvester head moves like looper and pushes the tree towards cutting system. The more dominant is roller based and the harvester is called roller harvester compared to stroke harvester. The cutting system is most commonly saw blade with chain but other type of cutting devices are also used in some cases. The front and back knives take care of delimbing.

In automatized systems the harvester crabs the tree with harvester head from butt end fells it and calculates automatically what type of logs to cut. The machine operator can insert a list of log types of which are needed at the saw mill, and according to that list the harvester automatically decides the best combination of logs that can be achieved from that particular tree. Harvester operator needs to make the decision if the tree is not good quality (for example if it is rotten) or some parts of it need to be rejected. In less developed harvesters the cutting decisions are made by the operator.

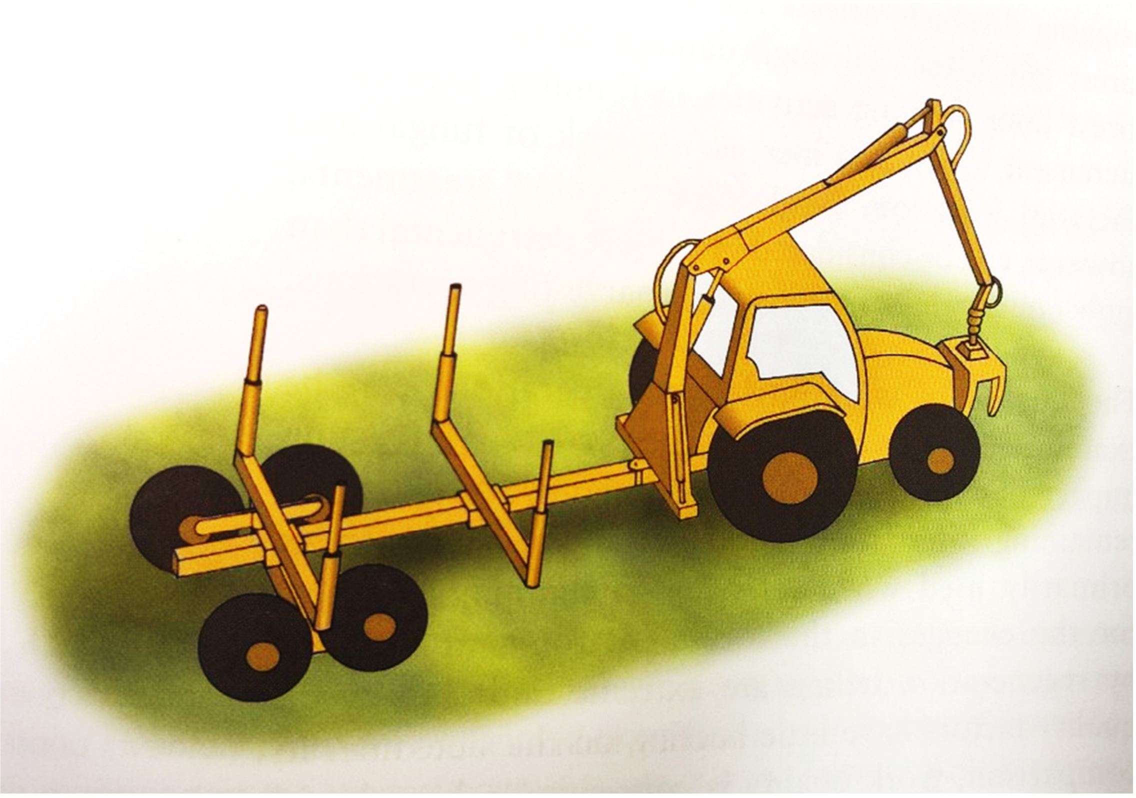
Operator moves the logs to a correct position so the logs piles can be easily collected by the forwarder operator (picture 7). Especially in Scandinavia the cutting process is very optimised to achieve the best possible efficiency with the least cost.

PICTURE 7. Harvester and forwarder at work (Uusitalo, 2010)

3.3 Forwarders

Forwarder is self-propelled machine used for transporting logs off forest. (Helms, 1998) After the logs are cut the forwarder picks up the logs to its log bunk using grapple. Grapple is attached to the boom that is attached to the forwarder (picture 8). The bunk is situated directly at the top of the rear frame. The bunk space is limited from the sides by stakes that keep the load from falling off. The stakes can be adjustable by height and/or width to accommodate different size of loads. (Uusitalo, 2010)

The forwarder cabin can be fixed or rotating and/or moving. Fixed cabin is durable and easy to manufacture and maintain, but it is not very ergonomic or efficient. Nowadays many forwarders (and harvesters) have rotating and levelling cabin structures to allow the cabin to face where needed. During driving the cabin is facing forward but during loading and unloading the cabin is facing backwards so the driver doesn't have to bend their neck in order to see what they are doing. The cabin can also be set to follow the boom movements. The levelling cab is balancing the cabin in bumpy terrain to keep the operator in ergonomically best position. Especially when working in a hilly landscape the levelling brings huge benefits compared to fixed cab since the operator can work in a position similar to flat landscape. The levelling and moving cab brings more moving parts and technology to the forwarder thus making the machine more expensive and more difficult to maintain and manufacture but the benefits still exceed the downsides in most cases.



PICTURE 8. Forwarder (Uusitalo, 2010)

After the logs are brought out from forest the trucks take them to sawmill. Forwarder operators separate different type of logs (log, pulp, energy wood etc.) to different piles so the truck driver knows what logs to take where.

3.4 Combi machines

There are also combi machines (i.e. combined harvester-forwarders) that are capable of carrying out both tree processing and transportation of logs to the roadside. (Nurminen et. al, 2006) They are more expensive than harvesters and forwarders individually but cheaper than buying both machines. Combi machines are compromise of two different machines so they might not be as good in harvesting as harvester or they cannot carry as high load as big forwarder but they have their advantages. In some cases they can be very good option i.e. there is only one operator who can do two men job with combi machine or one machine can be more financially achievable. Combi machines can be converted from forwarders by adding harvester head at the end of forwarder boom. (Picture 9)

PICTURE 9. Forwarder converted into combi machine (Nisula)

4 ROLL OVER PROTECTIVE STRUCTURES

4.1 History

Forest work is considered to be one of the most dangerous professions in the world although the increase of mechanization in harvesting operations has reduced the number of accidents. (Uusitalo 2010). In Finland, the mechanised harvesting started to play role from 1950's. (Kanninen, 1999) There are many types of risks in mechanised harvesting. The harvester head has feeding rollers moving the tree more than five meters per second, the chain saw and delimbing knives present severe risk if approached during operation. The tree can whoosh into the cabin from the harvester head or a tree can fall on the harvester, the harvester can fall down if operated in steep slopes, etc.

Nowadays the safety is one of the most important aspects in machine manufacturing and design work. Good ergonomics and safety are demanded by operators and by law and regulations (figure 1). (Kämäräinen, 2002) In Europe the machinery directive 2006/42/EC regulates the safety features the machines need to have. Also additional safety guidelines and regulations are applied to specific machines. Similar laws and regulations are also in use in many countries (for example OSHA 1910.212 in United States).

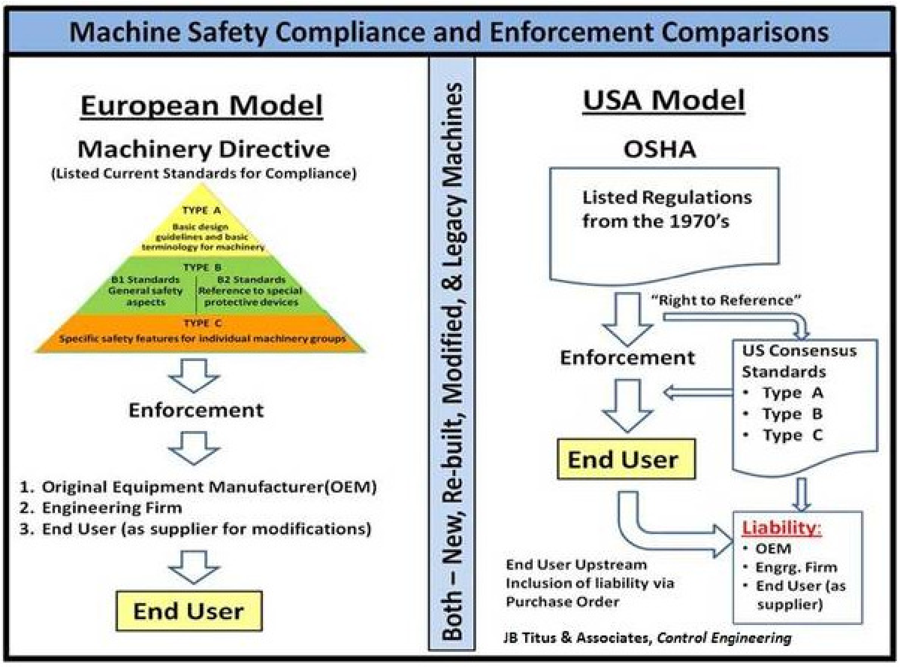


FIGURE 1. Machine safety procedures in Europe and in United States. (Titus, 2013)

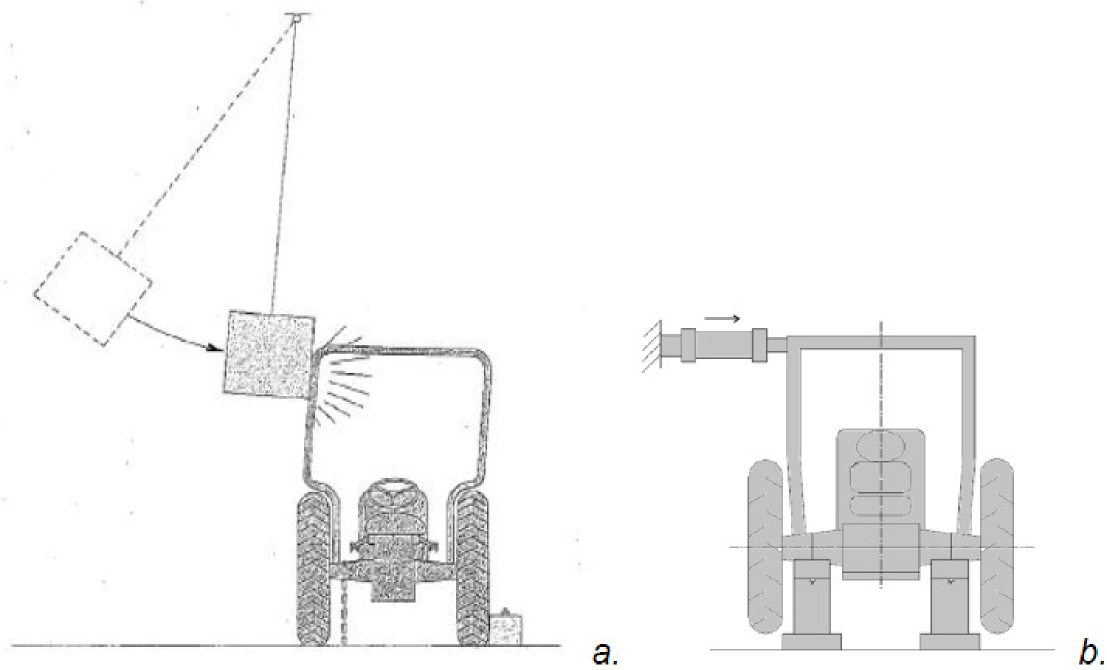
Rollover protective structure (ROPS) is a cab or frame that provides a safe environment for the operator even in case of rollover. The rollover structure prevents the operator from being crushed under the vehicle during roll over if the seatbelt is worn. The seatbelt keeps the operator within the safe zone of the ROPS. The ROPS need to be both stiff and flexible; stiff enough to offer protection and flexible to absorb most of the impact energy during rollover situation.

