Acknowledgement

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We were very lucky to get the chance of doing thesis work at Mack Trucks, a genuine engineering company. The project we worked with involved different engineering areas and suited our education perfectly. To be able to go abroad and do our thesis work has given us many valuable experiences and memories, both at Mack Trucks and on our free time.

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We would like to thank our supervisor Mathias Wallin at the Division of Solid Mechanics for his feedback, help and guidance through the project. We would also like to thank him for his contact to Altair helping us get access to Optistruct.

Lund, August 2006
Jörgen Marken and Kristina Tejler

Abstract

Mack Trucks wanted to examine the possibility to replace the sheet metal solution of the front end extension of the vocational trucks with a cast solution. There are different front end extensions and in this project only the 5.5” extension is treated. Mack would like to find a cast solution that reduces weight, cost and also improve assembly issues.

The front end extension has two tow alternatives. Single tow, with only one point to fasten the towing chain, or dual tow, with two points. The new cast design needs to withstand five different load cases that are set up as recommended practice from the American Truck Association. The largest test load is a forward pull of 430 kN. This load case corresponds to hanging the fully loaded truck from a hook in the ceiling.

Many different designs were suggested during the project. In the end there were three main designs left. First a modular design, consisting of two cast side brackets and a middle section of sheet metal. The side brackets are made from the same mould, but machined differently for the single and dual tow option. There are two different middle sections for the each options. Second design consists of one large cast piece. The design is only for the single tow. Third design is a three piece cast design, also only for the single tow option. The last two design options were developed with the help of an optimization tool, the program Optistruct. All designs was finally analysed with the finite element program I-Deas.

The final solution that passed the analyses was the modular design. The single tow option weighs 95 kg compared with 96 kg of the existing sheet metal design it aims to replace. The price is not yet final for this option. The dual tow weighs 46 kg compared to 64 kg of the existing sheet metal design. This is a decrease of 28 %. The quoting was $ 215 compared to $ 414, a reduction of 48 %.

Due to time limitations, the single and three piece cast solution was not redesigned after result of the stress analyses. All of them had unacceptable stress concentrations around the bolt interface connecting to the frame rail and need further work. The option that looks the most promising is the single cast piece with a pin as a towing device. In the stage where the design is now, the weight is almost 20 kg lighter than the existing single tow, 76 kg compared to 96 kg.

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1 Introduction

The background and the description of this design problem of the towing device for trucks. The goals are stated and the existing designs that the solution will replace is shown in detail. Finally, the limitations are given.

1.1 Background

Mack Trucks are looking for a new solution to the front end configuration of their vocational trucks. The front end configuration includes the towing device and is attached to the frame rails. It is also to the front end that the bumper mounts. Today, the designs of the different options of front ends are made in sheet metal. Mack Trucks would like to investigate the possibilities to find a cast design. The reason for replacing the sheet metal with cast material is to reduce manufacturing costs and weight, and also to solve some of today's assembly issues.

The 1st of January 2007, the Federal law containing restrictions on diesel engine emissions starts to apply. The project 'US07' at Mack Trucks is designing new trucks that meet the emission demands. The main difference to the front end of the 'US07'-trucks is that the cooling package for the engine is bigger. This Master Thesis Project will find a design solution to the front end that will fit the vocational trucks in the 'US07' project. Weight and price are the two most important matters when evaluating the designs, but also assembly issues should be taken into consideration.

1.2 Problem Description

The front configuration of the vocational trucks varies depending on which vehicle it is placed on, Figure 1.1.

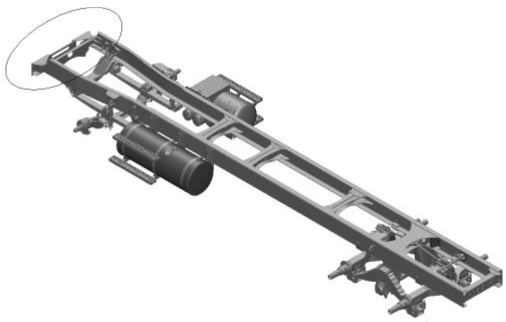


Figure 1.1 Typical Mack vocational truck frame system.

Truck front is marked with circle.

There are four basic options available today, shown in Figure 1.2:

1. Flush Bumper, Clevis Towing Device.
2. 5.5" Extended Bumper, Single Pin Towing Device.
3. 5.5" Extended Bumper, Dual Ring Towing Device.
4. 27" Extension, Dual Ring Towing Device, Provisions to mount a hydraulic pump.

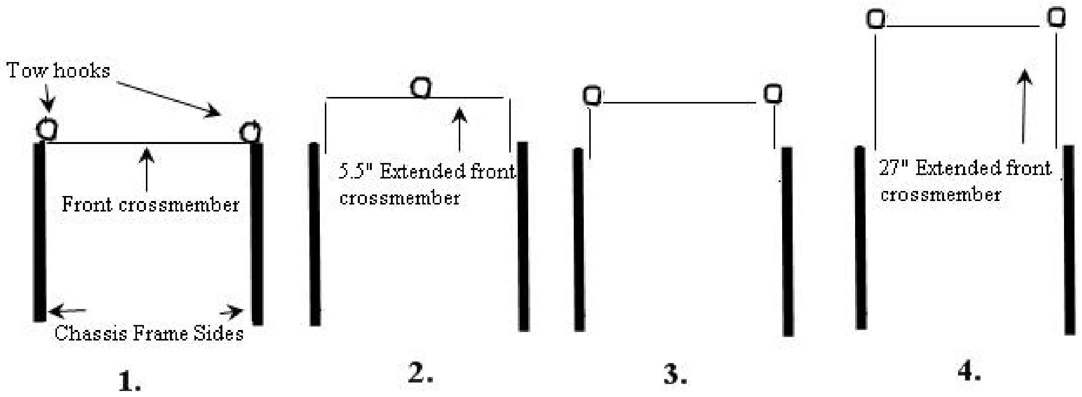


Figure 1.2 The four basic options for the front end of the chassis frame.

The emission laws for 2007 have led to the need of a larger cooling system to control the engine heat. This is what Mack referres to as the US07 project. The change in size affects the design of the surrounding parts, as for example the bolt interface of the connection between the front end configuration and the frame rails.

So far, Mack Trucks has used sheet metal solutions for the four front end options. Now, the company wants to examine the possibility to find a cast solution to the sheet metal parts used today. There are a number of issues that needs to be considered using cast material. One challenge is the shape of the structure to be able to manufacture the parts, another is the brittleness of the material.

1.3 Objective

The objective is to develop cast designs for the 5.5" front end extension for single and dual tow that can replace today's sheet metal solutions. A modular design is to prefer, i.e. interchangeable parts between the two options single and dual tow, but the large quantity of trucks sold with front-end extensions motivates separate solutions for the two options. The design shall:

- Maintain interfaces between frame and bumper
- Reduce cost compared to current fabrications
- Reduce weight compared to current fabrications
- Meet towing and structural requirements
- Improve assembly issues

Many steps of the product development chain are made. This project includes identifying the issues with the current design and its assembly issues, finding completely new design ideas some with the help of optimization tools, making FE analyses with given standard load cases, doing cost estimation and comparison with existing designs, and selecting a final solution.

1.4 Existing designs

There exist sheet metal designs for the US07 project which are not final but are planed to be ready to go into production January 2007. They are manufactured parts made of laser cut, bend and welded sheet metal. The thickness range from 6 to 15 mm. Seven fasteners on each side bolt on to the frame rails. Three 14 mm flange bolts attach to the bottom and four 20 mm Huck bolts attach to the side of the rail. On the dual tow, rings are used as tow interface. The single tow uses a pin, which can be lifted to fit the chain. Table 1.1 shows data for the existing designs.

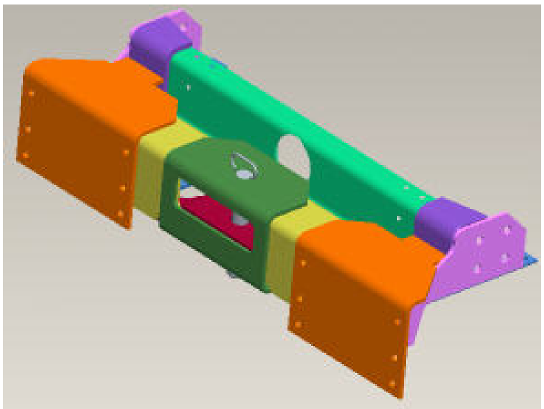


Figure 1.3 Existing design, single tow.

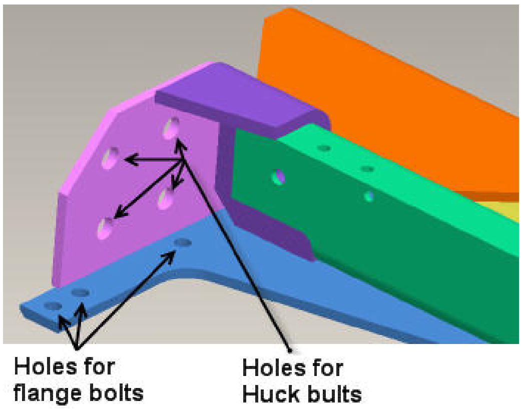


Figure 1.4 Seven bolts attach to the frame.

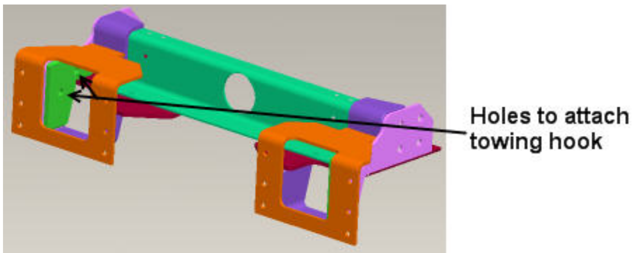


Figure 1.5 Existing design, dual tow.

Table 1.1

	Weight	Parts
Single tow	95.9 kg	13
Dual tow	64.2 kg	15

The dual tow has passed finite element analysis (FEA) with acceptable stresses for all load cases, Table 3-1. The single tow has not passed FEA, as seen in Figure 1.6. Notice that the scale range up to 1874 MPa. The material starts to yield at 355 MPa, which mean that in Figure 1.6 yellow, orange and red indicates yielded material.

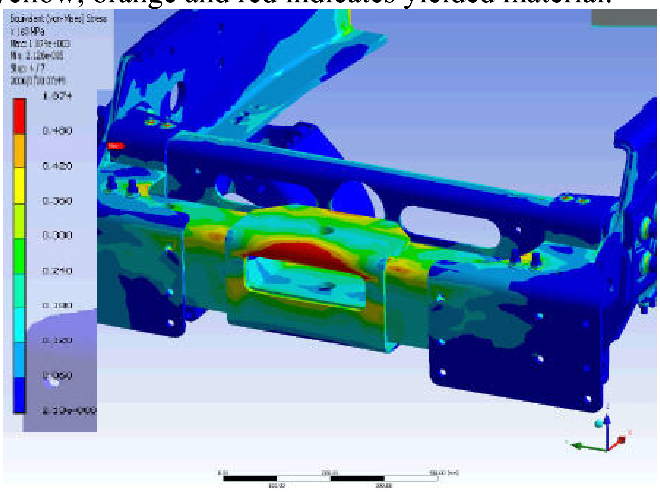


Figure 1.6. FEA of existing design, single tow.

Mack Trucks is currently modifying the sheet metals solution to make them withstand the stress criteria.

1.5 Delimitations

The project is limited to only find solutions to the 5.5” extension.

In all analyses the chassis frame has been considered stiff even though it is not in reality. No dynamic problems, such as frame rails movements causing vibrations and fatigue are included in the calculations. Neither is buckling treated. In reality, the start moment of the towing would give a snatch in the design. Only static load cases are considered. The analyses are made with the five static load cases defined in Mack Trucks procedure for front towing device test.

The project aims to find designs that stays below the yield limit of the material while loaded. Therefore only linear analyses are made. Depending on size and placement, most plastic regions are unacceptable. Exceptions are plasticity around boundary conditions and parts which Mack know from experience are reliable, for example the towing pin.

For the topology optimization the program Optistruct has been used for concept design. Further work with shape and size optimization has not been done. The problem was set up with an objective to maximize stiffness, given a certain percentage of volume to keep. It would have been possible to choose other objectives, for example minimize the stress or minimize displacement of some given points.

1.6 Disposition

After this problem description the company is introduced with a bit of history and an overview of Mack’s products. Then follows the problem specifications in chapter 3 and the theory that is relevant to the project and the computer programs that are used, chapter 4. How the designs were set up is treated in chapter 5 about computer modeling.

There has been several design ideas during this project, especially in the early weeks, and it is not possible to show all in detail in this report but some of them followed by the ones that was developed further are presented in chapter 6, with the final solution in chapter 7. Discussion and conclusion are in chapter 8 and 9.

In the appendix, the reader can find detailed computer analyses and different views of some of the later designs.

2 **Company Introduction**

A brief presentation of Mack Trucks and their assembly plants, LPP and Macungie. Mack Trucks product series Construction, Highway, Refuse Trucks and Power train is described with some extra details of the trucks that are using the bumper extensions.

2.1 **Mack Trucks' History**

At 1900 the three brothers Jack, Gus and William Mack started the company when they built and sold their first motorized vehicle. It was a sightseeing bus used in Brooklyn's Prospect Park, powered with a 24-horsepower engine. Five years later, the brothers began building trucks and moved their operation to an abandoned foundry in Allentown. The first trucks, with 50-horsepower engines, were built on the same bus chassis and shared the busses trade name, Manhattan.

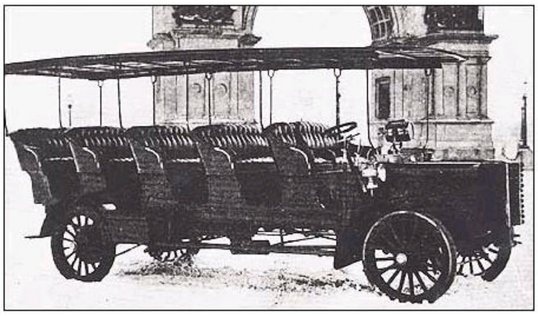


Figure 2.1 The 1900's sightseeing bus.[1]

1914, Mack became a leading truck manufacturer. The joint company was called International Motor Company (IMC) and had 700 employees in Allentown, 50 in Brooklyn, 75 in an engine plant in Newark.

Figure 2.2 The Mack Bulldog.[2]

In the early days of World War I, the British government had 150 Mack Trucks in their troops. They were so impressed with the ruggedness of the trucks that they gave them the nickname "Bulldog", after their beloved British bulldog (Figure 2.2). In 22 years of production, Mack built 40 299 Bulldog AC models, the very same that went to war. Later in World War II, Mack again supplied trucks and power trains for heavy trucks to the Allies. [1]

In 1922 the Bulldog symbol was fist used on a Mack truck. The image was placed on a sheet metal plate riveted to each side of the cab.[3]

Along with being one of the worlds leading truck manufacturers, Mack also has a long tradition of innovation in diesel engine and transmission technology. The company is the only U.S. Class 8[1] truck manufacturer to produce its own heavy-duty engines and transmissions. Mack dominates the construction and refuse markets (30% and 50% respectively). In the later years the company has focused on expanding on the market for highway trucks.

The company took the name, Mack Trucks, Inc., in 1922. Renault bought 10 percent share of Mack in 1979. Later, Mack became a wholly Renault owned subsidiary in 1990. Ten years later, Mack Truck Inc. became part of AB Volvo.[3]

The company has steadily evolved since the start over a hundred years ago. Today, the largest of Mack's own engines has 640 horsepower's, compared to the sightseeing bus' 24. Year 2005, Mack sold 27 300 trucks in the U.S.