From 1920s the agricultural tractor overturn deaths has been identified as a big problem. (Loringer & Meyer, 2008) In 1959 Sweden became the first country to make legislation for ROPS structures. And in 1965 the rollover protective structures became mandatory in all tractors operating in Sweden. (AFS 2004/6) In the 1960s manufacturers in United States started to offer rollover protective structures as an option and in 1976 United States Occupational Safety and Health Administration gave a regulation stating that all agricultural employers need to equip all employee-operated tractors with rollover protective structures. But since the regulation has facilitations concerning family members and small farm sizes it only affects small part of operators in the U.S. In 1985 manufacturers in the United States started to offer the rollover protective structure as standard equipment on all new machines but since there are still old tractors in use the retrofit of ROPS are also done. (Loringer & Meyer, 2008) As the rollover protective structure has been mandatory for several years now in most developed countries the fatalities in tractor overturn situations have decreased significantly. (Murphy et. al. 2010)

Due to the similar nature of operating agriculture and forest machines the rollover protective structures are mandatory in forest machines also. The international standards concerning self-propelled machinery for forestry are ISO 8082-1:2009 (E) and ISO 8082-2:2011 (E). The former is the general form of the standard and the later concerns machines having a rotating platform with a cab and boom on the platform.

4.2 ROPS –test

ROPS –test simulates the rollover situation of a vehicle or machine. In ROPS –test a force is applied to the top parts of tractor or other vehicle. In picture 10 part a) shows dynamic force test and part b) shows static force tests. These tests require the structure to withstand certain loading without any part of tractor or test apparatus entering the driver’s safe zone and a level of energy to be absorbed by the structure based on the tractor reference mass. (Franceschetti et. al. 2014)



PICTURE 10. ROPS-test (Franceschetti et. al. 2014)

The ROPS -test can be either a laboratory test or a field test. It can also be either dynamic or static. (OSHA 1926.1002, 2016) The dynamic test is an impact loading test. A pendulum blog is impacted against the ROPS from specific height. The static rollover protective testing involves three different static load tests; namely lateral, vertical and longitudinal loading.

In static test loads are applied to the side, upper parts and rear of the protective structure. (Fern, 2011) All three phases are applied to the same structure one after another in this sequence and the structure has to survive all of them. No parts may intrude into to the driver’s clearance zone in order for the machine to pass the test. (Franceschetti et. al. 2014) The loadings are determined in the tables 1 and 2 (ISO 8082:1 and 8082:2).

TABLE 1. Static forces in ROPS –test for machines with rotating cab

Performance requirement	Energy/force
Lateral load energy, U_s (J)	13 000 $(m/10\ 000)^{1,25}$
Lateral load force, F_s (N)	50 000 $(m/10\ 000)^{1,2}$
Vertical load force, F_v (N)	19,61m
Longitudinal load energy, U_f (J)	4 300 $(m/10\ 000)^{1,25}$

TABLE 2: The longitudinal force in ROPS –test

Machine type	Force F N
Wheeled forestry machines [a]	$F = 48\ 000\left(\frac{m}{10\ 000}\right)^{1,2}$
Tracked forestry machines	$F = 56\ 000\left(\frac{m}{10\ 000}\right)^{1,2}$
[a] Includes machines where tracks can be mounted on rubber tyres as optional equipment.	

The “m” in the table is the machine mass expressed in kilograms

A complete machine is not required in the test situation but the evaluated structure needs to represent the structural configuration of an operating situation. In static test the load is applied to the ROPS so slowly that the loading can be considered static. This means that the speed is < 5 mm/s. The loading is continued until the ROPS has achieved both the force and the energy requirements. The energy absorbed is calculated from the deflection of the ROPS according to equation

$$U = \frac{\Delta_1 F_1}{2} + (\Delta_2 - \Delta_1)\frac{F_1 + F_2}{2} + \cdots + (\Delta_N - \Delta_{N-1})\frac{F_{N-1} + F_N}{2}$$

where U is the absorbed energy, Δ is the deflection and F is the force applied (ISO 8082:2, 2011). The force – deflection –curve can be seen in figure 2.

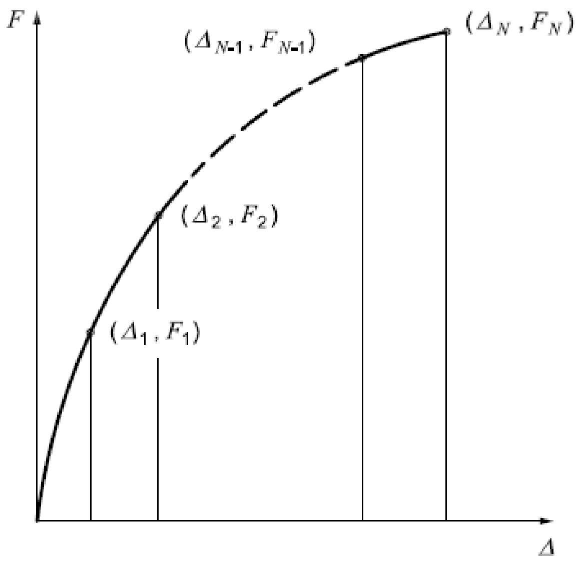


FIGURE 2. Force – deflection curve for lateral loading (ISO 8082:2, 2011)

After lateral loading the vertical load is applied perpendicular to the longitudinal centreline of the ROPS using a 250 mm wide beam until the force level is reached. The structure needs to support this load for a period of 5 min or until any deformation has stopped.

Finally, the longitudinal force is applied horizontally, parallel to the original longitudinal centreline of the machine assuming that previous stages of the test have changed the shape of the ROPS. The loading will continue until longitudinal energy requirement is achieved.

The temperature in test is at or below -18 °C or in some cases at higher temperatures. With higher temperatures the steel needs to have undergone for example Charpy V-notch impact test and all the nuts and bolts need to be appropriate property class in accordance with ISO 898-1 and ISO 898-2. (ISO 8082-2, 2011)

5 MATERIAL PROPERTIES OF STEEL

5.1 Linear elasticity

If a stress below material yield strength is induced to steel the material can fully recover its original shape upon unloading. This happens because the chemical bonds between atoms stretch but not break. When the loading is removed the atoms return to their original place. (Koivisto, 2008) This is called linear elastic behaviour. Linear elastic behaviour of metals is most commonly described by the stress-strain relationship of Hooke's Law. In Hooke's Law the stress (σ) and strain (ε) are connected to each other via Young's Modulus (E) that describes the stiffness of the material

$$\sigma = E\varepsilon$$

Young's Modulus is the slope of the elastic part of the stress – strain – curve.

5.2 Plasticity

When the stress exceeds yield strength in the material plastic deformation starts to occur. Material doesn't obey the Hooke's law from that point forward and deformation becomes permanent. The material stretches if the load continues to increase and at some point the material will reach its tensile strength and break down. This is the maximum stress that can be in the specific material. Example of material specific stress-strain curve is presented in figure 4. Area a) is the elastic area in which the part will return to its original shape after the load is removed. This part obeys Hooke's Law. When the load is still increased the material will have permanent deformations and due to this the material will also harden. This is shown as area b), as the load increases the material stretches but at the same time more load has to be brought to the material to stretch it even more. Only after tensile strength (R_m in the figure 4) the hardening will stop and the material will stretch with less force and finally break down as shown in area c). During hardening the material loses its ductility.

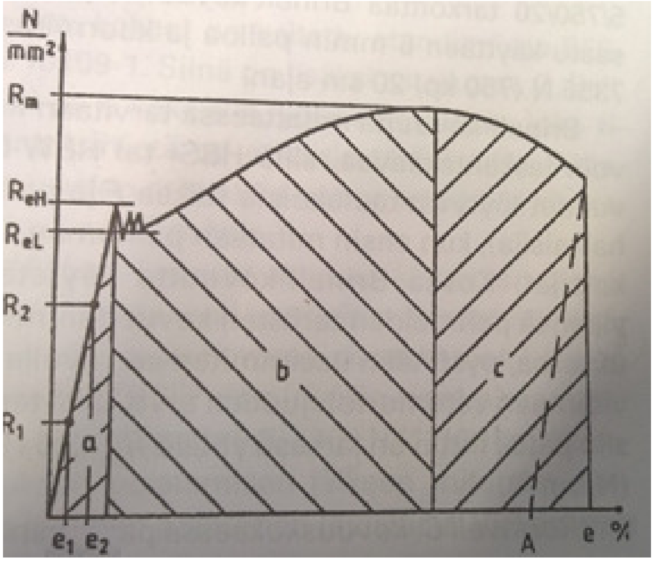


FIGURE 4. Stress-strain curve (Koivisto et al, 2008)

5.3 Bilinear material behaviour

As simplification the material properties can be described as bilinear. This means the material has linear properties before yield strength and linear after yield strength but with different slope. The slope after yield strength is called Tangent Modulus (E_t). An example of bilinear material's stress-strain curve is shown in figure 5. In reality the Tangent Modulus describes the real behaviour after yield strength by being the tangent of the curve in each individual point of the curve.

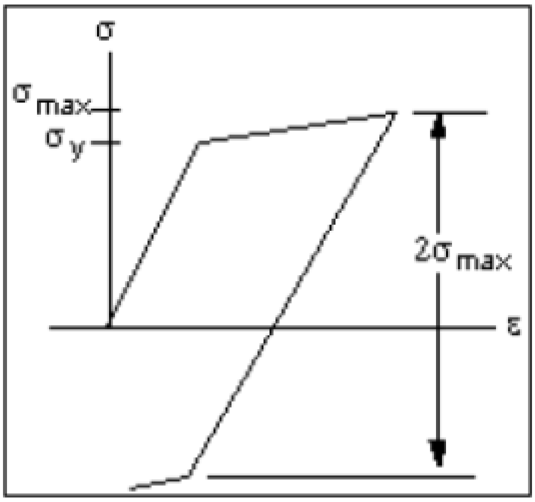


FIGURE 5. Ideal bilinear hardening stress –strain –curve (Ranganathan et. al)

6 FINITE ELEMENT ANALYSIS OF FORWARDER LEVELLING STRUCTURE

6.1 Levelling structure

The structure that was analysed for this thesis is the levelling structure. This contains structures between cabin rotating unit and wheels of Deere forwarder 1910G. The levelling structure of Deere 1110E forwarder is circled part in figure 6 and levelling structure of 1910G is shown in picture 12.

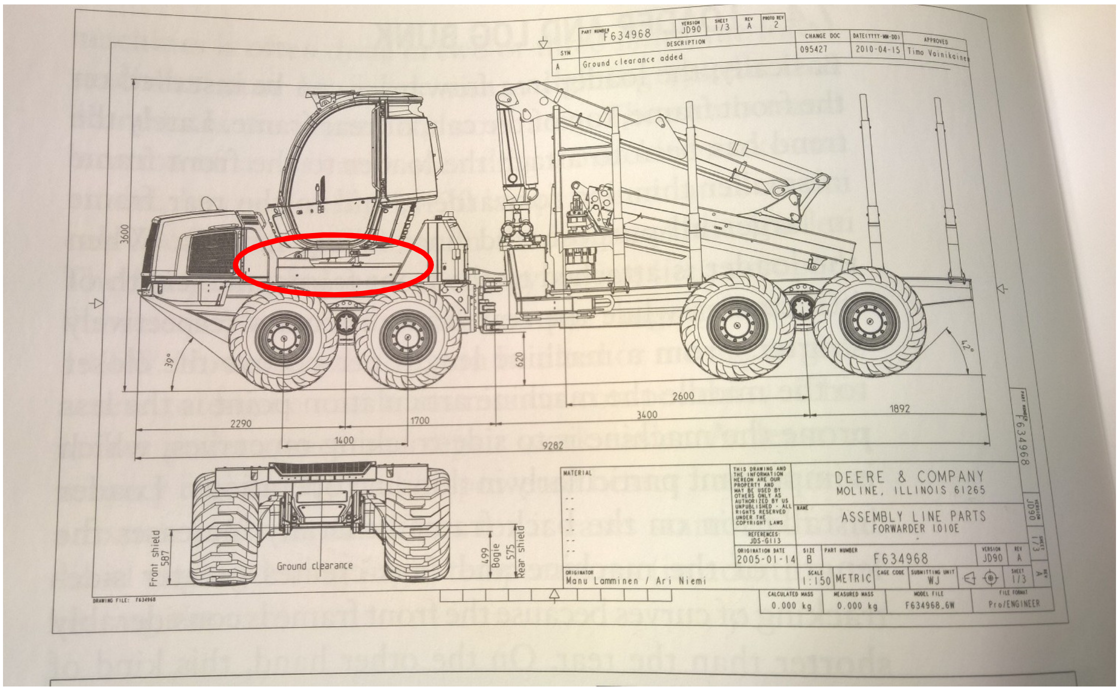
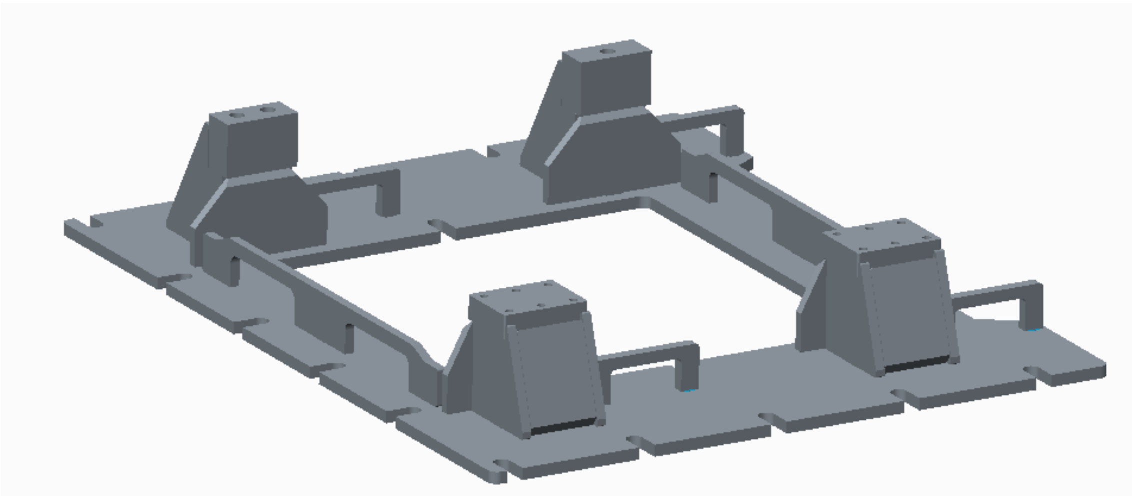


FIGURE 6. Deere 1110E forwarder (Uusitalo, 2010)

PICTURE 12. Levelling structure of 1910G.

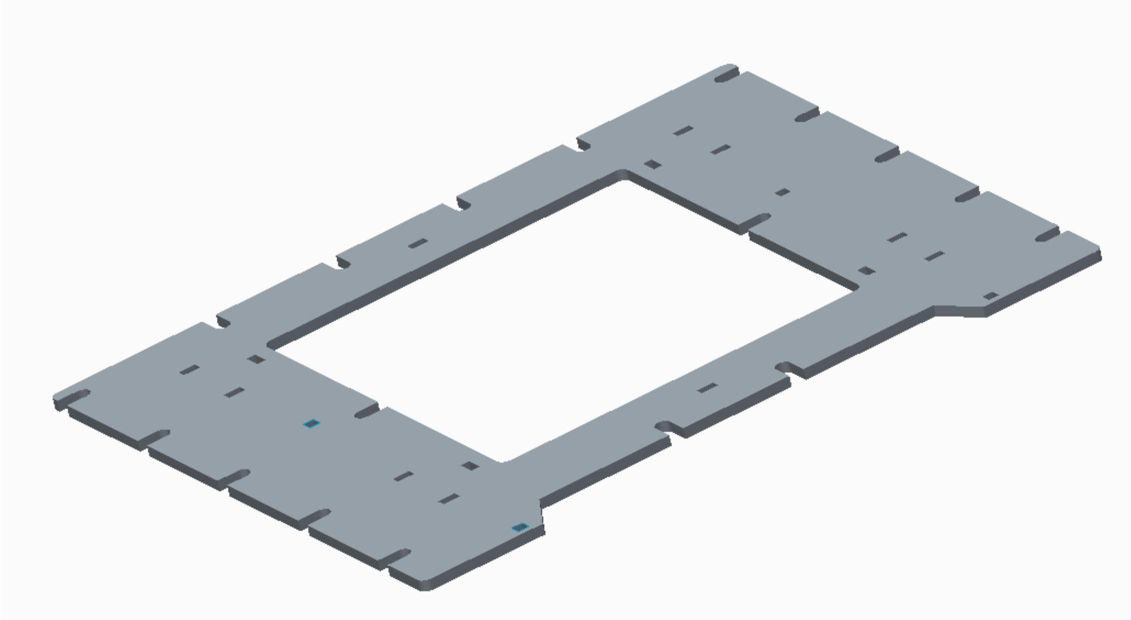
6.2 ROPS –test bench

The ROPS –test is done in laboratory conditions and the cabin and levelling structures are attached to a test bench instead of front frame. The test bench was modelled using Creo parametric and is seen in picture 13. The test bench was designed using the old test bench as basis for the structure. The test bench is used in the actual test instead of the real front frame since this is economically better solution. The structure needs to be flexible yet durable and simulate the front frame of forest machine as good as possible. The flexibility will ensure the structure absorbs enough of the energy in the test situation but it cannot break during the test.



PICTURE 13. Test bench for levelling structure

The test bench will be welded from structural steel plates and the four fixing stands remain hollow to add the flexibility. The base plate will have locating marks for the vertical plates to help the welding process (picture 14). The handles are for lifting the test bench with forklift.

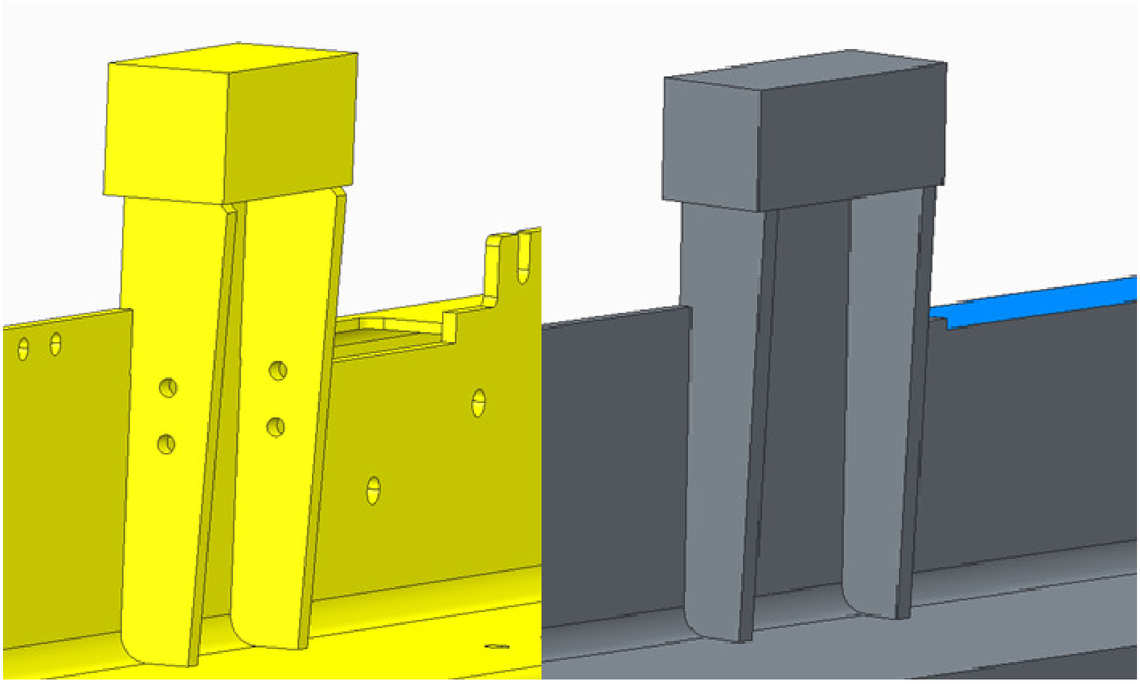


PICTURE 14. Base plate of ROPS –test bench.

6.3 Preparing model for analysis

The analysis was carried out using finite element method. Since the structure is complicated sheet metal structure the first thing was to simplify it for ANSYS –finite element analysis software. For ANSYS all unnecessary small holes and irregularities need to be removed otherwise good quality mesh cannot be generated to the model or the analysis will take too much time as small elements need to be formed to edges and corners. These irregularities have only local influence on behaviour so the removal is justified. (Piscan, 2010) Careful consideration needs to be done not to remove too many features in order for the structure to be as realistic as needed but at same time simple enough for the analysis to succeed.

The simplification was done using Creo Parametric modelling tools the same tool that was originally used to create the model. First a shrinkwrap –model was created from the assemblies that were too complicated to use as they are. Shrinkwrap is a simplified part created out of an assembly. All small holes were then filled and parts were put into contact even if in reality they are slightly apart (picture 15).



PICTURE 15. a) Part of original structure and b) structure after simplifications

This simplification also simulates the fact that most of the small gaps between edges are in reality also filled with welds or with screws and pins.

After the complex structures were simplified a new assembly was created using these simplified parts. The assembly was then exported to ANSYS.

6.4 Materials

The loads applied to the structure in ROPS –tests are extreme. These types of loads are not present during normal operation of the machine. Therefore the structure will be damaged permanently during the test and the steel structures will be stretched beyond elastic strain. This will lead to permanent deformation of the parts. For this reason the elastic analysis will not be sufficient. The analysis need to take into consideration the plastic hardening of the materials too.

In ANSYS –program material information of the structure was determined. Final modelling was done using materials with bilinear hardening properties. The materials are listed in table 3. ANSYS requires values for yield strength, Young's modulus (E) and

Tangent Modulus (E_t) to determine the stress-strain curve. In figure 7 the stress-strain – curve of used materials is presented.

TABLE 3. Material properties

Material	Yield strength [MPa]	Tensile strength [MPa]	Elongation [%]	Tangent modulus [MPa]
S355K2+N [1)]	345	470	20	1212
42CrMo4 [2)]	650	900	12	3248
34CrNiMo6 + QT [3)]	900	1100	10	3406
BISO 12.9 [4)]	1100	1220	8	3034
S650MC [5)]	650	700	14	1157
38MnVS6 [6)]	580	850	12	3032

[1)] S355K2+N data sheet, Oakley Steel

[2)] 42CrMo4 datasheet, Ovako (Appendix 1)

[3)] 34CrNiMo6 + QT data sheet, Ovako

[4)] BISO 12.9 data sheet, BS EN ISO 898-1:1999

[5)] Strenx 650 data sheet, SSAB (Appendix 2)

[6)] Cromax 482 data sheet, Ovako (Appendix 3)

Tangent modulus is calculated using equation

$$E_t = \frac{\sigma_2 - \sigma_1}{\varepsilon_2 - \varepsilon_1}$$

where σ_1 is Yield strength, the σ_2 can be defined from Yield strength and elongation (A_5) (Roylance, 2001) :

$$\sigma_2 = \sigma_1 \left(1 + \frac{A_5}{100}\right)$$

The strain ε_1 is calculated according to Hooke’s law from the Yield strength and Young’s modulus E, which is approximately the same for all steel, being about 210 GPa.