Mack Trucks has mainly two different assembly plants that build the trucks. Lancaster Preferred Partners (LPP), and Macungie. Lancaster Preferred Partners, located in Pennsylvania has for many years been a business partner to Mack Trucks. It is in this assembly plant the chassis are put together. Of all chassis assembled at LPP 90% are Mack Trucks chassis. With this dependency, and thanks to the long partnership, LPP are willing to accommodate the special equipped chassis orders that Mack often has.There are several different subassemblies put together that later are attached to the chassis rails, for example the front axle installation, the wheel suspension and the front extension that this master thesis project involves. On the line, each chassis combination is different depending on what the costumer ordered. The subassemblies are heavy and need cranes to elevate and mount to the frame rails. Lighter subassemblies means easier and faster assembling.

All trucks from Mack except the Highway series get assembled in the Macungie plant located outside Allentown in Pennsylvania not far from Mack World Headquarter.It is a modern plant where the assembly line moves the chassis between the assembly stations but stand still ay each station for more parts to be mounted. At the end of the assembly line complete trucks drive off. There is also a workshop were custom trucks are altered or built from scratch according to customer needs.

[1] U.S. Class 8: Vehicles are classified depending on the Gross Vehicle Weight Rating (GVWR) meaning, "the maximum total vehicle rated capacity, measured at the tire ground interface, as rated by the manufacturer"[4]. Class 8 is the highest class with GVWR higher than 33 000 lbs (14 800kg).

2.2 Mack Truck's Products

Mack products are divided into four major groups; Construction, Highway, Refuse Trucks and Power train.

The Construction series consists of CL, MR and different versions of Granite. Trucks in these series are mostly used at construction sites and other off road conditions were durability, maneuverability and high payloads are important. The Highway series consist of Pinnacle, Vision, CH and Rawhide. All trucks in the series are tractors and come either as a day cab or sleepers of different length. They are made for long distance travels and comfort, fuel economy and productivity is important. The Refuse series consist of MR, LE and Granite. MR and Granite are the same as in the construction series but used as a refuse truck. The LE stands for low entry and is a special version of MR which has an extra low cab. Price, maneuverability and good "stop and go" performance is important. Mack power train delivers electronics, engines, transmissions and axels for heavy trucks.

2.3 Trucks using Bumper Extension

The Granite series is as mentioned vocational trucks often used in construction sites. This means it works in rugged terrain with a lot of starts and stops. The granite series is known for its durability and tough design. The choices of engine range from around 300 to 500 HP. The truck can be delivered with a number of different transmissions, manual or automatic in the range of 5 to 18 gears. The Granite trucks also have two options of placement of the front wheel axle. Standard is axle front. In the Axle Back option the front axle is places further back to get a smaller turning radius. It can also be fitted with a central inflation system for more serous off road duty. Granite trucks are often used as mixers (with a cement mixer in the back) or dump trucks to carry loads of stone or sand.

The CL series is Mack's most heavy duty truck and used both as a tractor and as a conventional truck for both on and off road duties. The CL series is delivered with Cummins engines in the range from 500 to 565 HP and transmissions in the range 9 to 18 gears. It is often used as dump trucks and for logging.

The Granite Series and the CL series uses bumper extensions to give the truck a more personal look and protect the front of the hood better. Bumper extensions have the options shown in Figure 1.2: The standard bumper is a flush bumper that uses two steel loops that attach to the frame rails for towing. There are two options which ad 5.5 inch to the length of the truck, one with a single pin and one with two rings to attach the towing chain. The granite series also has a 27 inch extension as an option; this is used to carry a cement pump in the front when the truck is equipped as a mixer.

3 Problem Specifications

The limitations and specifications the designs needs to keep within and fulfill. The chapter starts with the towing and hood loads, followed by the surrounding interface and possible material. Comments from LPP about the problem of the existing design are stated in the last part.

3.1 Loads

There are two groups of loads that are considered; the TMC's Towing Loads and the Hood Loads.

3.1.1 TMC's Towing Loads

The organization American Trucking Associations, ATA represent the interests of the trucking industry. It is made up of three separate entities: ATA, representing the national interests; the 50 affiliated state trucking associations, representing state and local interests; the affiliated councils and conferences, representing specialized areas of the trucking industry. TMC, The Maintenance Council of ATA, states a recommended practice to test the towing device on a truck. TMC:s Recommended Practice is not a legal requirement, but because of ATA:s strong influence on the truck market Mack Trucks has decided to follow the stated guidelines with an additional 10% of the loads. [5]

In the Recommend Practice a cone represents five static load cases used to test the towing device; Forward, Up 45º, Left side 45º, Right side 45º and Vertical pull. Figure 3.1 shows the load cases for single tow and Figure 3.2 for dual tow device.

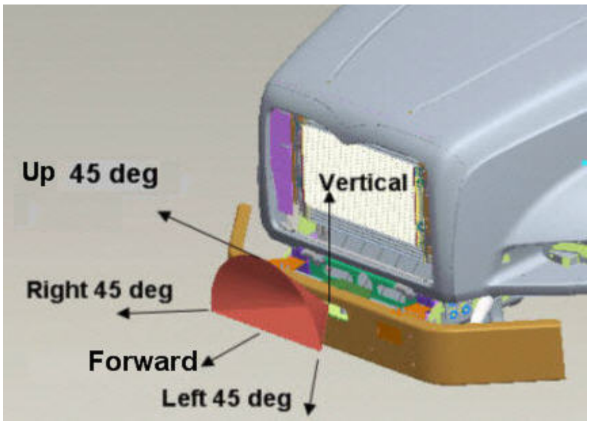


Figure 3.1 Cone with the five load cases for single tow device.

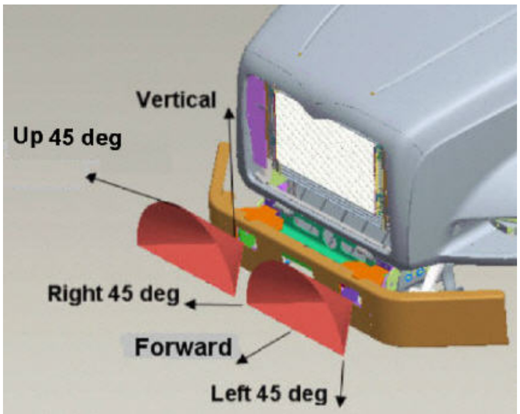


Figure 3.2 Cone with the five load cases for dual tow device.

The load cases are defined from either Front/Rear Axle GAWR or GVWR. Front/Rear Axle Gross Axle Weight Rating, GAWR, is the load-carrying capacity of a front/rear axle as measured at the tire-roadway interface. Gross Vehicle Weight rating, GVWR, is the maximum total vehicle rated capacity, measured at the tire ground interface, as rated by the chassis manufacturer.

Mack Trucks has decided to use TMC standard for a 3 axle truck with a Front Axle GAWR rating of 23 000 lb and Rear Axel GAWR rating of 65 000 lb, i.e. GVWR rating of 88 000 lb, +10%.

The loads are defined as:

Forward Pull
Dual attachment: 50% of GVWR for each device
Single attachment: 100% of GVWR

Cone Left, Right and Up 45º
Dual attachment: 70% of Front Axle GAWR for each device
Single attachment: 140% of Front Axle GAWR

Vertical Pull
Dual attachment: 50% of Front Axle GAWR for each device
Single attachment: 100% of Front Axle GAWR

Table 3-1 presents the resulting loads for a single and a dual towing device.

Table 3-1 Tow loads for dual and single attachments.

Front Towing Device Test Loads				
	Dual attachment		**Single attachment**	
Load case:	**TMC**	**+10%**	**TMC**	**+10%**
Forward	195 720 N	215 371 N	391 440 N	430 742 N
Left 45º	71 616 N	78 806 N	143 232 N	157 612 N
Right 45º	71 616 N	78 806 N	143 232 N	157 612 N
Up 45º	71 616 N	78 806 N	143 232 N	157 612 N
Vertical 90º	51 155 N	56 290 N	102 310 N	112 580 N

In this report, only the five load cases listed above are used to test the design solutions. The load cases are used as a standard test procedure even if it would be possible to test how other load cases within the cone affects the towing device, for example Up 15º with a load value somewhere between the value of Forward and Up 45º. There is an infinitive number of possible combinations this could add up to, and the question is what load value that would be correct for each direction. Depending on the chosen value some might show less acceptable results than the five cases listed above. However, Mack Truck has chosen to follow the stated five load cases of the TMC:s Recommended Practice with an additional 10%.

3.1.2 Hood Load

The hood of the truck mounts to the front end extension. It is approximated that the front end carries 80% of the weight of the hood that weighs 73 kg (160 lb). Measured acceleration forces on the hood are listed in Table 3.2 with the resulting force in N.

Table 3.2 Force from 73 kg hood on the front end extension.

Hood Load on Front End Extension		
	Acc	Force
Vertical	+5.25 g -5.10 g	+4467 N -2901 N
Longitudinal	+1.96 g -2.10 g	+1395 N -1495 N
Transverse	+4.55 g -4.80 g	+3239 N -3417 N

3.2 Truck Front Interface

There are a number of surrounding components limiting the available space to place the front-end extension such as the frame, the bumper, the cooling package and the spring hanger. The design needs to give space for tools to be able to assembly the bumper extension to the chassis.

The bumper limits the available space forward, upwards and downwards which is marked by arrows in Figure 3.3c. There are two bumper options available today, standard and stylized bumper (Figure 3.4 and Figure 3.3). The front end extension needs to fit both, or at least be adjustable to fit both.

The standard bumper has either one hole in the middle for the single towing device or two holes on each side for the dual towing. It attaches to the front end extension with 6 bolts on each side, marked with red rings in Figure 3.3.

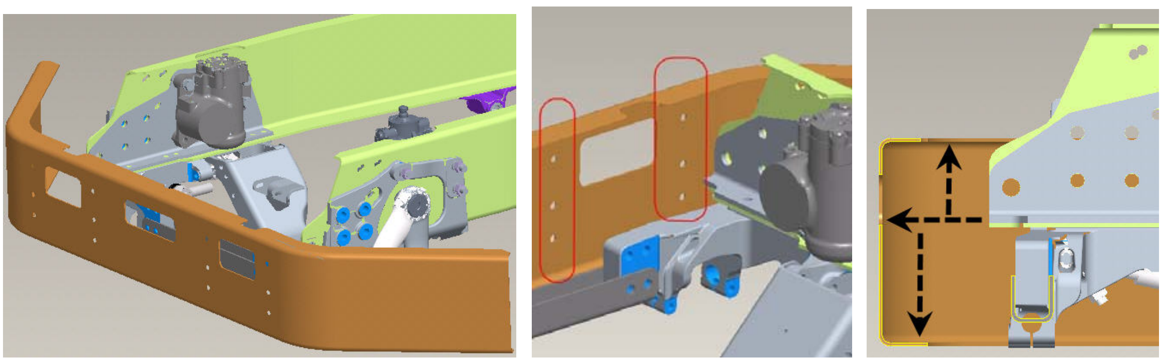


Figure 3.3 Front, inside and side view of the bumper, here shown with holes for both single and dual tow. The bumper attaches to the front end extension with the six bolt holes on each side.

The stylized bumper is attached in a different way. It has three bolts on each side. The lower part of the bumper is bent so that the last hole is in an angle. The fourth hole, indicated with a red arrow in Figure 3.4, is only for keeping the bumper in place on the assembly line with the help of a pin. The hole is left empty without a bolt after the bumper is mounted.

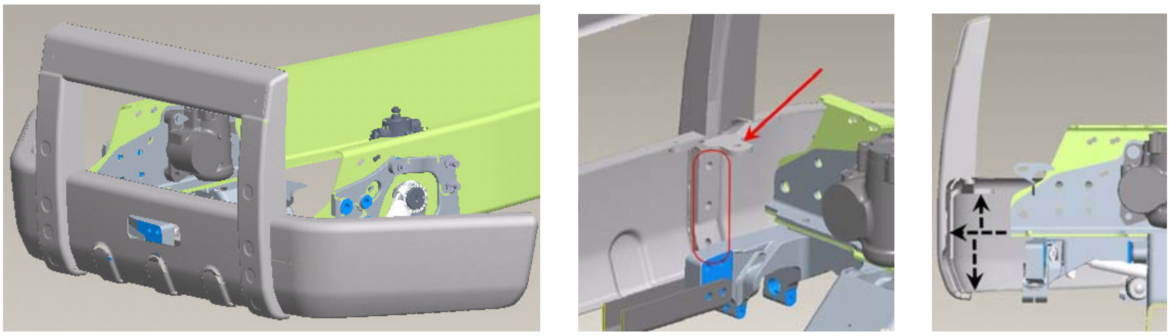


Figure 3.4 Front, inside and side view of the stylized bumper. Three bolt holes on each side.

Other limitations are the spring hanger and the two hood mounts. The spring hanger consists of three main parts that block the available space downwards, shown in Figure 3.5. It is attached sharing the same bolt interface on the front of the frame rail that the front extension will be using. The front end extension needs to include a part where the hood mounts can attach. The interface with three bolts shall be kept, Figure 3.6.

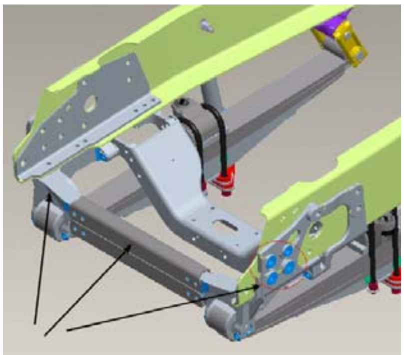


Figure 3.5 The spring hanger. Bolt interface marked with circle.

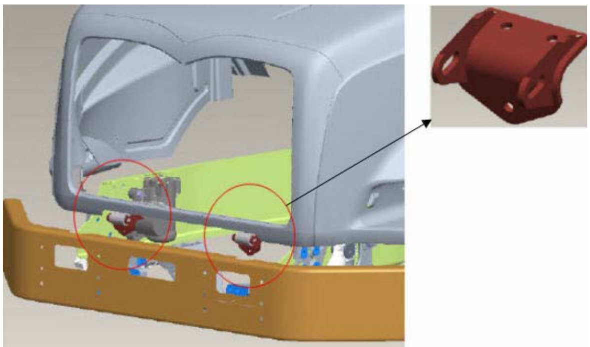


Figure 3.6 The two hood mounts marked with circles.

The design needs to fit both the vehicle types Axle Forward and Axle Back. It is the Axle Forward that is limiting the space around the bolt interface of the frame because of the spring hanger that is placed closer to the front and is using the same bolt interface to the frame as the front end extension.

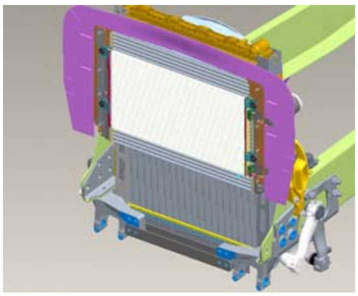


Figure 3.7 Cooling package for 2007.

An important difference from the earlier trucks is the cooling package. For the engine to handle the emission restrictions for 2007, the cooling package needs to be bigger than before. This decreases the space around the frame interface, Figure 3.7.

The stone guard plates are cut for each vehicle depending on the customer's choice of bumper and front end extension. The design guideline is that there should not be bolt heads sticking up from the front end extension that can not be covered by the stone guard. The stone guard needs to fit in under the edge of the bumper. This limits the space upwards.

3.3 Materials

Two different cast materials have been considered. Both are spherodial graphite iron but with different properties, Table 3.3 and Table 3.4. Note that values within brackets are estimations and not binding to the supplier.

Table 3.3: Properties for spherodial graphite iron 0722

Spherodial graphite iron 0722 (nodular iron)	
Volvo standard:	STD 1107,22
American standard:	ASTM A536 -84, grade 65-45-12
Swedish standard:	SS 14 07 22
$R_{p0.2}$: 310-(420) N/mm^2	$R_{cp0.2}$: 370-(500) N/mm^2 [6]
R_m: 450-(620) N/mm^2	R_{cm}: 800-(1100) N/mm^2
A_5: 10-(20) %	

Table 3.4: Properties for spherodial graphite iron 0737

Spherodial graphite iron 0737 (nodular iron)	
Volvo standard:	STD 1107,37
American standard:	ASTM A536 -84, grade 100-70-03
Swedish standard:	SS 14 07 37
$R_{p0.2}$: 440-(570) N/mm^2	R_m: 700-(960) N/mm^2
R_{cm}: 1100-(1300) N/mm^2	A_5: 3-(7) %

The earlier problem with using casting in the front bumper extension has been the lack of ductility in cast iron. After discussions with the chassis structure group we decided to use the spherodial graphite iron 0722 because of its greater ductility.

For the sheet metal plates Volvo standard structural steel is used.

Table 3.5: Properties for structural steel 2132

Steel 2132	
Volvo standard:	STD 1121,32
American standard:	ASTM A572, grade 50, class S91
Swedish standard:	SS-EN 10 210-1
$R_{p0.2}$(t<16mm): 355 N/mm	R_m(T>0°C): 510-(680) N/mm
A_5: 22%	R_m(T>-20°C): 470-(630) N/mm

Where:

$R_{p0.2}$= Tensile yield strength (non-proportional elongation, here as 0.2% limit)

$R_{cp0.2}$= Compressive yield strength

R_m= Ultimate tensile strength

R_{cm}= Ultimate compressive strength

A_5= Elongation at fracture when $L_0 = 5\sqrt{\frac{4S_0}{\pi}}$

S_0= Minimum original cross-sectional area of test piece.

t= Thickness of sheet metal plate

T= Temperature

3.4 Assembly Issues

Listed below are comments from the assembly plant LPP about the mounting of today's version of the front end extension that is in production.[7] The comments are of great importance to be able to understand the problems and preferences and then design something that can be manufactured and assembled easier in reality.

- Today the front extension is sub assembled into one big module. The workers have problems getting the front extension in place.

- Mounting today's version took 7 workers 12 minutes to get in place. The time will probably decrease considering this was one of the first times the workers had to mount this design, but the time was still considered a problem.

- LPP prefer a one-piece subassembly instead of several parts. Many parts would probably increase the assembly time.

- The sub assembled front end keeps the frame width stabile after it is mounted.

- Today's version of the front end extension has 7 bolts to fasten in each of the two frame rails. Fewer bolts would make the mounting easier.

4 Theory

Creating a design that fulfills the objective involves many engineering areas. In this chapter the basics of the, for this project, most important theories are explained.

4.1 Strain and stress

For further details about strain and stress see [8]. When creating a design it is of great importance to make sure that the structure withstand against fracture or permanent deformations due to loading. The most convenient representation of structural behavior is through the theory of stress and strain. Strain (ε) is defined as relative deformation. Stress (σ) is defined as internal force per unit area of a given cross section. Figure 4.1 shows the one-dimensional case.

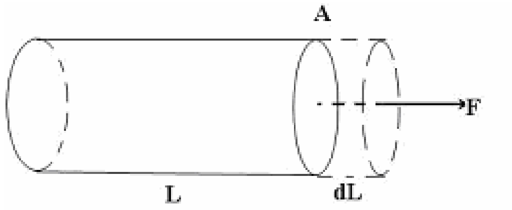


Figure 4.1 Strain in one dimension.

For the one dimensional case, stress and strain are defined as:

$$\sigma = \frac{F}{A} \qquad \varepsilon = \frac{L + dL}{L} \tag{4.1}$$

For the three-dimensional case strains can be generalized to:

$$\varepsilon_{ij} = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{4.2}$$

In (4.3) u_i represents the displacement field. Since $\varepsilon_{ij}=\varepsilon_{ji}$ the strain tensor involves six independent variables that in matrix form can be written as:

$$\boldsymbol{\varepsilon} = \begin{bmatrix} \varepsilon_{11} & \varepsilon_{12} & \varepsilon_{13} \\ \varepsilon_{12} & \varepsilon_{22} & \varepsilon_{23} \\ \varepsilon_{13} & \varepsilon_{23} & \varepsilon_{33} \end{bmatrix} \tag{4.3}$$

Imagine a body with a unit vector **n** normal to the surface and directed out of the body. Further, imagine an incremental force vector Δ**P** acting on the incremental surface Δ**A**. If Δ**A** approaches zero the ratio Δ**P**/Δ**A** approaches a value **t**. **t** is called the traction vector ans can be expressed as $\mathbf{t}^T=[t_1\ t_2\ t_3]$ in matrix format. The dimension of **t** is N/m^2.

The traction vector **t** is related to a surface with the outer unit normal vector **n**. In general, the traction vector will look different when different sections through the same point are considered.

$$\mathbf{t}_1^{\mathrm{T}} = \begin{bmatrix}\sigma_{11} & \sigma_{12} & \sigma_{13}\end{bmatrix} \tag{4.4}$$

With sections made perpendicular to the coordinate axes, Figure 4.2, the stress tensor can be defined. It contains the traction vectors and therefore all the information needed to describe the stress state for a particular point.

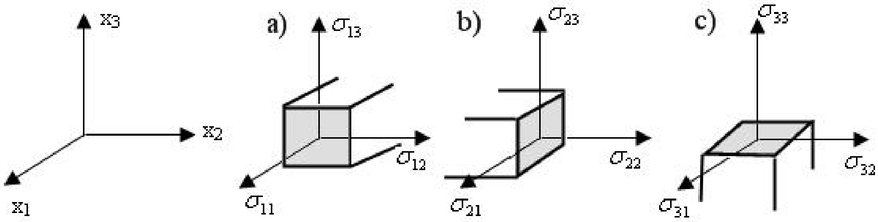


Figure 4.2 Stress components for a point.

$$\begin{bmatrix}\sigma_{ij}\end{bmatrix} = \begin{bmatrix}\mathbf{t}_1^{\mathrm{T}} \\ \mathbf{t}_2^{\mathrm{T}} \\ \mathbf{t}_3^{\mathrm{T}}\end{bmatrix} = \begin{bmatrix}\sigma_{11} & \sigma_{12} & \sigma_{13} \\ \sigma_{21} & \sigma_{22} & \sigma_{23} \\ \sigma_{31} & \sigma_{32} & \sigma_{33}\end{bmatrix} \tag{4.5}$$

It can be shown that the stress tensor is symmetric.

For a special choice of coordinate system the stress tensor takes a very simple form. By solving the eigenvalue problem with the characteristic equation

$$\det(\boldsymbol{\sigma}\text{-}\lambda\mathbf{I})=0 \tag{4.6}$$

three solutions are obtained. They corresponds to the principal stresses $\sigma_1 = \lambda_1$, $\sigma_2 = \lambda_2$ and $\sigma_3 = \lambda_3$. Each λ-value provides the corresponding principal directions $\mathbf{n_1}$, $\mathbf{n_2}$ and $\mathbf{n_3}$ by solving the equation:

$$(\boldsymbol{\sigma}\text{-}\lambda\mathbf{I})\mathbf{n}=\mathbf{0} \tag{4.7}$$

If the coordinate system is taken collinear with the principal directions the stress tensor takes the simple form:

$$\boldsymbol{\sigma}' = \mathbf{n}\boldsymbol{\sigma}\mathbf{n}^{\mathrm{T}} = \begin{bmatrix}\sigma_1 & 0 & 0 \\ 0 & \sigma_2 & 0 \\ 0 & 0 & \sigma_3\end{bmatrix} \quad \text{where} \qquad \mathbf{n}^{\mathrm{T}} = [\mathbf{n}_1 \quad \mathbf{n}_2 \quad \mathbf{n}_3] \tag{4.8}$$

The principal stresses are invariants and the principal directions are always orthogonal. The stress tensor gives the generic stress invariants:

$$I_1 = \sigma_{ii}$$
$$I_2 = \frac{1}{2}\sigma_{ij}\sigma_{ji} \qquad (4.9)$$
$$I_3 = \frac{1}{3}\sigma_{ij}\sigma_{jk}\sigma_{ki}$$

I_1 is referred to as hydrostatic stress. Further, the stress deviator tensor can be defined as:

$$s_{ij} = \sigma_{ij} - \frac{1}{3}\sigma_{kk}\delta_{ij} \qquad (4.10)$$

The σ_{ij}- and s_{ij}-tensors have identical off-diagonal elements and identical differences between the diagonal elements and therefore have identical principal directions. The generic invariants of the stress deviator tensor are given by:

$$J_1 = s_{ii} = 0$$
$$J_2 = \frac{1}{2}s_{ij}s_{ji} \qquad (4.11)$$
$$J_3 = \frac{1}{3}s_{ij}s_{jk}s_{ki}$$

Assuming Hooke's law can apply the relation between the stress and the strain tensor can be expressed using a 4th order tensor, D_{ijkl}, as follows:

$$\sigma_{ij} = D_{ijkl}\varepsilon_{kl} \qquad (4.12)$$

In this project only material with the same properties in all directions, i.e. isotropic material, is used. For this type of material the D-matrix takes the form:

$$D = \frac{E}{(1+v)(1-2v)}\begin{bmatrix} 1-v & v & v & 0 & 0 & 0 \\ v & 1-v & v & 0 & 0 & 0 \\ v & v & 1-v & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{2}(1-2v) & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{2}(1-2v) & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1}{2}(1-2v) \end{bmatrix} \qquad (4.13)$$

E is the elasticity modulus and v is Poisson's ratio.

4.2 **Initial Yield Criteria**

The effect of an applied external force will have on the structural behavior depends on the design shape and material properties. When a structure is loaded in tension or compression it starts to deform. Depending on the magnitude of the load, the deformation can be elastic or plastic. If a subject only to elastic deformation it returns to the original shape when the load is removed, if it is subject to plastic deformation some deformation is permanent. It is the yield limit of the material that decides when deformation becomes plastic.

The diagram in Figure 4.3 and Figure 4.4 shows the stress-strain relationship for typically ductile and brittle materials. A brittle material, such as concrete or glass cannot be deformed plastically as the curve implies.

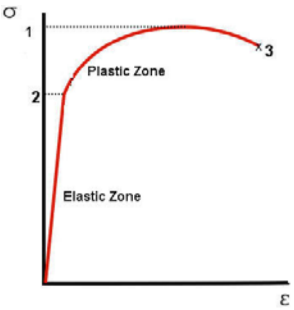


Figure 4.3 Ductile material [9].

1. Ultimate Strength
2. Yield strength
3. Rupture

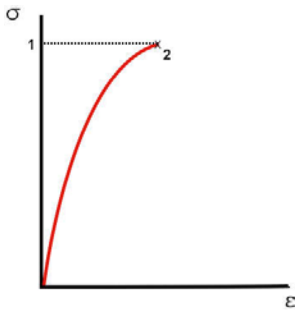


Figure 4.4 Brittle material [9].

1. Ultimate Strength
2. Rupture

When a structure is loaded in only one direction, so that the stress state is uniaxial, the stress and the strength can be compared directly to determine whether the part will fail. The method is simple, since there is only one value of stress to compare with one value of strength, for example the yield strength or the ultimate strength. But it becomes more complicated when the stress state is biaxial or triaxial. Only one strength to compare with and a multitude of stresses.

For isotropic material stress state can be described by the principal stresses described in the previous chapter 4.1. Also, theories for failure or initial yielding can be expressed with them but to avoid solving an eigenvalue problem the theories can more advantageously be expressed with the help of the invariants. As an example a general yield or failure criterion expressed with:

$$F(I_1, J_2, \cos 3\theta) = 0$$

$$\text{where:} \quad I_1 = \sigma_{ii} \qquad J_2 = \frac{1}{2} s_{ij} s_{ji} \qquad \cos 3\theta = \frac{3\sqrt{3}}{2} \frac{\frac{1}{3} s_{ij} s_{jk} s_{ki}}{J_2^{3/2}} \tag{4.14}$$

This formulation separates the influence of the hydrostatic stress I_1 and the deviatoric stresses expressed in J_2 and cos3θ.

To be able to analyze a general three-dimensional stress condition a number of failure or yield theories have been proposed. Next, a couple of them will be presented and compared, followed by a discussion of which theory that would be suitable in this project.

4.2.1 **Maximal Normal Stress Theory**

The Maximal Normal Stress Theory states that *yielding occurs whenever one of the three principal stresses equals the yield strength*. The principal stresses σ_1, σ_2 and σ_3 are ordered as:

$$\sigma_1 > \sigma_2 > \sigma_3 \qquad (4.15)$$

This means that this theory predicts failure when:

$$\sigma_1 \geq \sigma_{yt} \quad \text{or} \quad \sigma_3 \leq -\sigma_{yc} \qquad (4.16)$$

σ_{yt} and σ_{yc} are tensile and compressive yield strength. [9]

4.2.2 **Von Mises Stress Theory**

The equations and figures in this chapter are described in further detail in [9]. The von Mises Stress Theory is also known as Maximum Distortion Energy Theory or the Octahedral-Shear-Stress Theory. It originated because of the observation that ductile materials stressed hydrostatically showed yield strengths well beyond the limit given by the simple tension test. It was assumed that yielding was not a simple tensile of compressive phenomenon; instead it was related somehow to the angular distortion of the stressed element.

Consider a unit volume subjected to a three-dimensional stress state of the stresses σ_1 ,σ_2 and σ_3, Figure 4.5 (a). The stress state can be separated into σ_{av} and a remaining part. Some of the following equations will also be presented in index notation for reference. σ_{av} is the mean value, derived from:

$$\sigma_{av} = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3} \qquad \sigma_{av} = \frac{1}{3}\sigma_{kk} \qquad s_{ij} = \alpha_{ij} - \delta_{ij}\sigma_{av} \qquad (4.17)$$

This volume has a pure volume change. Since σ_{av} is a component of σ_1, σ_2 and σ_3 it can be subtracted from them, resulting in a pure angular distortion, Figure 4.5 (c).

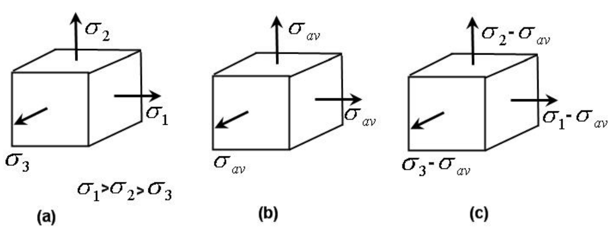


Figure 4.5 (a) Triaxial stresses. The unit volume undergoes both volume change and angular distortion. (b) Unit volume under hydrostatic tension undergoes only volume change. (c) Unit volume only with angular distortion without volume change.

The total strain energy for the element in (a) is given by the equation:

$$u_\sigma = \frac{\varepsilon_1\sigma_1}{2} + \frac{\varepsilon_2\sigma_2}{2} + \frac{\varepsilon_3\sigma_3}{2} = \frac{1}{2E}\left[\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2v(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)\right]$$

$$u_\sigma = \frac{\varepsilon_{ij} D_{ijkl} \varepsilon_{kl}}{2} \quad (4.18)$$

E is the elastic modulus and *v* is Poisson's ratio.