$$\varepsilon_1 = \frac{\sigma_1}{E}$$

The strain $\varepsilon_2$is determined using elongation (Roylance, 2001):

$$\varepsilon_2 = \ln\left(1 + \frac{A_5}{100}\right)$$

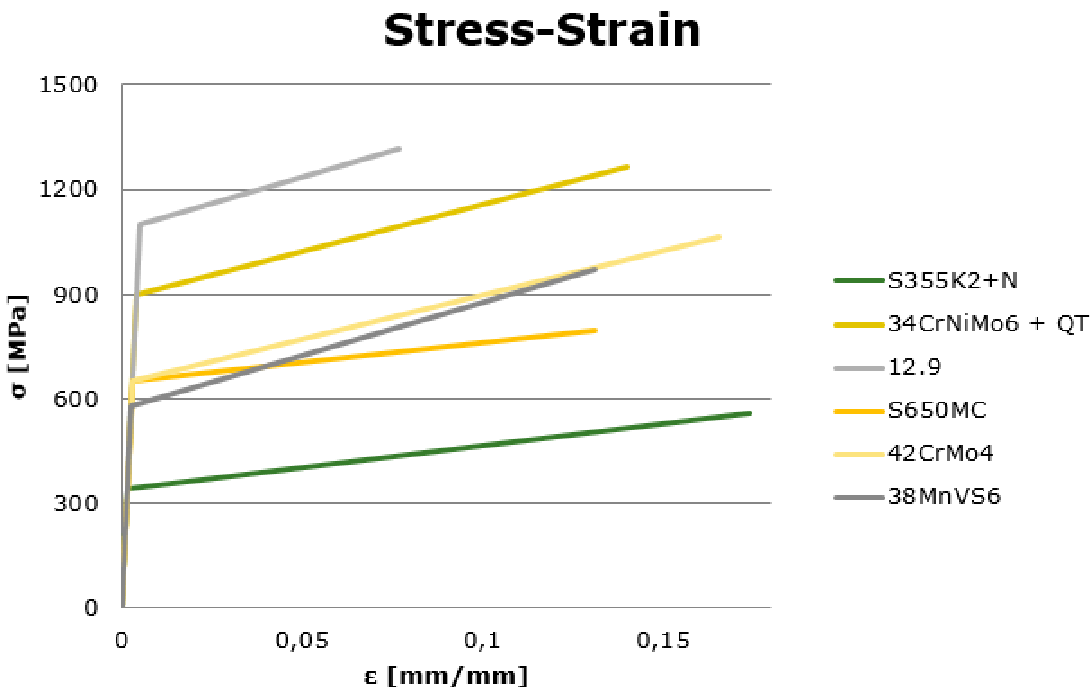
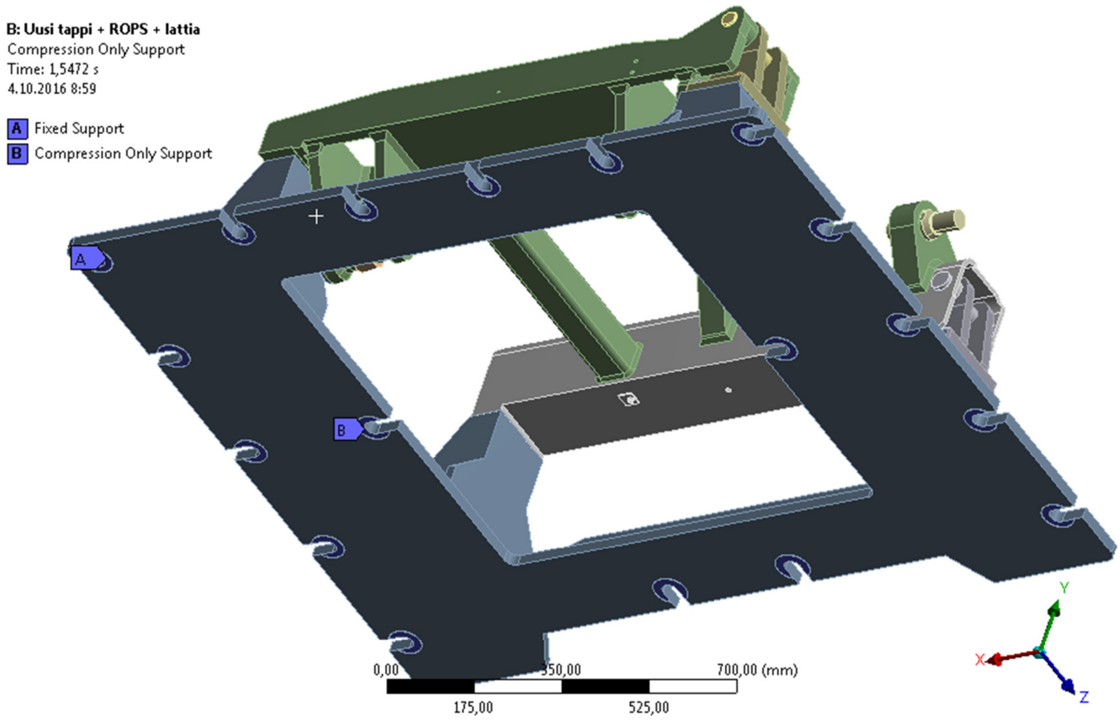


FIGURE 7. Stress-strain curves for materials used

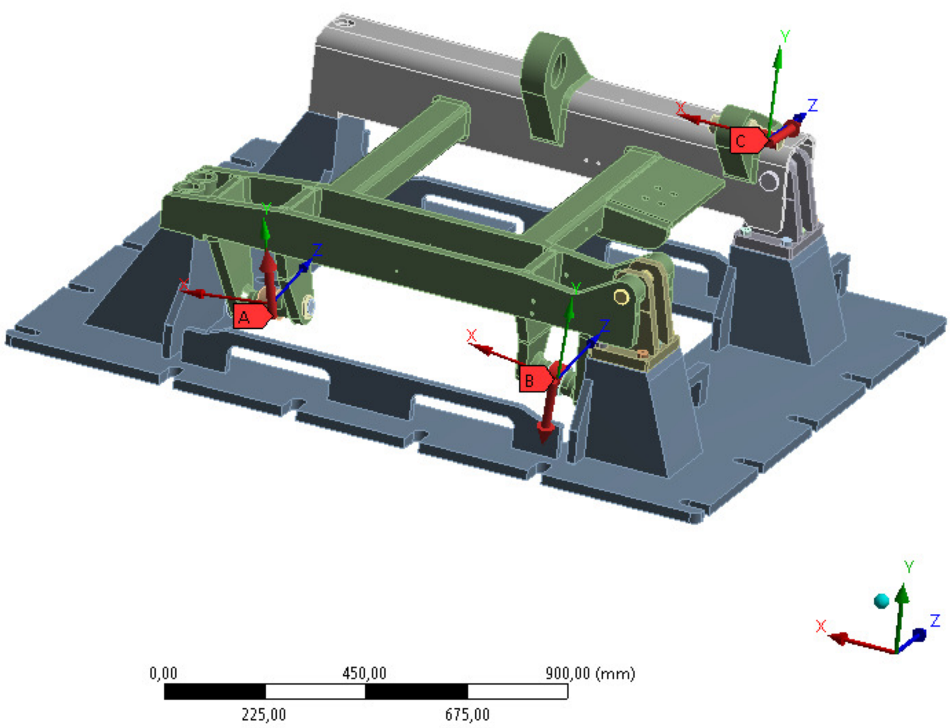
6.5 Supports and loads

The structure was supported using fixed supports and compression only –support (picture 16). The supports were defined to the bottom of test bench as it is fixed to the floor in test situation. The fixing was done via sixteen surfaces extruded around cuts at the outer edge of the plate (A). Fixed supports restrict movement of the surfaces completely. Compression only –support restricts the bottom part of the bench from penetrating the imaginary floor beneath but allows the compression. The supports were attached to the two surfaces extruded around cuts at the inner edge of the plate (B).



PICTURE 16. Supports

The loads applied to the structure in ROPS tests were calculated from the force that is used in the actual test. The forces were applied to the cylinder lower mount points and to the tie rod front mount point (picture 17). The recommended pretension was applied to the bolts and to the cylinder pins (Valtanen, 2009).



PICTURE 17. Loads

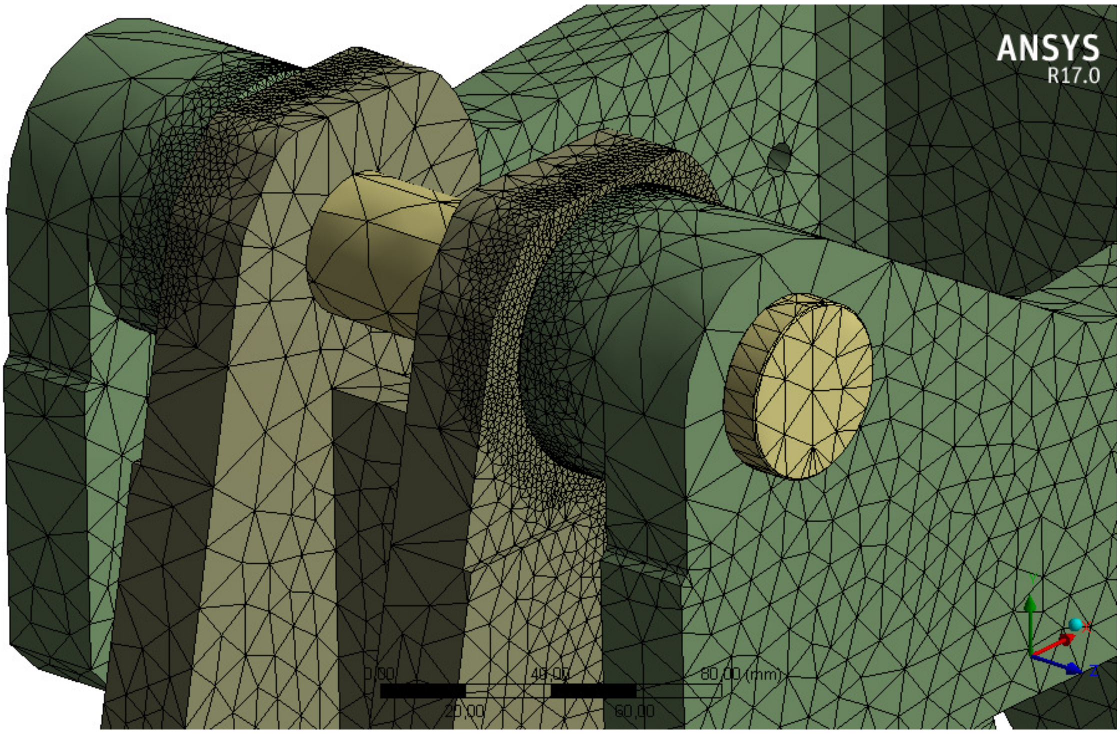
6.6 Contacts

Since there are no actual threads modelled to the bolts and cylinder pins the bonded contacts are used for the threaded zones. It is assumed that the thread is strong enough to survive the loads applied. Also the pins with flat heads are bonded to the levelling structure from one end to prevent the pins from parting from the levelling structure totally. Also the pin attaching the front end of tie rod is bonded to the levelling structure.

Frictionless contacts are applied to the under surface of the bolt head. Rough contacts are applied to the inside of cylindrical parts to ensure the parts don't part from the rods but to simulate their attachment to the rod. Other contacts are frictional contacts with friction coefficient of 0,2. Using frictionless contacts is in some cases faster and the results are close to real life but in this complex case the frictional coefficient actually settles the movements making the analysis converge faster.

6.7 Meshing

Coarse mesh with larger elements was applied to parts with less importance like front frame. With rough meshing (element size 80 mm) the structure was pre-analysed to determine the most critical parts of the structure. The most critical parts that had higher stress gradient were then meshed with denser mesh (element size 10mm). Finally one contact area was meshed with 2 mm element size to eliminate unwanted penetration of contact pairs (picture 18). The amount of nodes with final meshing was 937 955 and there were 608 648 elements in the model.



PICTURE 18. Meshing of the structure

Changing the element size and the mesh density has huge effect on the computing time. On the other hand with coarse mesh a lot of data is lost or not achieved so the optimizing the meshing is very important in complex constructions. The mesh quality has really big impact on the stress levels reached as seen from figure 8, so finding the perfect mesh quality is important. (Norton, 2011)

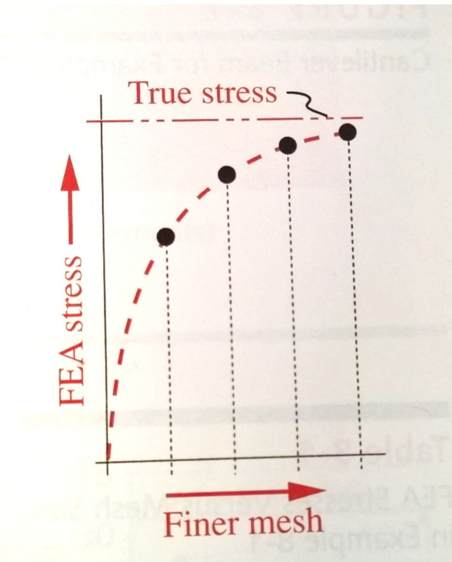


FIGURE 8. Effect of mesh quality to FEA stress compared to true stress (Norton, 2011)

6.8 Analysis sequence

The analysis was run by program controlled settings. The first step was loading the bolts with pretension and the second step loaded the full force to the set surfaces (figure 9).