The strain energy for producing only volume change (b) can be obtained by substituting σ_{av} for σ_1, σ_2 and σ_3 in the equation for the total strain energy (4.17). This gives the result:

$$u_v = \frac{3\sigma_{av}^2}{2E}(1-2v)$$

$$u_v = \frac{\sigma_{ij}\varepsilon_{ij}}{2} \quad (4.19)$$

To get the angular distortion energy u_d of (c), u_v in equation (4.18) is simply is subtracted from u_σ (4.17):

$$u_d = u_\sigma - u_v = \frac{1+v}{3E}\left[\frac{(\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_3-\sigma_1)^2}{2}\right]$$

$$u_d = \frac{s_{ij}\varepsilon_{ij}}{2} \quad (4.20)$$

If $\sigma_1=\sigma_2=\sigma_3$ the distortion energy is zero, as in case (b). Using this result as a yield criterion the von Mises Theory predicts that *yielding will occur whenever the distortion energy in a unit volume equals the distortion energy in the same volume when uniaxially stressed to the yield strength*. For a simple tension test $\sigma_1= \sigma'$, $\sigma_2=\sigma_3=0$. The distortion energy is:

$$u_d = \frac{1+v}{3E}\sigma'^2 \quad (4.21)$$

Setting equation (4.19) and (4.20) equal to each other gives:

$$\sigma' = \sqrt{\frac{(\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_3-\sigma_1)^2}{2}} \quad (4.22)$$

Looking back at the generic invariant J_2 of the stress deviator tensor, the expression in (4.21) can be written as $\sqrt{3J_2}$. Yielding is predicted to occur when $\sqrt{3J_2} - \sigma_{yield}>0$. Important to notice is that the criterion is independent of the hydrostatic stress I_1, only the deviatoric stresses influences. The von Mises theory does not take notice if it is load in tension or compression.

4.2.3 Drucker-Prager Theory

The Drucker-Prager criterion involves the hydrostatic stress I_1 and is expressed:

$$\sqrt{3J_2} + \alpha I_1 - \beta = 0 \tag{4.23}$$

α and β are positive material parameters. Since Drucker-Prager involves the hydrostatic stress term it takes into account if the load is in tension or compression. If α is zero, the expression is reduces to von Mises criterion.[8]

4.2.4 Mohr and Mohr-Coulomb Theory

The following chapter is based on Mischke and Shigley -89, for further detail see [9]. The Mohr or the Mohr-Coulomb Theory (also called Internal-Friction Theory) is used to predict failure for materials whose strength in tension and compression are not equal. For example, gray cast iron can have compression strength up to 3 or 4 times grater than the tensile strength. The Mohr Theory predicts failure only on basis of the largest of the three principal shear stresses that are defined as:

$$\tau_{12} = \frac{\sigma_1 - \sigma_2}{2} \qquad \tau_{23} = \frac{\sigma_2 - \sigma_3}{2} \qquad \tau_{13} = \frac{\sigma_1 - \sigma_3}{2} \tag{4.24}$$

When the principal stresses are ordered $\sigma_1 > \sigma_2 > \sigma_3$ it is τ_{13} that is the largest, as also can be seen in Mohr's circle, Figure 4.6.

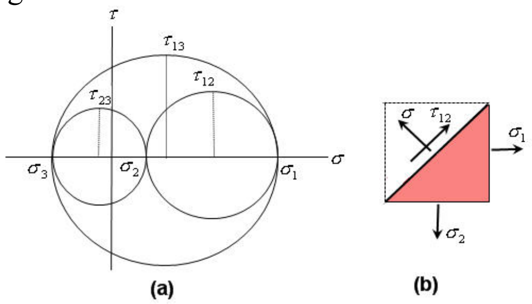


Figure 4.6 (a) Mohr's circle for triaxial stress with three principal normal stresses and three principal shear stresses. (b) Each principal shear stress occurs in two planes, one shown here.

Mohr's Theory can be shown with Figure 4.7. One circle represents the strength σ_{yc} from a uniaxial compression test, one represents the strength σ_{yt} from a uniaxial tension test, and the third central circle is from a yield test of pure shear. The theory predicts that *failure occurs for any other stress state in which the largest of the three Mohr's circles corresponding to σ_1, σ_2 and σ_3 and is tangent to the line AE.*

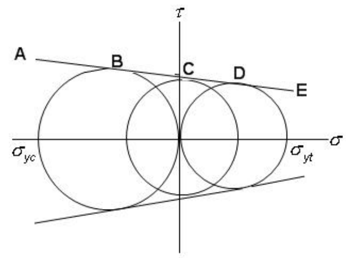


Figure 4.7 Three Mohr's circles.

The variation of this theory is Mohr-Coulomb. It is based on the assumption that the line BCD is straight. The principal normal stresses are ordered $\sigma_1>\sigma_2>\sigma_3$. Then, for any stress state producing a circle tangent to line BCD, between B and D, σ_1 and σ_3 has opposite signs. For this stress state the Mohr Theory applies and the two stresses and the strengths are related by the equation:

$$\frac{\sigma_1}{\sigma_{yt}} - \frac{\sigma_3}{\sigma_{yc}} = 1 \qquad \sigma_1 \geq 0 \,,\ \sigma_3 \leq 0 \tag{4.25}$$

A side note is that in geology, the Coulomb-Mohr Theory is often used to define shear strength of soils at different effective stresses.

4.2.5 Comparison of failure theories

Which failure theory is best suited to use for the analyses in this report?

The von Mises Theory is widely used to predict yielding of ductile materials, especially metals. The von Mises stress gives an absolute value and it is not possible to see whether there is tension or compression in a high stress area. This implies that the method is not to prefer when analyzing materials with large difference in yield strength for tension and compression, which is often the case with brittle materials. In the Drucker-Prager criterion there is a term that accounts for the hydrostatic stress, i.e. makes a difference for tensile or compressive load. The other criterion such as Maximal Normal Stress Theory, Mohr or Mohr-Coulomb Theory also separates tensile and compressive load. All of them would be more suited for brittle materials such as concrete, soil or rocks. The Maximum Normal Theory gives reasonably accurate predictions of failure in brittle materials as long as the normal stress has the largest absolute value in tensile. But, if the largest absolute value of the normal stress is in compression there are deviations from the criteria.

For the cast design in this report the spherodial graphite iron 0722 is used (material properties in chapter 3.3). To be able to choose failure criteria it has to be decided whether the material is ductile of brittle. The material is classified as Ductile Iron. To say that a material is brittle or ductile is a relative definition and there is no set limit that separates the two terms. Ductile iron has 20 % higher value in compressive yield stress than tensile yield stress, 310 MPa compared to 370 MPa.

For analysis with cast material Mack Trucks often use von Mises. If there are problem areas that need to be looked at further, Maximal Normal Stress can be checked. For the chosen ductile iron there is a difference between the tensile and compressive yield strength and it would have been more suitable to use for example Drucker-Prager that takes it into account. Because of Mack standards, decision has been taken to still use von Mises in general in the computer analyses of this project, but for the final solution the von Mises results will be compared with Maximal Normal Stress to see the difference in tension and compression. As will be seen later in the report some solutions include sheet metal plates. For them, it is motivated to only use von Mises as the material for the plates are clearly ductile.

4.3 Finite Element Method

For more detailed information about the finite element method see [10]. It is a numerical method for solving physical problems generated by differential equations. The differential equations can describe for example mechanical problems or heat flows. To solve a complex mechanical problem the structure is divided into many small parts or elements. Even though the structure is complex the small elements can be described by simple equations. This results in a system of equations that is usually presented in the following way (standard FE-formulation):

$$\mathbf{Ka} = \mathbf{f} \qquad (4.26)$$

where $\mathbf{K}$ is the stiffness matrix, $\mathbf{f}$ is the force vector and $\mathbf{a}$ is the result vector.

The force vector includes loads, boundary conditions and initial strains. For mechanical problems the stiffness matrix includes the geometry and material property of the structure. Changes in geometry due to loads and changes in material data due to nonlinear elastic or yielding material are not taken into account when doing a linear analysis. The stiffness matrix is symmetric. The result vector is the unknown which is calculated and then used to determine the stresses, strains and displacements for the whole structure.

4.3.1 3-D elasticity with isotropic material

When acceleration and inertia of mass is not taken into consideration the principle of virtual work can be formulated as:

$$\int_V \boldsymbol{\sigma}\delta\boldsymbol{\varepsilon}^{w}\, dv = \int_S \mathbf{tw}\, ds \qquad (4.27)$$

The virtual work inside the volume (V) is equal to the work carried on the boundary (S). $\boldsymbol{\sigma}$ is the stress vector, $\boldsymbol{\varepsilon}^{w}$ is the virtual strain vector (4.29), $\mathbf{w}$ is the weight function (4.28) and $\mathbf{t}$ is the traction vector. The weight function is defined as:

$$\mathbf{w}=\mathbf{Nc} \qquad (4.28)$$

where $\mathbf{N}$ is a matrix containing the shape functions and $\mathbf{c}$ is arbitrary. $\delta\boldsymbol{\varepsilon}^{w}$ can be expressed as:

$$\delta\boldsymbol{\varepsilon}^{w}=\frac{1}{2}\left(\frac{\partial \mathbf{w}}{\partial \mathbf{x}}+\left(\frac{\partial \mathbf{w}}{\partial \mathbf{x}}\right)^{T}\right) \qquad (4.29)$$

Since $\mathbf{B}$ is the derivative of $\mathbf{N}$ with respect to $\mathbf{x}$ (4.29) turns into:

$$\delta\boldsymbol{\varepsilon}^{w} = \mathbf{Bc} \qquad (4.30)$$

$\mathbf{c}$ is not dependent on V or S so inserting (4.30) and (4.28) into (4.27) gives:

$$\mathbf{c}\left(\int_V \mathbf{B}^{T}\boldsymbol{\sigma}\, dv - \int_S \mathbf{N}^{T}\mathbf{t}\, ds\right)=\mathbf{0} \qquad (4.31)$$

Since **c** is arbitrary (4.31) can be expressed as:

$$\int_V \mathbf{B}^T\boldsymbol{\sigma}\, dv = \int_S \mathbf{N}^T\mathbf{t}\, ds \qquad (4.32)$$

σ is can be expressed as a function of **ε** using **D** from chapter 4.1 equation (4.13):

$$\boldsymbol{\sigma} = \mathbf{D}\boldsymbol{\varepsilon} - \mathbf{D}\boldsymbol{\varepsilon}_0 \qquad (4.33)$$

where **ε** is the total strains and $\boldsymbol{\varepsilon}_0$ is the initial strains. The total strains:

$$\boldsymbol{\varepsilon} = \frac{1}{2}\left(\frac{\partial \mathbf{N}}{\partial \mathbf{x}} + \left(\frac{\partial \mathbf{N}}{\partial \mathbf{x}}\right)^T\right) \qquad (4.34)$$

No initial strains ($\boldsymbol{\varepsilon}_0$=0) combined with (4.33) and (4.34) gives:

$$\boldsymbol{\sigma} = \mathbf{Ba} \qquad (4.35)$$

(4.35) and (4.32):

$$\int_V \mathbf{B}^T\mathbf{DBa}\, dv = \int_S \mathbf{N}^T\mathbf{t}\, ds \qquad (4.36)$$

The boundary condition for a point can be expressed in two ways. Either by the traction vector **t** (known as the natural boundary condition) or a displacement vector **u** (known as the essential boundary condition). The boundary is divided in two parts S_h and S_g. On S_h the traction vector is known and on S_g the displacement vector is known. This gives:

$$\mathbf{t} = \mathbf{h} \quad \text{on } S_h$$

$$\mathbf{u} = \mathbf{g} \quad \text{on } S_g \qquad (4.37)$$

where **h** and **g** are known vectors. Combining (4.36) and (4.37) gives:

$$\left(\int_V \mathbf{B}^T\mathbf{DB}\, dv\right)\mathbf{a} = \int_{S_g} \mathbf{N}^T\mathbf{t}\, ds + \int_{S_h} \mathbf{N}^T\mathbf{h}\, ds \qquad (4.38)$$

The following matrices are defined:

$$\mathbf{K} = \int_V \mathbf{B}^T\mathbf{DB}\, dv$$

$$\mathbf{f}_b = \int_{S_g} \mathbf{N}^T\mathbf{t}\, ds + \int_{S_h} \mathbf{N}^T\mathbf{h}\, ds \qquad (4.39)$$

$$\mathbf{f} = \mathbf{f}_b + \mathbf{f}_1 + \mathbf{f}_0$$

With no internal loads and no initial stresses $\mathbf{f}_1$= 0 and $\mathbf{f}_0$ = 0. This gives the standard FE-formulation:

$$\mathbf{Ka} = \mathbf{f} \qquad (4.40)$$

When the result vector is calculated the following three equations are used to determined the stress, strain and displacement.

$$\boldsymbol{\sigma} = \mathbf{DBa}$$

$$\boldsymbol{\varepsilon} = \mathbf{Ba} \qquad (4.41)$$

$$\mathbf{u} = \mathbf{Na}$$

where $\boldsymbol{\sigma}$ is the stress vector, $\boldsymbol{\varepsilon}$ is the strain vector and $\mathbf{u}$ is the displacement vector.

4.3.2 Four-node Shell Element

This four node element is used to simulate surfaces with a thickness. It is drawn as a surface but handles forces in all three dimensions. It is often used to simulate sheet metal. This means that in addition to placing the nodes of the element and giving it material properties, a thickness also needs to be decided. It is a first order element which means the shape functions for the element is of the first order. The element has four nodes.

4.3.3 Ten-node Tetrahedral Element

This element has the shape of the simplest four node element containing a volume, but is of second order. This means that there are six more nodes on the borders of the element to make it a total of ten nodes. This is the element I-Deas recommend when modeling solid structures since it is accurate and relatively fast. Since it is a second order element the shape functions for the element is of second order.

4.4 Structural Optimization problems

For a more detailed description of optimization problems see [11]. An example of a structural optimization problem can be formulated as follows:

“find the value of x that minimizes the function f(x) while satisfying the function g(x)”

The problem formulation could also have been to maximize or find a target value for f(x). From the statement above, three basic entities are found that are used in optimization problems:

- Objective function –the function that will be minimized, maximized or that will reach a target value. Examples of objective functions can be to maximize the stiffness in a structure or to minimize the stress at a certain point in a structure.

- Design variables –a set of variables or unknown that affects the value of the objective function.

- Constraints –the bounds on the response function that needs to be fulfilled in order for a design to be acceptable.

Mathematically, the optimization problem can be formulated as:

min $f(x)$	-the objective function
while $g(x) < 0$	-the constraint function
if $x_l < x < x_u$	-the design variable (variables if x is a vector)

There are different kinds of structural optimization problems. Shape optimization changes the outer boundary of a structure to solve the optimization problem. The structure is divided in finite elements and the grip point locations are changed. Size optimization modifies the properties of structural elements, for example mass, spring stiffness, shell thickness and beam cross-sectional properties. Topology optimization distributes the material within a given design space by using material properties. Finally, topography optimization is an advanced form of shape optimization. It creates shape variable-based reinforcements within a defined design region. Splitting the design region into several variables allows the user to create any reinforcement pattern in the design space instead of being restricted to a few.

A designer using optimization analyses could start with topology optimization on an allowed design space. This would generate a suggestion for placement of material. After creating a design based on the results from the topology optimization, a shape or size optimization could be used for further fine-tuning.

In this project, only topology optimization has been performed on the available design space with the computer program Optistruct. This generated a concept design, a guideline that suggested where to place material in the design structure. The theory behind this optimization type will be treated in further detail.

4.4.1 Topology optimization

In topology optimization the objective can be to maximize the stiffness or equivalently minimize the compliance under the constraint that the mass is limited and the material will be distributed in the design space. This is exactly what has been done in the optimization analyses of this project. The mathematical theory behind this problem formulation will be presented here.

Minimum Compliance Design Formulation

Consider a mechanical element as a body occupying a domain Ω^m which is part of a larger reference domain Ω in $\mathbf{R}^2$. The reference domain Ω is chosen so as to allow for a definition of the applied loads and boundary conditions. The design problem can be said to be the problem of finding the optimal choice of elasticity tensor $E_{ijkl}(x)$ which is a variable over the domain. The internal virtual work of an elastic body at the equilibrium u and for an arbitrary virtual displacement v can be written in the energy bilinear form as:

$$a(u,v) = \int_{\Omega} E_{ijkl}(x)\varepsilon_{ij}(u)\varepsilon_{kl}(v)d\Omega \qquad \textbf{(4.42)}$$

With linearized strains $\varepsilon_{ij}(u) = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$ and the load linear form

$$l(u) = \int_{\Omega} pu\mathrm{d}\Omega + \int_{\Gamma_T} tu\mathrm{d}s \qquad \textbf{(4.43)}$$

the minimum compliance (maximum global stiffness) problem takes the form:

$$\begin{aligned}
&\underset{u \in U, E}{\text{minimize}}\ l(u) \\
&\text{subject to:} \\
&a_E(u,v) = l(u), \ \text{for all } v \in \mathrm{U} \\
&E \in E_{ad}
\end{aligned} \qquad \textbf{(4.44)}$$

The equilibrium equation is written in its weak form with U denoting the space of kinematically acceptable displacement fields, p are the body forces and T the boundary traction part $\Gamma_T \subset \Gamma \equiv \partial\Omega$ of the boundary. The index E is used to indicate that the bilinear form a_E depends on the design variables. E_{ad} denotes the set of acceptable rigidity tensors for the design problem. In this case of topology design, E_{ad} could consist of all rigidity tensors that attain the material properties of a given isotropic material in the unknown set $\mathbf{\Omega}^{\mathrm{m}}$ and zero properties elsewhere.

Design Parameterization

In the topology design of a structure the decision is to be made which points that should be material points and which should be empty material points, remain void. Looking at the reference domain $\mathbf{\Omega}$, it is the optimal subset $\mathbf{\Omega}^{\mathrm{mat}}$ of material points that needs to be determined. Mathematically written:

$$\begin{aligned}
&\mathbf{E} = \rho\mathbf{E}^0 \\
&\rho = \begin{cases} 1 \text{ if } \mathrm{x} \in \Omega^{mat} \\ 0 \text{ if } \mathrm{x} \in \Omega \setminus \Omega^{mat} \end{cases} \\
&\int_{\Omega} \rho d\Omega = Vol(\Omega^{mat}) \le V
\end{aligned} \qquad \textbf{(4.45)}$$

where $\rho \in \{0,1\}$ can be seen as the density of the problem. $\mathbf{E}^0$ is the stiffness tensor for the given isotropic material. V is the limit of the amount of material to work with for the objective function to be satisfied for a limited fixed volume.

In the ideal case all elements would have ρ with either the value 0 or 1. But in reality it is very difficult to avoid elements with intermediate values. By using a discreet parameter in a so called penalty formulation it is possible to obtain a more 0 and 1 or “black and white” solution. This can be done with a proportional stiffness model:

$$
\begin{aligned}
&\mathbf{E}(\mathrm{x}) = \rho(x)^p \mathbf{E}^0 \quad , \quad p > 1 \\
&\int_\Omega \rho(x) d\Omega \le V \\
&0 \le \rho(x) \le 1 \\
&x \in \Omega
\end{aligned} \tag{4.46}
$$

With this formulation, elements with intermediate values of the density give very little stiffness in comparison to the amount of used material. Mainly the high density elements contribute to the stiffness. So by choosing a higher value than 1 for the *p* parameter it is inefficient for the algorithm to choose intermediate density values.

Solving the optimization problem

One way of solving the optimization problem is with the Method of Moving Asymptotes, MMA. The method is described in further detail in [12].

Looking back at the mathematical formulation of the general problem in the beginning of the chapter, the implicit functions f_i are approximated with the explicit functions $\tilde{f}_i^{(k)}$. It is then assumed that this approximation for the objective function and the constraints can be made:

$$
\begin{array}{lll}
\text{min } \tilde{f}_i^{(k)}(x) & & \text{-the objective function} \\
\text{while } \tilde{f}_i^{(k)}(x) \le \bar{f}_i & i=1,2, ..\mathrm{M} & \text{-the constraint functions} \\
\text{if } 0 < x_{\min} \le x^e \le x_{\max} & e=1,2, ..\mathrm{N} & \text{-the design variables}
\end{array} \tag{4.47}
$$

were k is the number of iterations, M the number of constraints and N the number of elements. In MMA, each approximation function $\tilde{f}_i^{(k)}(x)$ is obtained by a linearization of $f_i(x)$ in variables of the of the type $1/(U_e\text{-}x^e)$ or $1/(x^e\text{-}L_e)$, depending on the signs of the derivatives at $x^{(k)}$, where L_e and U_e are parameters that satisfy $L_e < x^{e(k)} < U_e$:

$$
\begin{aligned}
&\tilde{f}_i^{(k)}(x) = \sum_{e=1}^{N} \left(\frac{p_{ie}}{U_e - x^e} + \frac{q_{ie}}{x^e - L_e} \right) + r_i \\
&\text{if } \frac{\partial f_i}{\partial x^e} > 0 \text{ at } x^{(k)} \text{ then: } p_{ie} = (U_e - x^{e(k)})^2 \frac{\partial f_i}{\partial x^e} \;\wedge\; q_{ie} = 0 \\
&\text{if } \frac{\partial f_i}{\partial x^e} < 0 \text{ at } x^{(k)} \text{ then: } q_{ie} = -(x^{e(k)} - L_e)^2 \frac{\partial f_i}{\partial x^e} \;\wedge\; p_{ie} = 0
\end{aligned} \tag{4.48}
$$

r_i is chosen such that $\tilde{f}_i^{(k)}(x^{(k)}) = f_i^{(k)}(x^{(k)})$.

The values of the asymptotic points L_e and U_e are normally changed between iterations and therefore these points are also a function of k.

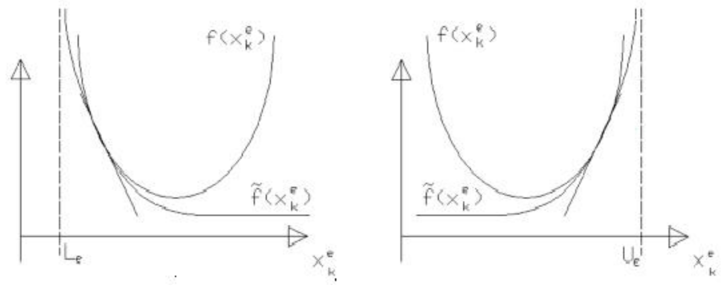


Figure 4.8 Asymptotes L_e and U_e.

Figure 4.8 shows the procedure of using MMA for one design variable x^e. The function $f(x)$ is the implicit function and the $\tilde{f}(x)$ is the approximated function.

The asymptotic points L_e and U_e are always given finite values. A heuristic way can then be used to update the two asymptotic points. The asymptotes move closer to each other when the optimal design is iterated.

In the topology optimization the compliance is equal to $f_0(x)$ and the structure is restricted to a certain amount of the design domain V_0, $f_1(x) = fV_0$. The derivative of the objective function is found by using equilibrium:

$$ku = F \rightarrow \frac{\partial k}{\partial x^e} u + k \frac{\partial u}{\partial x^e} = 0$$
$$\frac{\partial C}{\partial x^e} = \frac{\partial u^T}{\partial x^e} ku + u^T \frac{\partial k}{\partial x^e} u + u^T k \frac{\partial u}{\partial x^e} = -p(x^e)^{p-1} u^e k_0 u^e = -p(x^e)^{p-1} q_c \tag{4.49}$$

In the equation it is assumed that the loads are independent of the design, $\frac{\partial F}{\partial x} = 0$.

The derivative of the constraint function is:

$$\frac{\partial V}{\partial x^e} = v^e \tag{4.50}$$

The big advantage of using MMA is that $\tilde{f}_i(x)$ is convex and then closer to the behavior of the objective- and constraint functions.

4.5 Cast Methods

There are several different cast methods such as lost foam casting or die casting, but for ductile iron in the size and quantities in question sand casting is the natural choice. Lost foam would give more freedom to design without having to consider draft or cores but would be more expensive. Casting has several advantages but also disadvantages over using fabricated parts [13].

Advantages using castings:

- Freedom to design more complex structures.
- Several manufactured parts can be replaced by one more complex casting cutting down on welding and bolted joints.
- Cheep compared to manufactured parts, especially for high volumes.
- Since no mechanical deformations occurs the material properties are uniform in all directions (isotropic).

Disadvantages using casting:

- The casting needs machining afterwards to reach the necessary surface finish and tolerances.
- Cast iron is more brittle and/or has less strength than for example sheet metal.
- The number of parts must be quite high to make up for the high tooling cost.

The cast process can be divided into the following steps [14]:

- **Obtaining the Casting Geometry**.
  To get a design ready for casting draft angels needs to be added. This is so that the sand in the mold will come off the pattern in good condition. The necessary draft angles vary depending on the geometry. 3 degrees is normal but for small areas it can be reduced and for deep pockets there may need to be more. The shrinking also needs to be taken into account. When the metal gets stiff it shrinks. The shrinking is different depending on how thick the structure is, how fast it cools down and where the feeding of metal is located.
- **Making the Pattern**
  The pattern is a physical model of the casting used to make the mold. The high tooling cost for making the pattern is the main reason parts that are manufactured in high volume are more suitable for casting than low volume parts. The pattern is of made metal and can be used for as long as the part is in production.
- **Coremaking**
  When there are cavities in the desired structure cores must be used. Depending on the geometry one or more cores can be needed. Cores, like the mold, are made of sand and the cores are put inside the mold to create cavities in the casting.
- **Molding**
  The mold is made by packing sand and binder around the pattern. When the pattern is withdrawn, its imprint is the mold cavity. It is because of this process there need to be draft angels. In the mould cavity possible cores are placed and another mold seals it tight except for one hole to pour in the liquid metal.
- **Melting and Pouring**
  The metal is melted and possible alloys are added. Ductile iron should be in the temperature range of 1340-1480ºC when poured into the mould. After the metal has got stiff the mold is destroyed to free the casting.
- **Cleaning**
  Cleaning means all operations necessary to remove sand, scale and excess metal from the casting. After the casting is separated from the mould the sand and scale

is removed mechanically, usually by a “shaking” transportation band. Then the excess metal is removed by a hit from either a worker with a hammer or an automated machine. The castings are then inspected for defects before going to painting and machining.

- **Painting and Machining**
  The casting are usually painted and then machined to keep the machined surfaces clean. Examples of machining are drilling or surface machining.

4.6 Bolted Joint

Bolts are divided into different standardized strength classes. If a bolt is marked 8.8, the first digit gives the fracture limit of at least 800 N/mm^2. The second digit indicates the relationship between the yield and the fracture limit is 0.8. This means that the yield limit is 800 x 0.8 = 640 N/mm^2. Other examples of standardized classes are 4.6, 5.6 and 10.9.

A bolt joint is a removable joint connecting two or more details. When a bolt joint is meant to transfer shear forces it is preferred that it be done with friction between the contact surfaces of the connection. It is of importance that vibrations do not cause the joint to loosen up. To prevent this a large prestress is applied to the bolt. The prestress also increases the friction used to handle shear force. The bolt is very sensitive to fatigue since its design creates large stress concentrations in the thread. The prestress makes large variations in the axial force, F_A, and gives small variations in the bolt force, F_S. The decrease of the bolt force is caused by elastic deformations in the bolt and the surface. When a bolt is prestressed, the load creates a prolongation of the bolt and a compression of the surface. If the prestress is not too high the deformations are elastic. With external loading the surface is first relieved while the bolt is slightly stretched. Thus, with a large prestress (larger than the extern load) force variations in the bolts are small, while force variations in the surface are large. This can be shown with an F-δ diagram, Figure 4.9.

When applying a prestress to a bolt it gets the force F_S, and the elongation δ_S. An equally large force acts on the surface, F_U, which leads to a compression δ_U. In the diagram F_U and δ_U are positive. Since both forces are introduced with the prestress they are marked as F_P in the diagram. When a bolt joint is loaded with the axial force F_A the force and deformation relations are changed. Assuming that F_A is acting straight under the bolt head all forces can be put into the diagram and the important relationship can be seen: the bolt force F_S changes a lot less than the applied force F_A if the bolt joint is properly prestressed.

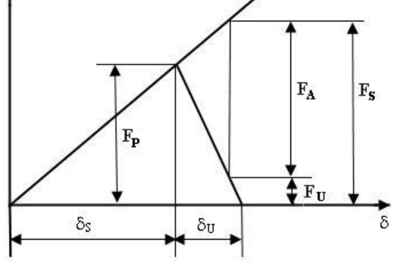


Figure 4.9 F-δ diagram [15].

This relationship exists as long as F_U is greater than zero. If not, the bolt joint no longer has any compressive force on the surface. Adding a shear force, no friction between the connected surfaces is taking up the force. Instead the bolt takes the whole shear force which is, as written before, not to prefer. More theory about bolted joints are found in [15].

5 Computer Analyses with Finite Element Method

The two computer programs ProMechanica and I-Deas were used for stress analyses. The program Optistruct was used to give an indication of how to distribute material. This chapter describes how the designs problems were set up and simulated in the three programs.

5.1 Modeling in ProMechanica

ProMechanica is compatible with ProEngineer and the models are easy to transfer between the two programs. ProMechanica automatically chooses the mesh independent of boundary condition, coarse where possible and smaller only when necessary for the geometry. The accuracy is assured by the edge order of the element. The program runs one pass with the edge order three on all elements. This means all elements have same number of nodes. The first run is followed by a second run were the edge order is different between the element depending on the results from the first run. The program keeps the geometry of the element but changes the number of nodes in each element when changing the edge order. The edge order varies between one and eight. The elements are triangular solid elements.

All analyzes are made without the pin to save computing time. The nodes in the inner surface of all bolt holes attaching to the frame are locked in all degrees of freedom. The force is equally distributed on all nodes on the inner surface of the hole for the towing pin. The bolted joint against the frame rail is simulated in a simple way to save time. The inner surface of all holes included in the joint are locked in all directions.

When a model is imported from ProEngineer it comes as one solid, not different parts. This means the program treats it like one volume and surfaces between parts that are in contact with each other size to exist. Therefore bolts and welding are unnecessary to have in the model since the parts they were meant to hold together merged to become one part.

Only one load case at a time can be applied in ProMechanica which means a number of different analyses for every part has to be done.

When analyzing the results the von Mises stress is studied according to Mack standard procedure. The scale for the cast parts were 0 to 310 MPa since spherodial graphite iron 0722 have a lower yield limit of 310 MPa. When looking at the results for the sheet metal parts the scale ranged up to 355 MPa since that’s the lower yield limit for structural steel 2132 (which is used in the sheet metal parts).