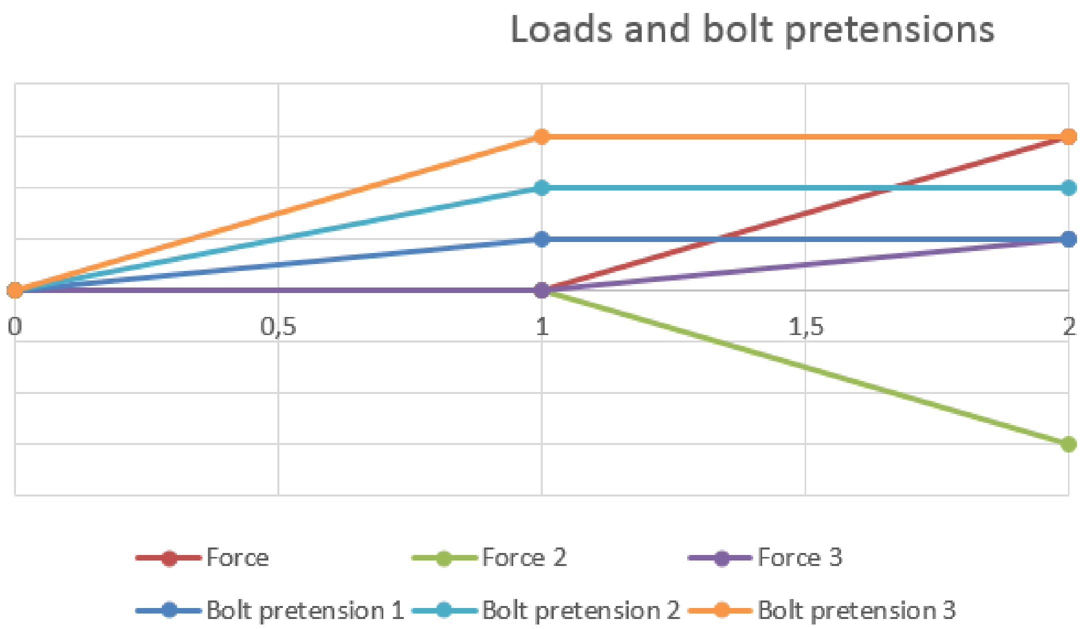


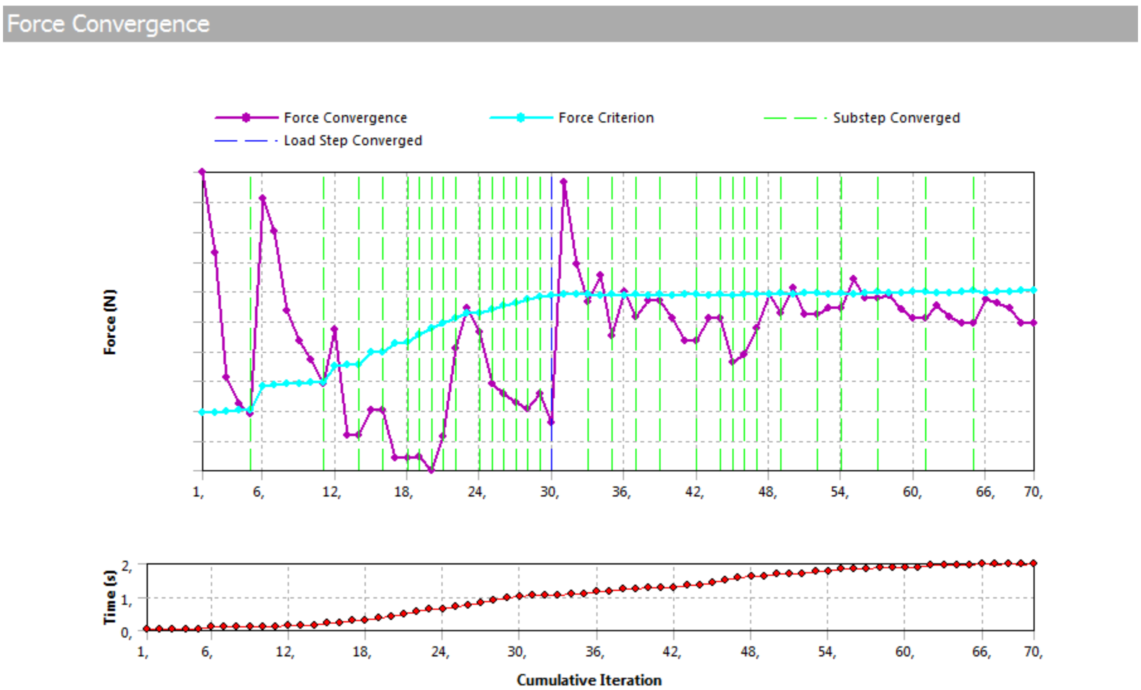
FIGURE 9. Loading steps.

7 RESULTS

7.1 Solving the model

First analyses were made using linear materials to test the model. Since the loads applied are so heavy the stresses in structures go beyond yield strength and in some points very close to tensile strength so the results weren't very realistic. The purpose of these analyses were to check the model and try to make sure it is suitable for the analysis since the time consumed for the nonlinear analysis is so long.

After first results showed the model was appropriate for the analysis the materials were changed to materials that are currently used and bilinear properties were added to them. The model was elaborated and the analysis was carried out again. From the result could be seen that some pins are under very big stress and they will probably not survive it. The analysis was monitored from the force convergence curve (picture 19).



PICTURE 19. Force convergence curve.

Since there is no explicit solution for non-linear model the numerical iterations need to be made to solve the problem. Convergence function compares the externally applied loads to the nodal forces from internal element stresses. From the figure 10 the idea of the residual force can be seen.

$$\underbrace{\left\{\mathbf{R}^{\mathrm{ext}}\right\}}_{\substack{\text{externally}\\ \text{applied loads}}} = \underbrace{\left\{\mathbf{R}^{\mathrm{int}}\right\}}_{\substack{\text{nodal forces}\\ \text{from internal}\\ \text{element stresses}}} \quad \Leftrightarrow \quad \left\{\mathbf{R}^{\mathrm{res}}\right\} = \left\{\mathbf{R}^{\mathrm{ext}}\right\} - \left\{\mathbf{R}^{\mathrm{int}}\right\} = \{\mathbf{0}\}$$

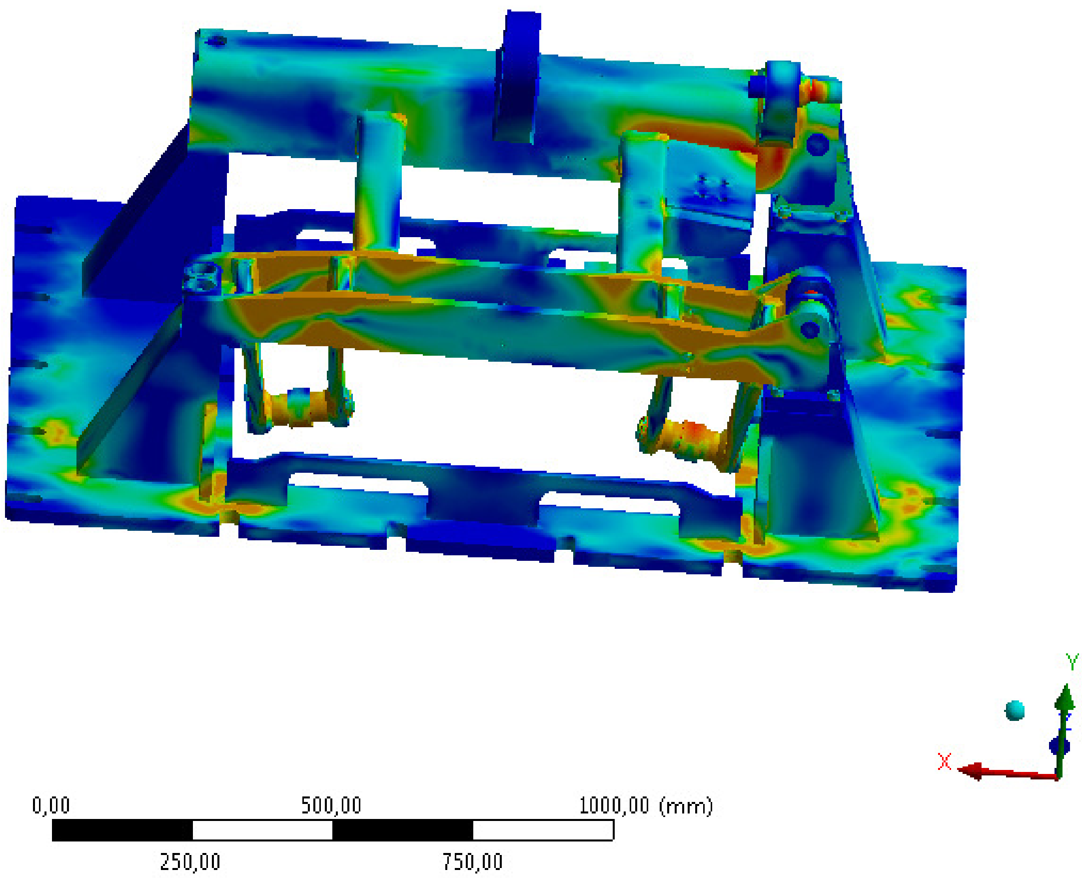
FIGURE 10. The force convergence residual equation (Mathisen, 2012)

In ideal situation the residual would be zero, but in the real case it is not. When the difference between input and output forces is below certain level the model for this force level is converged and the analysis continues on. This iteration cycle goes on until the load levels are reach with convergence values acceptable. The level of convergence value was left for ANSYS to decide. The time in the picture 19 is not time in seconds but it shows the load steps. Step one is the bolt pretension step and the step two is the actual loading of forces.

7.2 Equivalent (von-Mises) stress

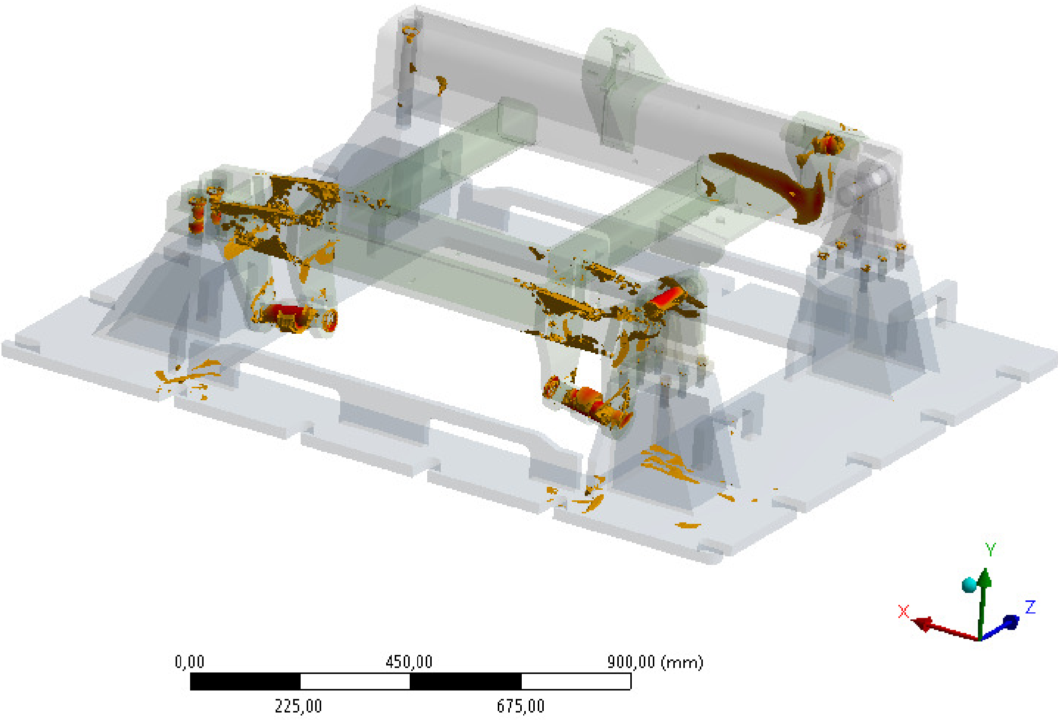
The equivalent stress that is the von-Mises stress is based on the distortion energy failure theory. This means that the material will break if the distortion energy in actual case is more than the distortion energy in a simple tension case at the time of failure. (Lähteenmäki, 2012) This theory is widely used in ductile material cases and thus selected from ANSYS' stress options for this analysis.