5.2 Modeling in I-Deas

Models from ProEngineer can not be imported directly into I-Deas. The models have to be saved in a file format that is supported by both programs, for example AEGIS which we used. When an AEGIS model is opened in I-Deas it consists of surfaces that is suppose to contain a clearly defined volume to make it a solid model. Since the file format of the model have been changed twice there are often small gaps between the different surfaces that have to be fixed manually to make the model tight. When there are no more free edges left I-Deas can create a solid 3-D model.

The sheet metal parts are simulated by shell elements. To accomplish this a surface is created in the middle of every sheet metal part in I-Deas. On that surface a 2-D mesh of first order, four node elements with an average length of 8mm is laid. The data for the surface elements correspond to the material and thickness of the sheet metal part.

To generate a 3-D solid mesh over cast part, a 2-D mesh of triangular element is projected onto the surfaces of the part. The 2-D mesh is checked to ensure the quality of the element. With the 2-D mesh as a base the 3-D solid mesh is created inside the volume of the part. After that the 2-D mesh is deleted. The 3-D mesh consists of tetra element of the second order with material data corresponding to the material of the part. The average length of the element is 8mm where the geometry allows it, the length is smaller where the geometry so demands.

All parts are meshed separately. They are then brought together in an assembly where the coordinate systems are connected to each other so that the parts come in the right place relative each other. The parts are attached to each others using bolted joints or welding.

To simulate a bolted joint, a node is put in the center of all holes in the joint. A rigid element including the center node and all nodes on the inner surface of that particular hole is created. For the shell elements the center node is in the same plane as the shell elements and for the tetra element the center node is placed in the center of the hole. Beam elements with circular cross sections of the same diameter as the bolts are then created between all the center nodes of that bolted joint. The beam elements have the same diameter and material as the bolt which it represented.

Welding exists only between sheet metal parts at a 90 degree angle against each other. Since the shell element used to simulate sheet metal is located in the center plane of the sheet metal, two shell element mesh representing two steel plates does not touch each other, even if the plates they are simulating are. This is because the shell elements have no thickness but the sheet metal does. To simulate the places where there were welding, element that connected the two meshes are added manually between nodes in the two elements. The added elements are of the same type as the plates they were attached to.

The boundary conditions used in I-Deas are different from those used in ProMechanica. The holes for the bolts that fasten the design to the frame are not locked in all directions. Instead they are implemented in a manner similar to the bolted joints. Nodes are placed in the center of the holes next to the frame and rigid elements fixated them to the holes. Attached to the center nodes are beam elements of the same length and diameter as the hole. The beam elements are fixated in all degrees of freedom in the ends opposite the center nodes. If the structure has a towing pin, the forces are applied at a node in the center of the pin. The pin itself is simulated by beam elements with a circular cross section of diameter and material in question. If there is a ring or a hook, a node is placed outside the structure. A rigid element including all the nodes on the surface to which the force is to be applied and the extra node is created. The forces are applied in that extra node. I-Deas calculates all different load cases for one structure at the same time.

The results are viewed using von Mises stress. In critical areas maximum principal stress is also viewed for comparison. The scale was 0 to 310 MPa for the cast parts and 0 to 355 MPa for the sheet metal part, same as the lower yield limit for each material respectively.

5.3 Modeling in Optistruct

Modeling in Optistruct starts with an imported CAD-module. The same problem as described when importing to I-Deas appears and some gaps and edges on the geometry has to be adjusted.

Creating the mesh in Optistruct starts with a 2D-mesh with triangular elements applied to all surfaces. The elements are of first order with an average length of 10 mm. When the surface mesh passed the element check (element angles, duplicates, etc) a volume mesh of first order tetra elements can be created by the program automatically. Some parts of the volume mesh is defined as ’non-design’, i.e. the program is not allowed to remove this volume in the analysis. The remaining elements are defined to the ‘design’ volume. This volume is free for the program to work with.

The TMC’s five towing loads is applied in different ‘load cases’. This means that the program calculates one solution that works for all of the five loads applied separately. As boundary conditions, the nodes of the elements in the bolt holes thought to connect to the chassis frame rail are locked in all directions.

The bolt joint is simulated by separating the sidepiece from the centerpiece 3 mm. Cylinders are created to fill up the holes and act as bolts in the gap. If the parting had not been made, the Optistruct program would have calculated the whole contact surface as being attached. With this set up it should give some flexibility to the bolts as in reality.

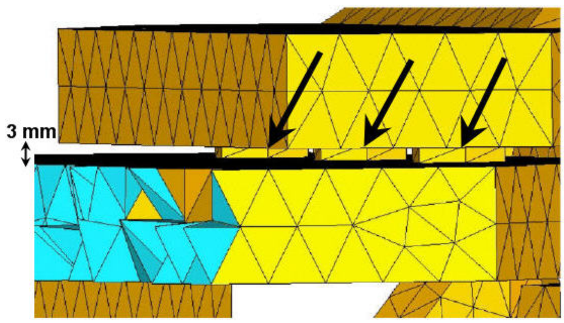


Figure 5.1 Gap of 3 mm between the sidepiece and the center piece.

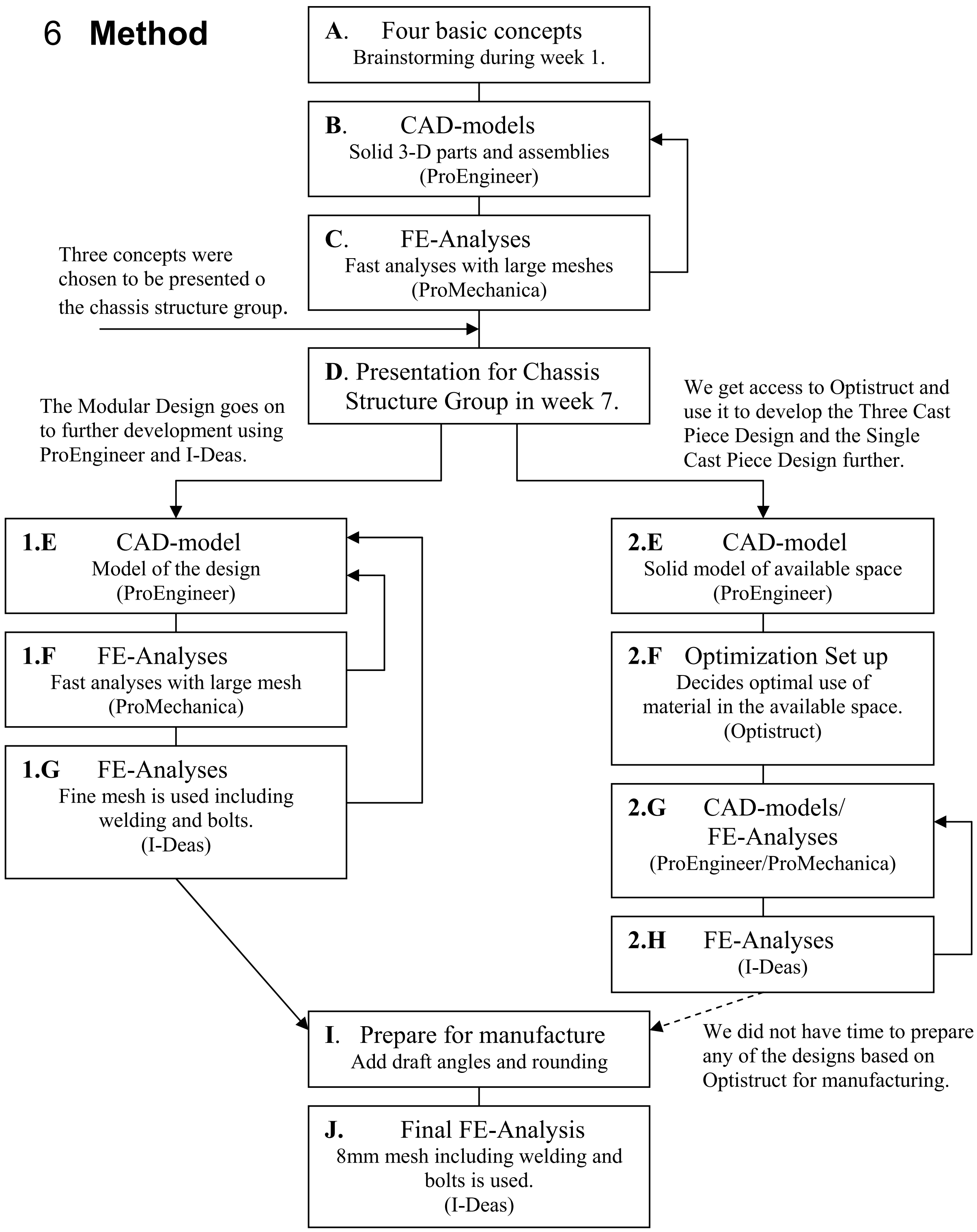
As objective for all Optistruct analyses, the program is set to keep 10-20% of the ‘design’ volume. To get a result with a design as stiff as possible the analysis is set to minimize the compliance, i.e. maximize the stiffness. The compliance is the strain energy of the structure.

There is an application in the Optistruct program to create cast able parts, either with single or split draw direction. The analysis results in a shape that can be created with a one-piece mould or a split mould. Here the analyses are made with split draw direction.

In the program the element density of each element is penalized so that only the high-density elements contribute to the stiffness. In the ideal case the density of all the elements is either 0.0 or 1.0, but it is very difficult to avoid mid-density elements. By using different manufacturing constraints many mid-density elements can be pushed towards 0.0. Getting rid of all of them is very difficult, and even in the converged solution there will be some mid-density elements left. The penalization is made automatically in the program.

The topology optimization is a concept design, meaning that it indicates where the material has to be left and where the material has to be removed. To be able to see a structure from the analyses isosurfases can be viewed by choosing to show all elements with a density above a certain limit. All the elements above the limit will be displayed as if they had the density 1.0 and the elements below displayed as density 0.0. By testing different levels of isosurfaces, it is concluded that 0.3 was a good level to visually see a cast able part of the analyses. Lower value took away too much of the material, higher added too much instead.

6 Method



The schedule is an overview of how the design development work has progressed during this 20 week project. In this chapter the steps of the schedule from the early ideas to the final solution are presented.

Detailed description of the schedule:

A) The first step of the design process was brainstorming where four different basic concepts were decided (see 6.1.2).

B) The basic concepts were modelled in ProEngineer. After making a model the design was checked too fit with the current interface (available space and bolt joint to frame, bumper and hood mount). This is an iterative process where the basic idea is drawn and then adjusted to fit and make best use of the available space. Some of our earlier designs are presented in chapter 6.2.

C) ProMechanica was used to do basic FE-analyses. The program is fast and compatible with ProEngineer and the result is accurate enough for a first analysis. The program is described further in chapter 5.1. After an analysis was done and the results were examined, the model was brought back to ProEngineer and changed to make it lighter and/or stronger depending on the results. During this process two of the basic concepts were combined to one, which left three to go on to the next step.

D) In week 7 three designs were presented to the Chassis Structure group for feedback and advice how to proceed with the design process. It was decided that the Modular Design (chapter 6.3.3) were to be developed further using ProEngineer and I-Deas (chapter 5.2) and that Optistruct (chapter 5.3) were to be used to help design the Three Cast Piece Design (chapter 6.3.1) and Single Cast Piece Design (chapter 6.3.2).

1.E) ProEngineer was used for making the models later used in ProMechanica and I-Deas for FE-analyses.

1.F) The analyses in ProMechanica are faster but less accurate than the analyses in I-Deas. After reaching a design that was optimized and looked like it was going to be strong enough the next step was to make it into an IGES-file and import it to I-deas.

1.G) More accurate FE-analyses were made in I-deas. The results from the I-deas analysis were used as a ground to start over from 1.E. and make changes in the design to minimize weight and concentrations of stresses.

2.E) A model of all available space was set up in ProEngineer. The interface of the frame and bumper was also included in the model.

2.F) The Optistruct program gave suggestions over the best way to distribute material in the available space to make the design as stiff as possible regarding the input load cases.

2.G) A solid model with the result from Optistruct as inspiration was made in ProEngineer and tested in ProMechanica to find the best shape.

2.H) The model was then transferred to I-Deas for a more accurate FE-analysis.

I) The models were prepared for manufacturing by adding draft angles and rounding in ProEngineer. It often meant redrawing the model.

J) The final model was analyzed in I-Deas to insure that it was strong enough for the given loads.

6.1 Design Guidelines

There are a number of guidelines and preferences given from the Chassis Structure Group that was considered while developing new designs. The guidelines lead to the basic ideas. The bolt interface to the chassis frame rail was discussed in more detail with the group and is therefore treated separately here.

6.1.1 Mack Truck Preferences

Casting should be used as much as possible and the number of moulds kept down to insure the cost low. Where sheet metal is used the constructions should be simple and the welding, bends and cuts should be kept to a minimum. When sheet metal is used in conjunction with castings, the sheet metal should yield before the castings to prevent sudden breakage.

There are no specifications for how much deformation that is acceptable in the towing device. A design guideline has been that the design should show visible deformation before fracture, to make it possible for the truck user to notice the problem before it breaks. The demand will be more difficult to meet with a cast solution than sheet metal, since cast material is more brittle and has lower elongation.

The bumper extension should be designed as low as possible to allow air to flow through to the radiator. The outer bumper should mount to a sheet metal plate or plates since the interface with the bumper may change in the future and sheet metal plates are easier to adjust. There are also stylized bumpers with different interface to take into consideration.

There are different ways of connecting the towing chain. Discussions lead to focus on three different options:

- A pin, which can be lifted up to put a chain or a ring around it. This option is available the existing design for the single tow.
- A hook, located behind an opening in the bumper similar to the existing dual tow.
- A ring, coming out of the front of the bumper, which a chain can be put through.

The minimum required space for the towing chain is 50x50 mm, but if possible more space is to prefer.

6.1.2 Basic Design Concepts

Brainstorming for ideas lead to four basic concepts:

- Three Cast Piece Design (single tow)
  Two cast side brackets and one cast middle section. Uses little sheet metal and is easy to assemble to the frame since the side brackets can be mounted first to the frame rails followed by the middle section.

- Two Cast Piece design (dual tow)
  Two cast side brackets to mount the hooks/pins for towing and a simple middle section in sheet metal to mount the bumper.

- Single Cast Piece Design (single tow)
  One big cast piece spanning between the frame rails. Uses little sheet metal (only to the bumper plates) and has no internal bolted joints which makes it easier and quicker to mount.

- Modular Design (single and dual tow)
  Two cast side brackets, could have hooks or pins for dual tow and a middle section for single tow. The side brackets uses the same mould for dual and single tow. The middle section is made of sheet metal, which yields before the castings to prevent sudden breakage.

6.1.3 Bolt joint interface

On the existing designs and on the early design ideas of this project four bolts connects the towing device to the side of the frame rail and one or three bolts connects it to the lower side of the frame rail. After some discussions with the Chassis Structure Group the fifth hole on the frame rail side was also used. Previously, the fifth hole has been used for the towing hook in the Flush Bumper, the front end option without an extension (see Figure 1.2). The thought was that by using this extra hole the forces would be distributed better. To show the force distribution on the bolts an approximate example is made.

Two bolt joints with five M20 bolts connect the towing device to the chassis frame rail, one bolt joint on each side. (In reality there is a smaller sixth bolt that is connecting the towing device and the spring hanger to the lower side of the frame rail. That bolt will be ignored in this example since this is an approximate calculation.) It is a symmetric problem and the right side is shown in Figure 6.1 and Figure 6.2. Calculating on the towing test load of the TMC´s Forward pull with a force of 430 kN, can the bolt joint withstand the force? The clamp load, F_{CL}, for the M20 bolt is 181 kN. The friction coefficient on all surfaces is μ=0.3.

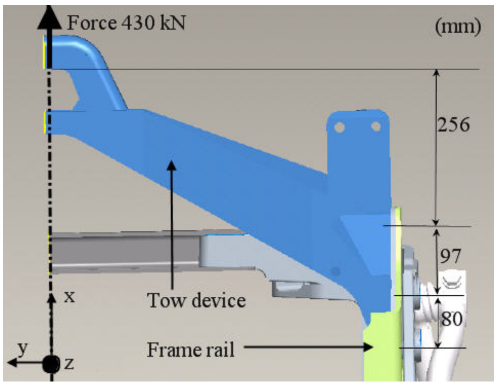


Figure 6.1 Top view.

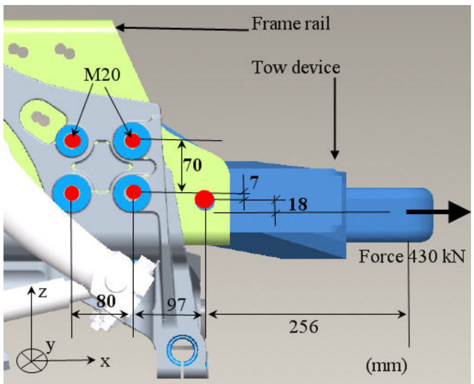
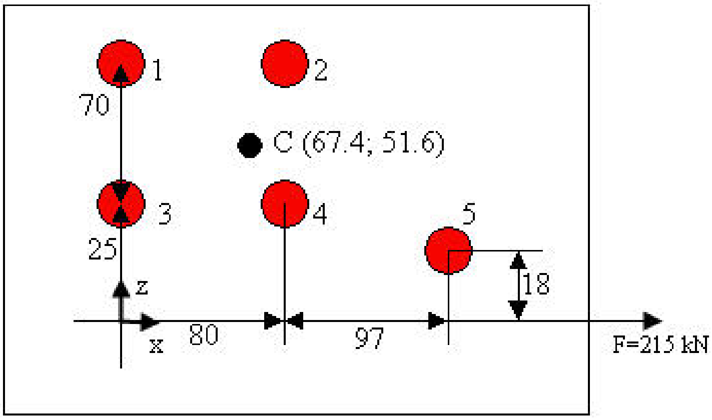


Figure 6.2 Side view.

It is assumed that the bolt joints of the left and the right side take equally much load, i.e. 430/2= 215 kN. The towing device is stiff so that there will only be a moment in the xz-direction and no moment in the xy-direction.

The placement of the centroid (C) was calculated as follows:



$$2 \cdot 0 + 2 \cdot 80 + 1 \cdot 177 = 5x$$
$$\Rightarrow x = 67.4mm$$
$$1 \cdot 18 + 2 \cdot 25 + 2 \cdot 95 = 5z$$
$$\Rightarrow z = 51.6mm$$

The towing force, F, creates a moment acting around C:

$$M_{xz} = z \cdot F = 11kNm$$

When calculating the shear force on each bolt, T_{tot}, the force F is divided on the number of bolts, *n*, and the moment is weighted to each bolt by their distance to the centroid (z or x). The equations are:

$$T_x = \frac{F}{n} + \frac{M_{xz} \cdot z}{\sum_{i=1}^{n} z_i^2}$$
$$T_z = \frac{M_{xz} \cdot x}{\sum_{i=1}^{n} x_i^2}$$
$$T_{tot} = \sqrt{T_x^2 + T_z^2}$$

The total forces on the bolts are:

Bolt 1:	48kN	Bolt 4:	90kN
Bolt 2:	34kN	Bolt 5:	117kN
Bolt 3:	96kN		

The friction force created by the clamp load on the bolt can be calculated as:

$$F_F = \mu F_{CL} = 54.3kN$$

This is what the surface around the bolt is assumes to be able to handle. Comparing with the total forces on the bolts it can be seen that bolt 3-5 has values much higher. This means the friction force is not enough and there will be shear forces in the bolts. The resulting stress in the bolts can be calculated with von Mises stress as:

$$\sigma_e = \sqrt{\left(\frac{F_{CL}}{A}\right)^2 + 3\left(\frac{T_{tot}}{A}\right)^2}$$

A is the bolt cross section area, here for a M20 bolt approximated to $A=\pi r^2$, with r=20mm. The von Mises stress for the bolts are:

Bolt 1:	159MPa	Bolt 4:	190 MPa
Bolt 2:	151 MPa	Bolt 5:	216 MPa
Bolt 3:	196 MPa		

The yield or fracture limit of M20 class 8.8 the values are 640/800MPa and 900/1000MPa. The conclusion is that even though the friction force is not enough and the bolts has to take shear forces there is not a risk of yielding or fracture.

6.2 Early Designs

In this part our early design suggestions are presented together with feedbackfrom the Chassis Structure Group.

6.2.1 Three Cast Piece Design (single tow)

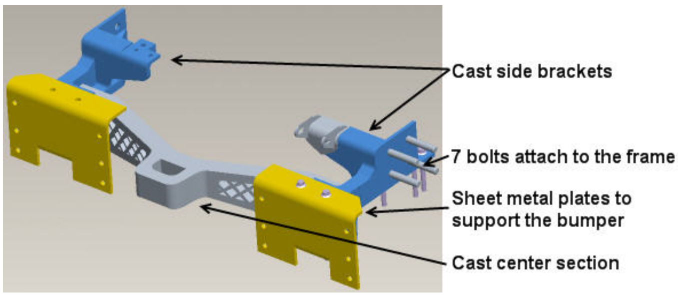


Figure 6.3 Three cast piece design (single tow).

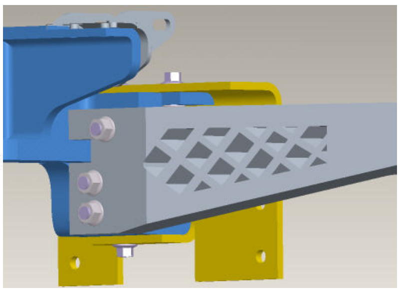


Figure 6.4 Three bolts holds the center section.

Weight: 97 kg

Description: A cast centre section is sticking out in front of the bumper forming a ring, which is used for towing. The centre section is attached to two cast side brackets using three bolts in each side, Figure 6.4. The bumper is mounted on two sheet metal plates, which are supported by the side brackets. The side brackets bolts on the frame rails using seven bolts on each side. The hood mounts rests on the side brackets. The design had stresses that were higher than allowed both in the side brackets and centre section. In the side brackets the highest stresses in the area in front of the seven bolts, where the space is limited by the hood on the top and frame underneath. The lower side in the middle of the centre section also displayed high stresses.

Feedback [16]:It will be hard to cast the pattern in the centre section. It is an interesting idea with a ring coming out of the bumper.

6.2.2 Two Cast Piece design (dual tow)

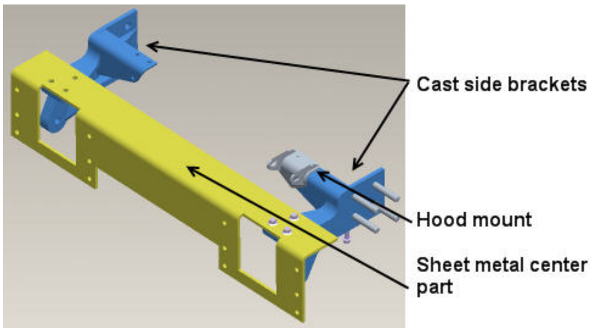


Figure 6.5 Two cast piece design (dual tow).

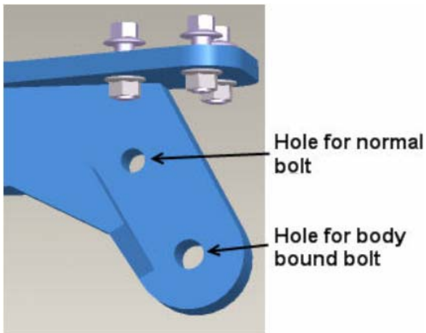


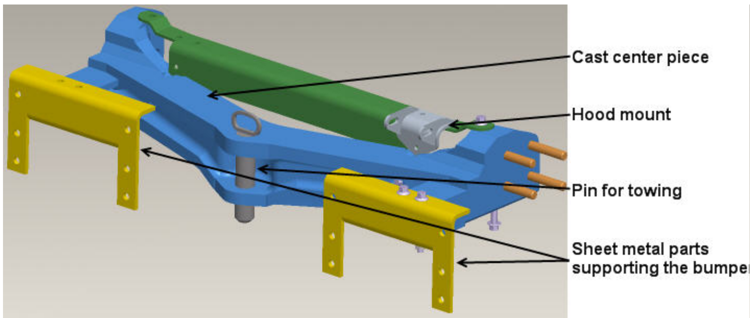
Figure 6.6 Holes to attach the towing hook.

Weight: 50 kg

Description: Two cast side brackets that are connected by three fasteners each to a sheet metal centre part, which also mounts the bumper, Figure 6.5. The brackets are fixed to the frame rails using five fasteners on each side. Hooks from Holland, similar to the ones used today, is to be mounted in the side brackets using one normal and one body bound bolt, Figure 6.6. The side castings support the hood mounts on both sides. The bottom side of the side brackets displayed higher than allowed stresses, especially for the load case where the force is directed up 45 degrees.

Feedback [16]:Make sure there is sufficient space for the bolt heads and tools to assemble them. Try to avoid body bound bolts. Don’t be limited to use hooks just because it’s used today, pins may make it easier when towing. Think about the stability, when taking away the front modules rear cross member it can cause stability problems.

6.2.3 **Single Cast Piece Design (single tow)**



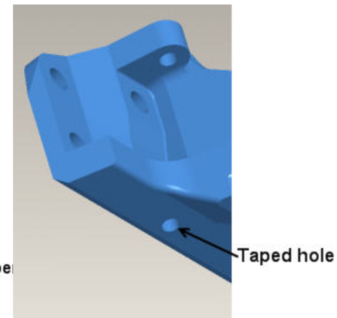


Figure 6.7 Single cast piece design (single tow)

Figure 6.8 Taped hole in the casting for the bottom fastener

Weight: 113 kg

Description: One cast which connect to the frame rails using five of the seven existing fasteners on each side. The bottom hole is not drilled through but taped so no nut is needed, Figure 6.8. The bumper is connected to two sheet metal parts using the existing 12 fasteners and the hood mounts rests on one piece of sheet metal, Figure 6.7. Draw directions are forwards and backwards and no core is needed. A single pin is used for towing. The stresses shown in the analysis in ProMechanica were acceptable in the whole structure.

Feedback [16]:Difficult to assemble with taped holes against the frame rails. Not a lot of space around the fasteners in the side bracket. Good to use a single piece of sheet metal for both hood mounts, keeps it strait and within tolerances.

6.2.4 **Modular Design (dual and single tow)**

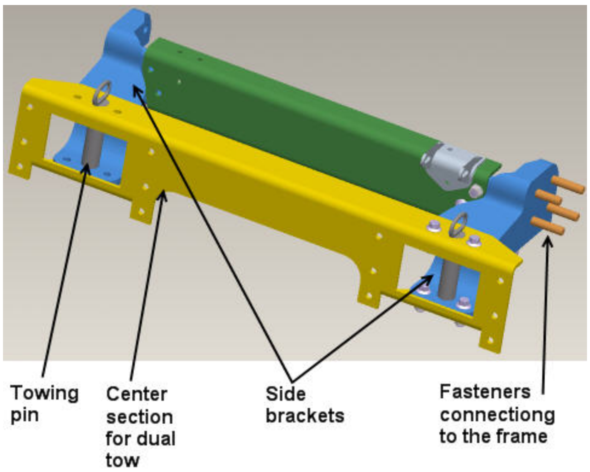


Figure 6.9 Modular design, dual tow.

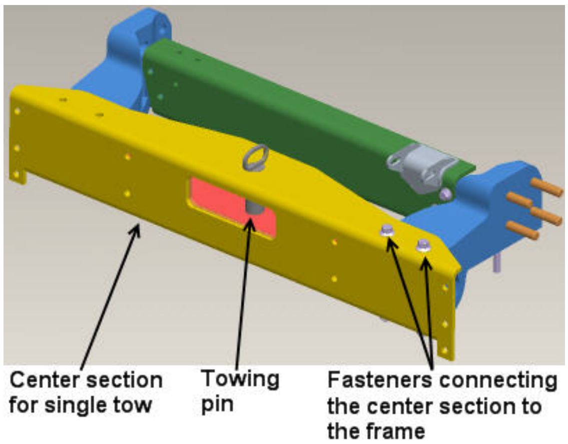


Figure 6.10 Modular design, single tow.

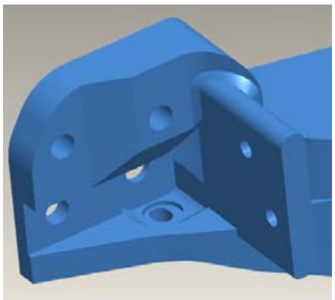


Figure 6.11 Countersink on side bracket.

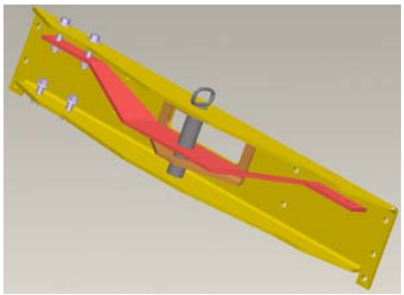


Figure 6.12 Center section, single tow.

Weight: 100 kg (dual tow) 125 kg (single tow)

Description: The two cast side brackets connect to the frame using five of the seven existing fasteners on each side. Between the side brackets there are one piece of sheet metal on which the hood mounts sit. The centre section on which the bumper rests is different between the single and dual tow. On dual tow the middle section is one plate of sheet metal, only used to support the bumper, Figure 6.9. The towing pins are located in the side brackets. On single tow, three pieces of sheet metal makes up the middle section, which also houses the towing pin, Figure 6.10.
There is a countersinking to fit the head of the bottom fastener connecting to the frame and the screw hole is oval to make it possible to insert the bolt. The side brackets displayed acceptable stresses in both single and dual tow. The centre section in single tow displays too high stresses in all sheet metal plates in the area in the middle, both around the pin and in the front.

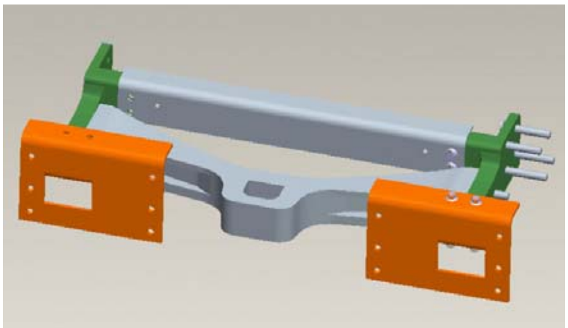
Feedback [16]:The countersink to fit the bolt head (Figure 6.11) is hard to make, either to have in the cast or machine afterwards. The design seems to be over dimensioned when used as dual tow. Good idea to have pins instead of hooks, also for the dual tow.

6.3 Improved designs

Three designs were presented for the chassis structure group to get feedback and suggestions for further improvements.