The results showed clearly that some of the parts are under very high stress and they will probably not survive it. The overall analysis results are in picture 20.

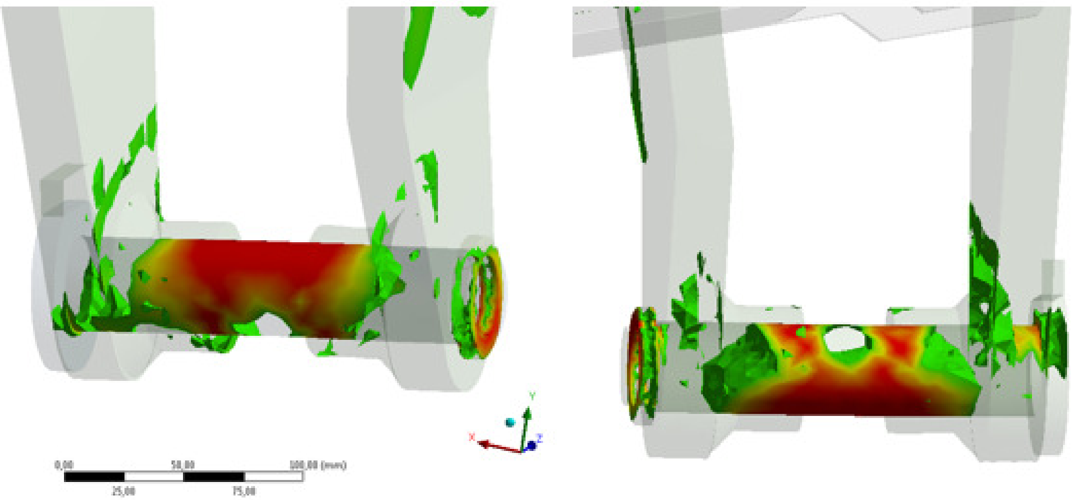


PICTURE 20. Equivalent (von-Mises) stress levels in levelling structure in ROPS –test

The individual parts were carefully monitored to see the most critical parts of the structure (pictures 21 and 22). Using the capped isosurfaces analysis tool in ANSYS made it easier to spot the parts where stress was at highest. With this tool the stress levels beyond specific value can be highlighted against the whole structure as seen in the pictures.

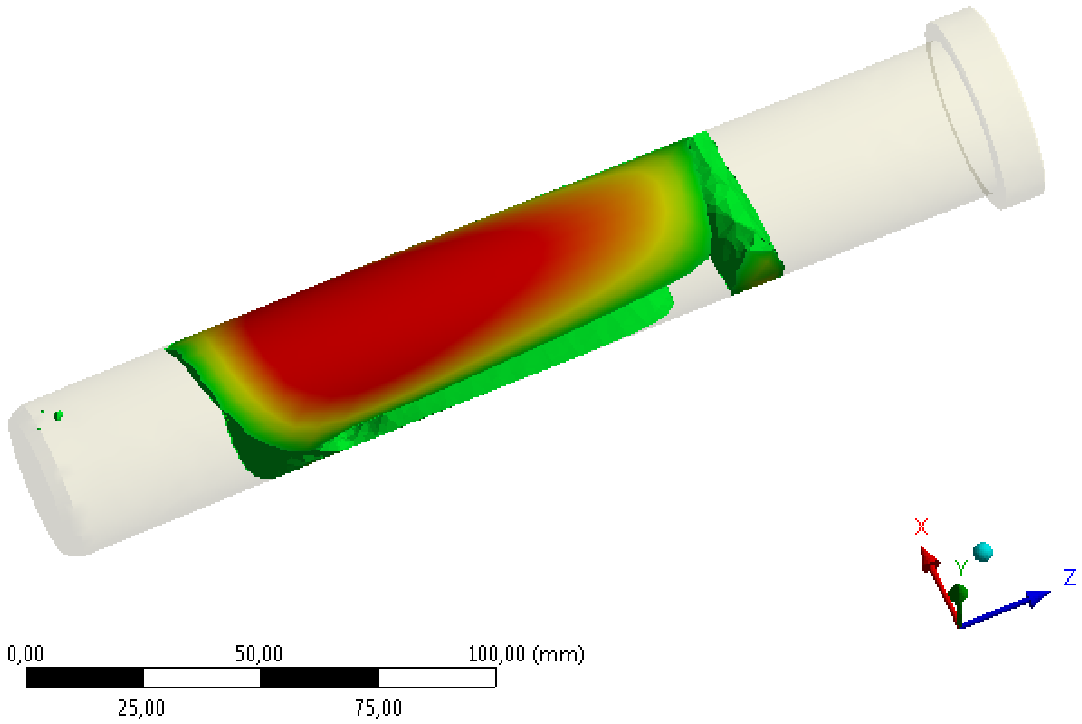


PICTURE 21. High stress levels in the frame.



PICTURE 22. Critical stress levels in cylinder pins.

The hinge pin also has high stress levels (picture 23) going even close to tensile stress.

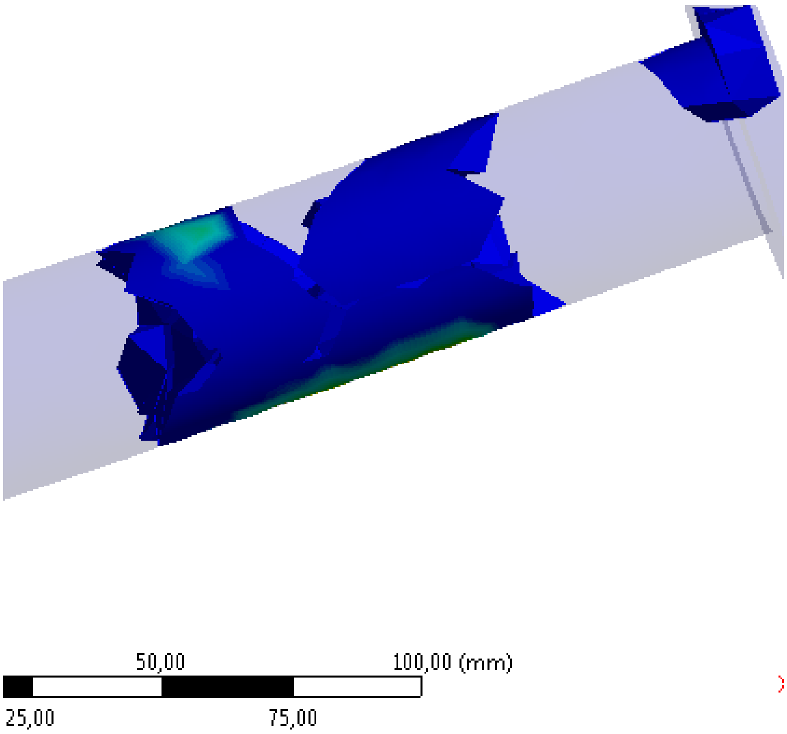


PICTURE 23. Critical stress level area in a hinge pin

The stress levels were compared to both yield stress and tensile stress and careful estimation was done to determine if the stress levels can be acceptable.

7.3 Equivalent plastic strain

Since the loads applied to the structure were heavy and caused high stresses the plastic strain was very important measure to investigate. In different phases of the analysis process the plastic strain was carefully analysed. In picture 24 the highest strain points are highlighted against the rest of the part.

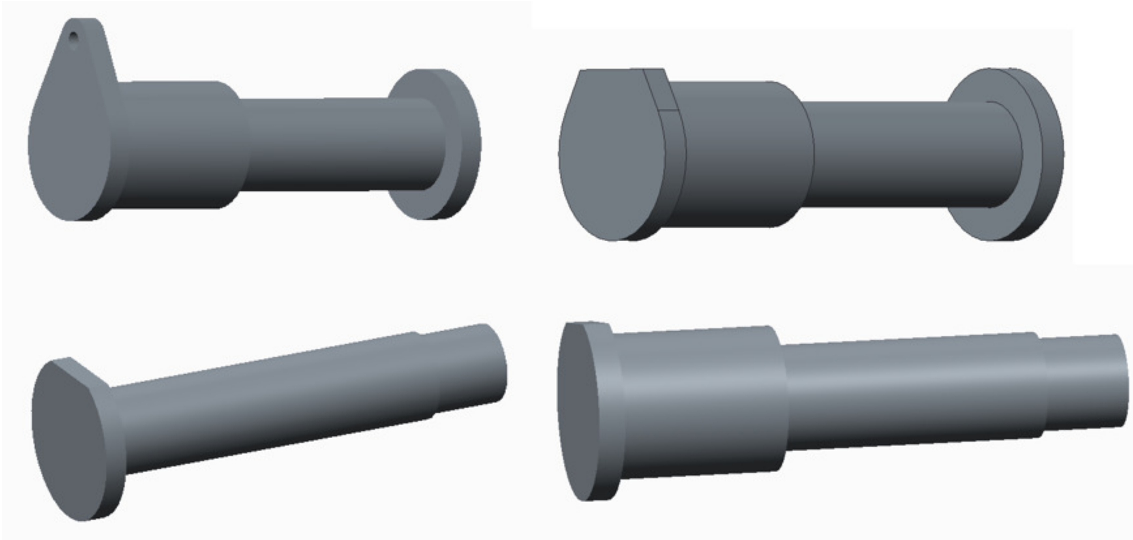


PICTURE 24. Plastic strain in pin.

The plastic strain values detected in the structure were compared to the limits (namely ε_2) the material has to determine if the material can survive the strain.

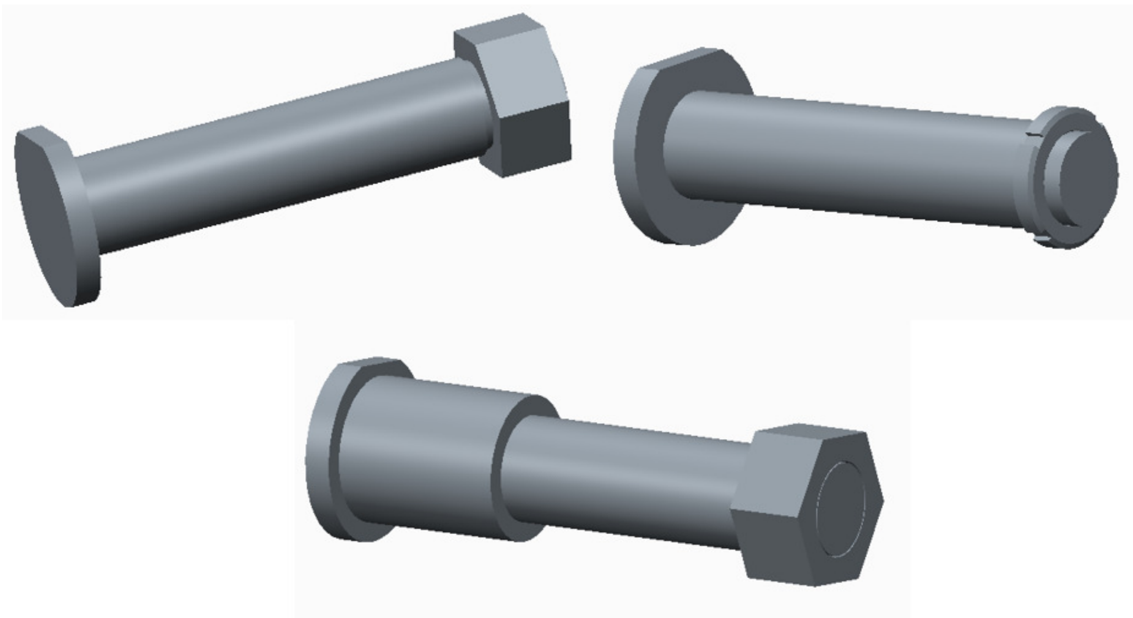
7.4 Redesign and modifications

It was seen in the first analysis that some parts will suffer more stress than they can bear so redesign and some material changes needed to be made. A cylinder lower mount pin will be under heavy stress, so the material needs to be changed. As the geometry makes restrictions to material choices the geometry needs also to be modified. Different pin geometries (picture 25) were tested in the model. The first geometry had too narrow tip for locking the pin and it didn't survive the analysis. The second geometry was good but it needed a little more modifying to take into consideration the manufacturability.



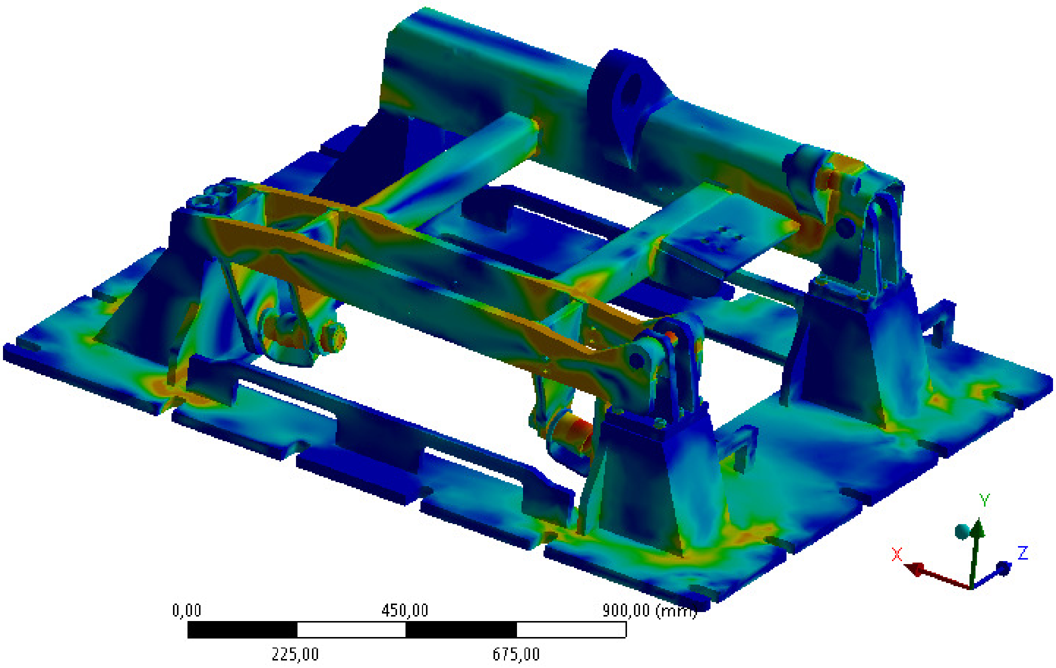
PICTURE 25. Different pin geometries for cylinder pin.

At the beginning the nuts were modelled as fixed structure at the end of pin (as seen in the picture 25 the top two pins), but later the nut geometry was also defined more carefully (picture 26). The first version was hexagon nut but since the space is very tight it was changed to a locking nut. The last pin version with additional support structure needed sturdier nut to survive so the hexagon nut was again taken into the model. The space limitations need to be checked carefully when the final version is chosen.



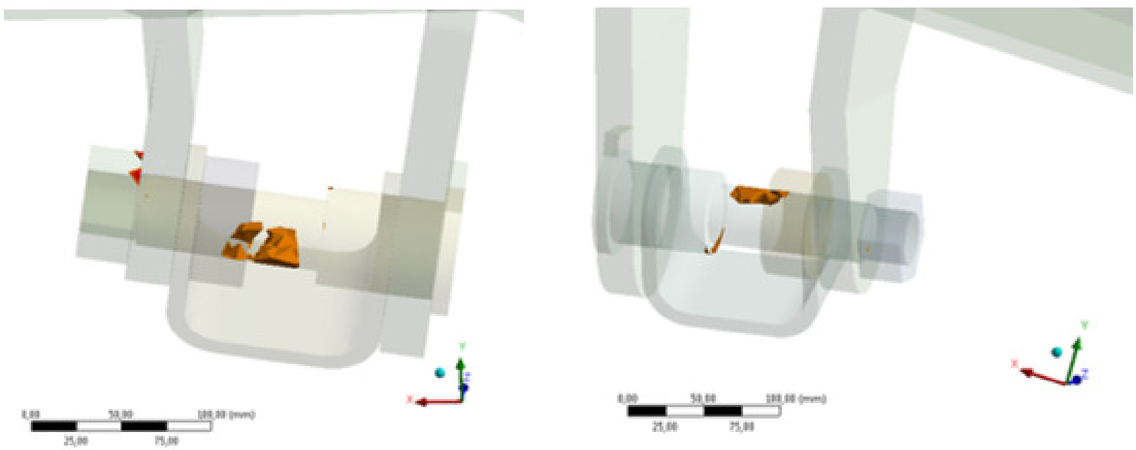
PICTURE 26. Nut geometry designs

Since the change of pin geometry didn’t bring enough relief for the stress it became clear that the material change and pin geometry change would not be enough. A Ushaped support structure was designed for the lower part of the cylinder mounting point. The new structure with stress distribution is shown in picture 27.



PICTURE 27. Modified test structure with von-Mises stress levels.

Stress levels dropped throughout the structure with the modifications. Tensile strengths are not reached anymore. The stronger materials will enable the pins withstand higher stress levels. (Picture 28)



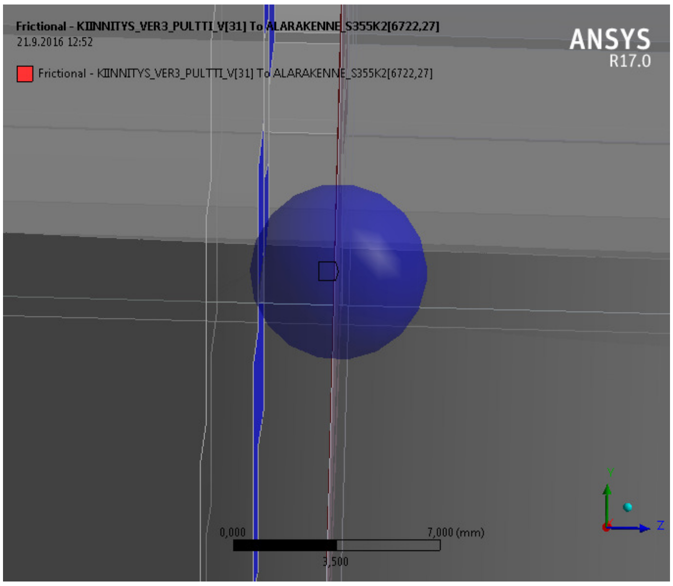
PICTURE 28. Dropped stress levels in cylinder pins.

The stress levels are still high and go beyond yield strength but after careful estimations the structure seems strong enough to go through ROPS –test with success. Plastic strain seems to be in sustainable level so it shouldn't cause the structure to break down unwantedly.

7.5 Problems during analysis

The first problem that occurred was meshing problem. The large complex sheet metal parts didn't mesh properly. The program couldn't form a mesh to all parts of the structure and it was quite difficult to figure out what was the problem. Many iteration rounds on shrinkwrap simplifications needed to be made in order for the structure to be simple enough to mesh fully.

There were also a lot of problems with the connections. Since the model is under a really heavy loading it deforms significantly. This bending influences on how the material behaves in ANSYS. With standard settings ANSYS doesn't always recognise where the boundaries of parts are and lets the parts penetrate each other the way that is not really possible. This was the main problem during analysis. The corrective actions was to increase the pin ball –region. This helps ANSYS recognise where the boundaries of the parts are (picture 29). Pin ball –region tells ANSYS that the edges/surfaces of two parts are within the pin ball –region.



PICTURE 29. Pin ball –region in blue

The other corrective was to increase the steps to the loading of force. ANSYS automatically applies the force with certain steps it decides are the best. By setting manually the steps in which ANSYS increases the applied load can help ANSYS recognise the boundaries better. When adding sub steps to loading the possible movements of parts happen more slowly and the program has more chances to recognize the upcoming boundaries better. This though increases the already very long analysis time even more. With automatic settings the analysis time was something between 6 hours to 12 hours but with manually increased amount of steps to 35 – 45 steps the time increased up to one and a half days.

After some test runs the number of sub steps were limited between 15 to 25 steps and the program was allowed to optimize the number of steps within the limitation (table 4). The addition of sub steps increases the time of analysis significantly and also requires more disc space witch caused problems during analysis since the program crashed just before saving results in the end.

TABLE 4. Analysis sequence and sub steps

Step Controls	
Number Of Steps	2,
Current Step Number	2,
Step End Time	2, s
Auto Time Stepping	On
Define By	Substeps
Carry Over Time Step	Off
Initial Substeps	20,
Minimum Substeps	15,
Maximum Substeps	25,

7.6 Reliability of the results

Since the analysed structure is very complicated a lot of simplifications needed to be made. That will automatically lessen the reliability of the results. However the aim was to get pre-results in order to make estimations if the structure would survive the actual tests. For this purpose the results were good enough and the second phase is to finalize the structure and make the final material decisions. The strain will be measured very carefully during tests so if the structure fails the tests new analysis can be carried out via finite element analysis with more accurate information on what the force really is for the critical parts.

The loads that are applied during the tests are transferred in the simulation as forces to the cylinders. This was done to simplify the structure; with this simplification the cabin itself could be left out from the analysis. Calculation of forces was done manually using statics and will only give approximate information on how the load really affects.

The analysis process was started with as simple structure as possible; linear materials and coarse meshing. With results from the analysis the model was defined more accurately by increasing the density of the mesh, adding bilinear material properties to the parts, bringing more parts to the assembly (for example nuts and bolts) and defining the contacts more accurately. In the beginning the contacts were made either frictionless or bonded but as the model evolved they were made frictional with actual friction coefficient or rough to make the model describe the reality more accurately.

ROPS-test is done below freezing point which will also have an effect on the results. As can be seen from the figure 11 the temperature has enormous effect on the ductility of the material. The impact strength on the y-axis grows significantly with temperature (x-axis).

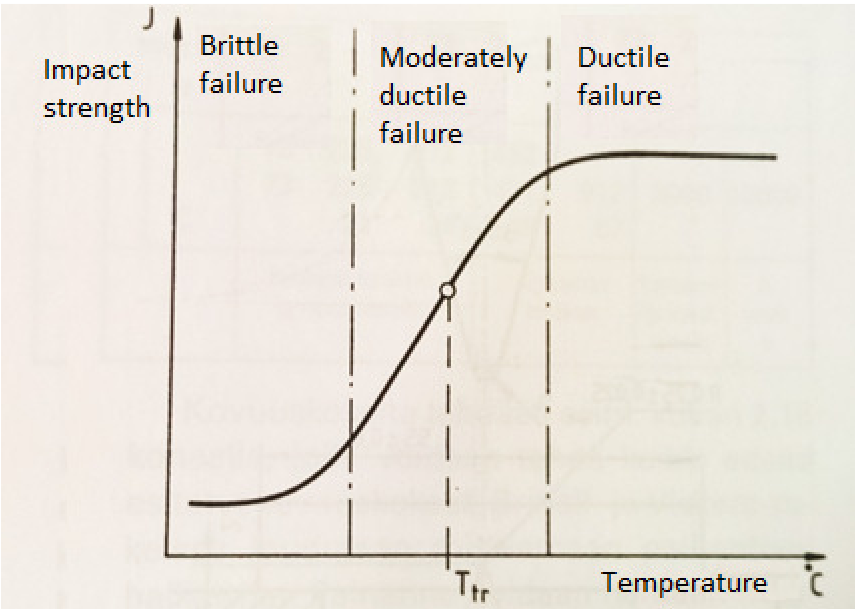


FIGURE 11. Impact strength of steel (Koivisto et. al, 2008)

As the stress values are close and beyond yield strength and even close to tensile strength the actual material properties need to be known thoroughly to make estimations of the durability of the structure. For the most critical parts materials need to have certification from the manufacturer to be sure of the actual properties. Normally the properties are stated as limits in which the material is but in this case accurate values are needed to be sure that the parts are also tested against SFS-EN 10204 in cold temperatures too.

8 DISCUSSION

The objective of this thesis was to study the forwarder levelling structure and analyse its behaviour in ROPS –test. In the ROPS –test a load is applied to a machine cabin or other protective structure to see if it can protect the machine operator in case of rollover of the vehicle. The structure needs to be both stiff and flexible at the same time so that it can stand the loads without breaking and to absorb as much energy as possible.

ROPS –test was chosen to be static and it will be done in laboratory circumstances. In the test set up the stripped version of cabin will be mounted to the levelling structures as it is during normal operation. The levelling structures are mounted to specific test bench that is designed specifically for ROPS –test. The test bench is designed to mimic the front frame of the forwarder as well as possible.

The analysis was done with ANSYS – software using finite element method. The analysis was carried out as static structural –analysis. The model of the ROPS –test set up was transferred into ANSYS Workbench. Appropriate loads and support were added and all parts were connected to each other as realistically as possible.

It was guesstimated before analyses that the 29 tonne ROPS would need some strengthening to pass the 33 tonne ROPS –test. This assumption was vindicated during process. The parts with highest stress levels were detected and then strengthen by changing the material, redesign of some parts and adding supportive structures. Further optimisation and analyses improved the results. At the end of the process the structure was evaluated to probably withstand the stress levels that will be present in the actual test. As the forwarder needs to be as light as possible and the spacing inside is very tight careful consideration needs to be made in order for the structure to be strong enough but not too heavy, and that will bring some uncertainty to the test situation.

As the loads in case of rollover are so much higher than any loads during normal operation the analysis is very difficult as the structures are at their limits. Due to cost of the test and preparations the analysation beforehand is important but final proof of the strength of the structure is gotten from the actual official test. There are also studies done to improve the analysis methods to be able to get reliable results from calculations without the need to do actual destructive tests. (Clark, 2005) If the calculations are the only method to certify ROPS in the future the need to understand finite element analysis in detail is critical to get reliable results. As this is not sufficient according to current laws and regulations also Deere will need to go through the ROPS –tests for this forwarder. The tests will be done after the new structure is finalized and the parts for the test ordered.

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APPENDICES

Appendix 1. 42CrMo4 – data sheet by Ovako

OVAKO — PRODUCT SPECIFICATION — Sivu 1(3)

42CrMo4		
	EN 10083-3:2006	42CrMo4
	Imatran tunnus	MoC 410 M
	Hoforsin tunnus	327A
	Smedjebackenin tunnus	42CrMo4

Yleistä

42CrMo4 on nuorrutusteräs, jolla on hyvä lujuuden ja sitkeyden yhdistelmä nuorrutetussa tilassa. Öljykarkaisussa se karkenee läpi noin Ø60 mm:n pyörötankona. Suuremmissa mitoissa keskustan rakenne ei ole täysin martensiittinen. Teräs soveltuu hyvin induktiokarkaisuun. Pintakovuudeksi saadaan vähintään 53 HRC. 42CrMo4 voidaan toimittaa M-käsiteltynä hyvän lastuttavuuden takaamiseksi.

Standardi

Tämä aineslehti perustuu europpalaiseen standardiin SFS-EN 10083-3:2006, jonka vaatimukset teräs täyttää.

Teräs voidaan toimittaa myös seuraavien europpalaisten standardien mukaisena:

- EN 10250-3 Open die steel forgings for general engineering purposes. Part 3: Special alloy steels
- EN 10263-4 Steel rod, bars and wire for cold heading and cold extrusion
- EN 10269 Steels for fasteners with specified elevated and/or low temperatures properties[a]
- EN 10277-5 Bright steel products. Steels for quenching and tempering[a]
- EN 10297 Seamless circular steel tubes for mechanical and general engineering purposes
- EN 10305 Seamless cold drawn tubes for precision applications
- prEN 10343 Steels for quenching and tempering for construction purposes

[a]Julkaistu myös SFS-EN -standardina

Kemiallinen koostumus

	C%	Si%	Mn%	P%	S%	Cr%	Mo%
min.	0,38	0,15	0,60			0,90	0,15
max.	0,45	0,40	0,90	0,025	0,035	1,20	0,30

Mekaaniset ominaisuudet

Tila	Tangon halkaisija Ø mm	Myötöraja R_e min. MPa	Murto-lujuus R_m MPa	Murto-venymä A_5 min. %	Murto-kurouma Z min. %	Iskusitkeys KV min. J	T °C	Kovuus HBW
Kuuma-valssattu	25...180							...350[b]
Hehkutettu								...241
Nuorrutettu	25...40	750	1000...1200	11	45	35	20	300...350[b]
	>40...100	650	900...1100	12	50	35	20	270...320[b]
	>100...160	550	800...950	13	50	35	20	240...280[b]
	>160...180	500	750...900	14	55	35	20	220...270[b]

[b]Ohjearvo

Appendix 2. Strenx 650 MC – data sheet by SSAB

STRENX™
PERFORMANCE STEEL

STRENX 650 MC

STRENX 650 MC

Tuotteen kuvaus

Erikoisluja rakenneteräs 650 MPa
Strenx™ 650 MC on kylmämuovaukseen soveltuva kuumavalssattu rakenneteräs, jonka myötölujuus on vähintään 650 MPa ja joka tekee rakenteista vahvempia ja kevyempiä.
Strenx 650 MC vastaa EN 10149-2-standardissa S650MC-teräkselle asetettuja vaatimuksia tai ylittää ne.
Tyypillisiä käyttökohteita ovat vaativissa olosuhteissa käytettävien kantavien rakenteiden komponentit ja osat. Strenx 650 MC-terästä on saatavana keloina, rainoina ja määrämittaisina levyinä.

Mittavalikoima
Strenx 650 MC-terästä on saatavana 2,00–10,00 mm:n paksuisina ja 1 600 mm leveinä keloina, rainoina tai määrämittoihin leikattuina nauhalevyinä 16 metrin pituuteen asti.

Mekaaniset ominaisuudet

Paksuus (mm)	Myötölujuus R_{eH} [1)2)] (min MPa)	Murtolujuus R_m (MPa)	Venymä A_{80} [3)] (min %)	Venymä A_5 (min %)	Pienin sisäpuolinen taivutussäde, taivutuskulma 90°[5)]
2- 3	650	700- 850	12	14[4)]	0.8 x t
3.01- 6	650	700- 850		14	1.2 x t
6.01- 10	650	700- 850		14	1.5 x t

Mekaaniset ominaisuudet testataan pitkittäissuunnassa.
[1)] Jos R_{eH} ei sovellu, käytetään Rp 0,2:ta.
[2)] Jos paksuus on > 8 mm, vähimmäismyötölujuus voi olla 20 MPa pienempi.
[3)] Arvo A_{80} koskee paksuuksia < 3,00 mm.
[4)] Arvo A5 koskee levypaksuutta t ≥ 3mm.
[5)] Pitkittäis- ja poikittaissuunta.

SSAB

Appendix 3. Cromax 482 – data sheet by Ovako

Induction-hardened **Cromax® IH 482** is based on a medium carbon, micro-alloyed steel, which is characterised by high strength in the as-rolled condition, i.e. without heat treatment. The 482 base steel is a cost-effective alternative to traditional low-alloy, quenched and tempered grades with, in the context of piston-rod applications, equivalent properties.

The analysis of the base steel in Cromax IH 482 is well adapted to induction hardening and a high and uniform hardness is achieved throughout the case irrespective of diameter. In consequence, the resistance to even high-energy external impact is excellent.

Average chemical analysis Cromax® IH 482

C %	Si %	Mn %	S %	V %	C.E. %(*)
0.39	0.40	1.20	0.02	0.13	max 0.72

*C.E. = % C + % Mn/6 + (% Cu + % Ni)/15 + (% Cr + % Mo + % V)/5

Corresponding standards

The table shows the closest equivalent standard for the steel in Cromax IH 482. In most cases, the correspondence is only approximate.

Cromax	EN	DIN	AFNOR	SAE/ASTM
482	38MnV56	38MnSiV55	30MV6	1045V

Mechanical properties

Yield stress, $R_{p0.2}$, N/mm², min.	Ultimate tensile stress, R_m, N/mm²	Elongation, A_5, %, min.	Hardness, HB	Toughness, KV
580	850 - 1000	14	250 - 300	No guarantee given, but normally 15-30 J at 20°C

Chrome layer

The thickness of the chrome layer is minimum 20 µm.

Surface roughness

The surface roughness (Ra) is always less than 0.2 µm and normally in the range 0.05-0.15 µm. Rt (ISO) is always less than 2.0 µm and normally in the range 0.5-1.5 µm.

Surface hardness, induction hardening

The chrome layer hardness is 850 $HV_{0.1}$ min. The surface hardness in the induction-hardened zone immediately beneath the chrome layer is 55 HRC min.

The depth of hardening, which is defined as the distance from the steel/chrome interface at which the hardness has dropped to 400 HV_5, is dependent on diameter as tabulated below:

Size, Φ mm	Hardening depth, mm
≤ 28	1.0 - 1.5
> 28 - 40	1.3 - 1.7
> 40	1.7 - 2.3

Other hardening depths can be supplied by special arrangement.

Straightness

The maximum deviation is 0.2 mm/1.0 m.

Roundness

The out of roundness is maximised at 50% of the diameter tolerance interval.

Diameter tolerance

ISO f7 is standard. Other tolerances can be supplied upon request (narrowest range is ISO level 7).

Tolerance ranges

Size, mm	ISO f7, µm upper	ISO f7, µm lower
> 18 - 30	- 20	- 41
> 30 - 50	- 25	- 50
> 50 - 80	- 30	- 60
> 80 - 120	- 36	- 71
> 120 - 180	- 43	- 83

Standard sizes

Dia., mm	kg/m	Dia., mm	kg/m	Dia., inch	kg/m
25	3.85	60	22.19	1	3.97
28	4.83	63	24.47	1¼	6.22
30	5.55	65	26.05	1½	8.94
32	6.31	70	30.21	1¾	12.19
35	7.55	75	34.68	2	15.91
36	7.99	80	39.46	2¼	20.13
38	8.90	85	44.54	2½	24.87
40	9.86	90	49.94	2¾	30.09
42	10.88	100	61.65	3	35.81
45	12.48	110	74.60	3¼	42.03
50	15.41	120	88.78	3½	48.72
55	18.65	125	96.33	4	63.65
56	19.33			4½	80.55

Other sizes can be supplied upon request but only within the range 12-140 mm inclusive.

Delivery lengths

Production lengths are between 4.0-7.6 m. Standard is 6.1+0.1/-0 m. Bars with length 7.6+0.1/-0 m can only be supplied for diameters between 40-80 mm.

The "unchromed length" of each bar, i.e. the distance at each end over which the chrome-layer properties and tolerances can not be guaranteed, is at most 0.15 m per end, i.e. 0.3 m in total per bar.

Fixed, cut lengths can be supplied if required, but at a higher price than production lengths.

Weldability

Cromax IH 482 can be MMA or MAG welded at elevated temperature. Preheating to 200-300°C is recommended; however, the upper limit should not be exceeded because of risk for deterioration of the chrome layer.

Cromax IH 482 can normally be friction welded without problems. Special procedures may, however, be necessary for larger diameters.