In the early designs five fasteners on each side is used to attach the bumper extension to the frame rail. By using one more fastener (a total of six on each side) near the front end of the frame rail, weight can be saved and the force is distributed away from the bottom fastener, which in the early designs takes most of the load. All designs using a pin is using the same pin that is used today since it turned out to be reliable and cheap. All designs presented to the chassis structure group on March 16th use six fasteners on each side to bolt on to the frame rail. The three most promising designs, which also reflected the different ideas we had, were chosen for the presentation.

6.3.1 Suggestion A: Three Cast Piece Design



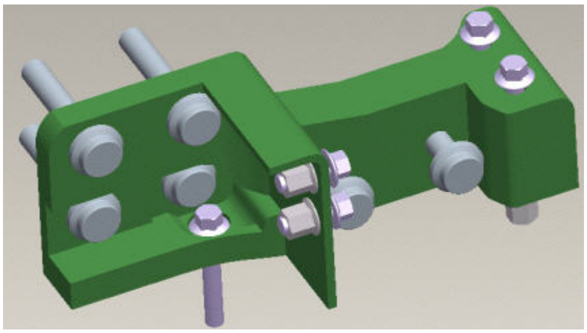


Figure 6.13 Three cast piece design with single cast ring. **Figure 6.14 Alternative A, cast side bracket.**

Weight: 123 kg

Description: Two side brackets and a centrepiece are cast. The centrepiece sticks out through the bumper and forms a towing ring. He side brackets can be mounted to the frame rails before the centrepiece, which makes assembly easier. The three sheet metal plates support the bumper and hood mounts.

Feedback [17]:The chassis structure group thought that the design with a ring coming out of the front of the bumper would fit the Mack Trucks look. Possible changes could be the sheet metal plates holding the bumper; it is hard to manufacture the way it looks now. The bottom bolt connecting to the frame must be slid in sideways which will make it hard to mount to the frame, also the countersink around the bolt head will be expensive to manufacture. Try to simplify the design by making the sheet metal holding the bumper larger and attach it further back.

6.3.2 Suggestion B: Single Cast Piece Design

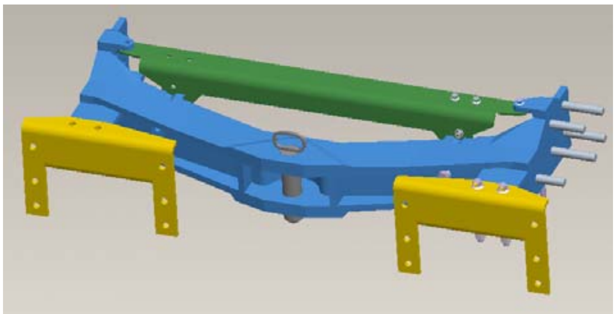


Figure 6.15 Single cast piece design with single pin.

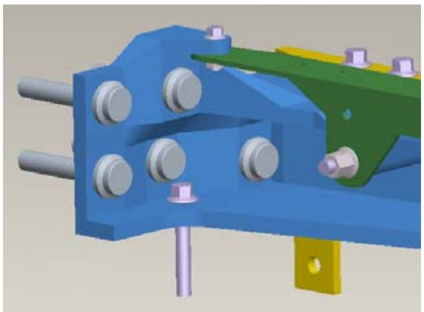


Figure 6.16 Six fasteners attach to the frame rail.

Weight: 82 kg

Description: A single casting is bolted to both frame rails. An I-beam span from the frame to the towing pin placed in the middle. The interface towards the user is similar to current designs. Go-through bolts attach the plates supporting the bumper to the casting. A bent piece of sheet metal bolted on the casting with four bolts supports the hood mounts.

Feedback [17]: The chassis structure group thought the design would be difficult to assemble; the whole casting must be mounted at the same time as the spring brackets, which also use the bolts through the frame rails. We should make it easier to cast, redesign the support for the sheet metal holding the hood mounts. Consider having more sheet metal that attach further back to support the bumper, and don't let the casting go as far forward. Investigate the possibility to bolt the hood mounts direct on the casting.

6.3.3 Suggestion C: Modular Design

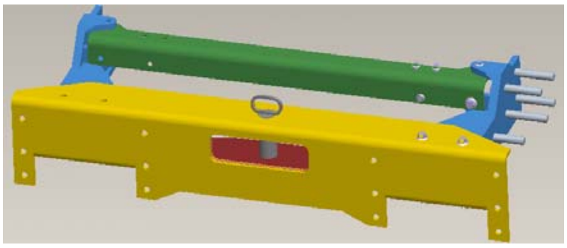


Figure 6.17 Modular design, single tow.

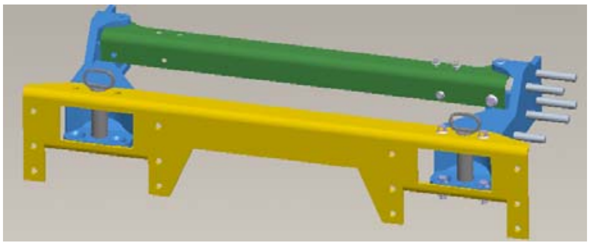


Figure 6.18 Modular design, dual tow.

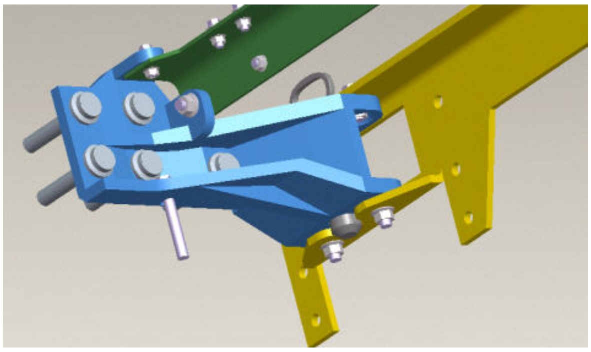


Figure 6.19 Side bracket with bolts.

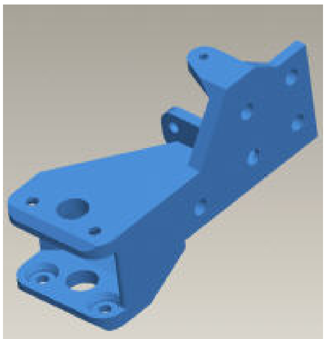


Figure 6.20 Side bracket.

Weight: 98 kg (single tow) 53 kg (dual tow)

Description: Two cast side brackets use the same mould but are machined differently. The sheet metal plate that support the hood mounts are the same for both single and dual tow but the centre section that supports the bumper is different. Two pins located in the side brackets have replaced the hooks for dual tow. One pin is located in the middle of the centre for single tow.

The design consists of two cast pieces and four respectively two pieces of sheet metal for single and dual tow.

Feedback [17]: The chassis structure group thought it was the best suggestion, deals with both assembly and cost issues. A few tips was to make the machining of the holes more different between the dual and single tow. Change the way the countersinking for the bolt heads in the front pocket is shaped, it is hard to machine the way it is now. Pull back sidewall of front pocket to make the pocket deeper and leave more room for the chain. Consider keeping the side brackets but changing the sheet metal for a cast centrepiece similar to the one in Suggestion A, chapter 6.3.1, for single tow. Investigate possibility to use the same plate to support both the bumper and hood mounts.

6.3.4 Data for the improved designs

Table 6.1

Single tow	**Weight**	**Acceptable stresses**	**Cast parts**	**Sheet Metal Parts**	**Pins/ Hooks**	**Parts**
A. Three piece cast	123 kg	No	3	3	0	6
B. Single cast	82 kg	Yes	1	3	1	5
C. Modular (single tow)	98 kg	No	2	4	1	7
Existing design (single tow)	95 kg	No	0	12	1	13
Dual tow						
C. Modular (dual tow)	53 kg	Yes	2	2	2	6
Existing design (dual tow)	64 kg	Yes	0	13	2	15

6.3.5 Conclusion

The comments from the Chassis Structure Group were that suggestion C is the most promising and focus should be to further development this alternative. The sheet metal centre section needs to be reinforced for single tow with a pin. The cast side brackets should be redesigned to make more room for the towing chain and the machining can be more different between single and dual tow. Further FE analyses needs to be made in I-Deas for the stress results to be more reliable.

Work should still continue with suggestion A and B since there are new and different ideas to what the group had seen before. Since the computer licenses to the optimization tool Optistruct from Altair arrived around the time of the presentation, decision was made that suggestion A and B will be further developed with the help of this software since it is well suited for the large cast designs. The Chassis Structure Group was interested to see what the program could do. The towing ring in suggestion A was an idea that the group thought should be kept in addition to the pin for the large cast designs.

6.4 Further Development of Single and Three Piece Design

To distribute the forces in the two large cast solutions, the single and three piece cast designs for single tow, the optimization program Optistruct was used in addition to the CAD-program ProEngineer and the FE-analysis program I-Deas. The problems were set up in the following order:

1. A cad module of all available space was imported from ProEngineer.
2. Generate FE-mesh from CAD-model. Decide design/non-design volume; apply loads, BC's and objectives.
3. Solve the optimization problem.
4. Develop new CAD-model, based on Optistruct results.
5. Analyze CAD-model in I-Deas.

In chapter, 6.4.1, the single cast design solution is treated. Chapter 6.4.2 treats the three pieces cast design. For both the single and the three pieces design, there are two options for the tow application, a ring or a pin. The different options will be referred to as Single-Ring, Single-Pin, 3-Ring or 3-Pin.

6.4.1 Topology optimization single piece cast design

To optimize the single cast piece design two CAD-module of the available space in the front end were set up. The hood mount was included in the space to see if the program would find a good solution to include them in the casting. The available space for the ring and the pin, option Single-Ring and Single-Pin, are shown in Figure 6.21.

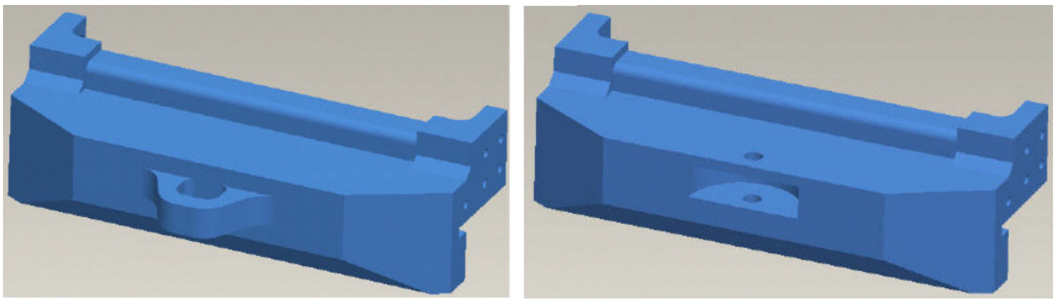


Figure 6.21 The available design space for Single-Ring and Single-Pin.

A finite element mesh was applied to the available space. Option Single-Ring had 142 661 volume elements and options Single-Pin had 181 479. The elements around the bolt holes and hood mount were defined as 'non-design'. Part of the ring and the volume around the hole for the tow pin was defined 'non-design', i.e. not allowed to use in the optimization run. All other elements were defined as 'design', meaning the program could use or remove without restraint.

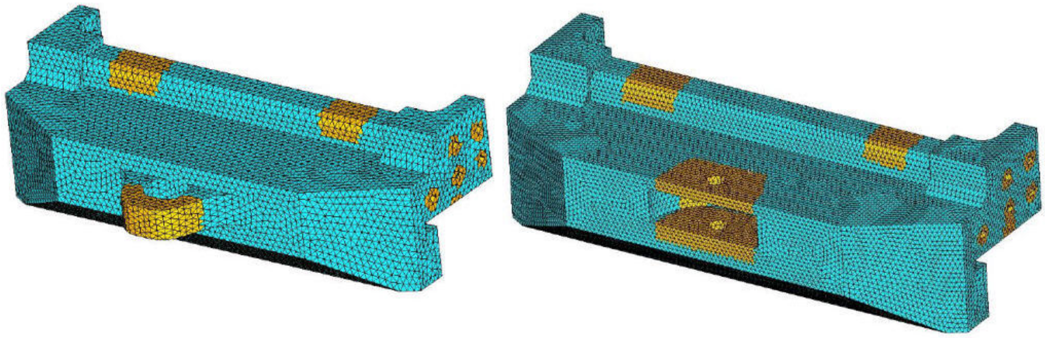


Figure 6.22 Volume mesh, Single-Ring and Single-Pin. ‘Design’ volume is blue and ‘non-design’ volume is yellow.

The nodes on the surfaces of the bolt holes were constrained in all directions to simulate the bolt joint to the frame rail. The five towing forces from TMC (Table 3-1) were applied to nodes on the ring or nodes on the surfaces in the pin hole. The program was set to keep 10-20% of the ‘design’ volume in the optimization analysis and to minimize the compliance. A with split draw direction was defined in the straight forward (and straight backward) direction.

In the results shown in Figure 6.23 and Figure 6.24, the density limit is set to show elements with higher value than 0.3.

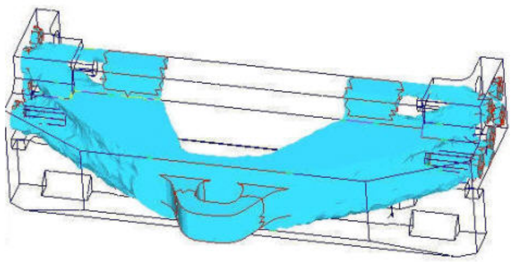


Figure 6.23 Optimization result for Single-Ring.

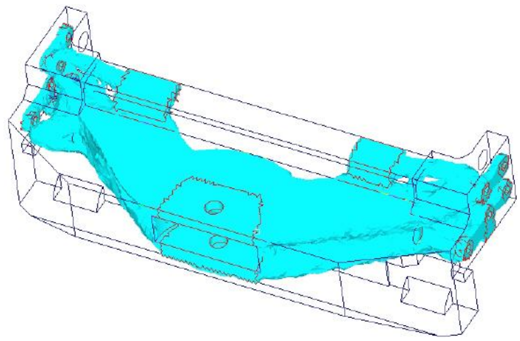


Figure 6.24 Optimization result for Single-Pin.

Both results show I-beam configurations in the arms that lead on each side from the tow center to the center of the bolt joint in the chassis frame rail. For Single-Pin, the I-beams bottom and top sections are connected behind the tow hole.

CAD-modules were created from the results of the optimization analyses.

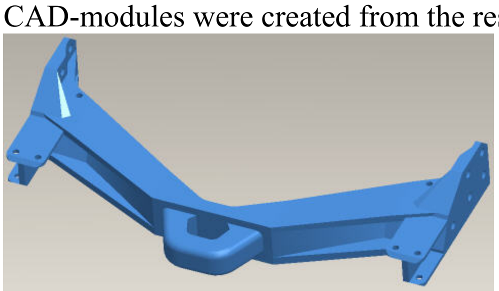


Figure 6.25 CAD-model for Single-Ring.

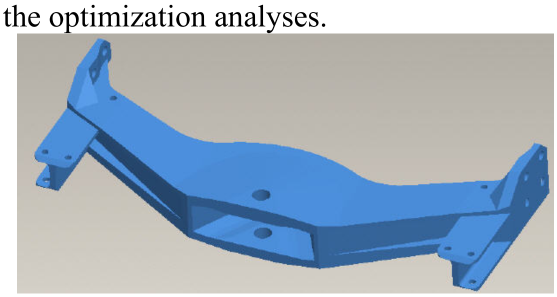


Figure 6.26 CAD-model for Single-Pin.

The hood mount and the bumper plates are made to fit both the options, Single-Ring and Single-Pin. The hood mount plate is of thickness 5 mm and the bumper plates 7.9 mm.

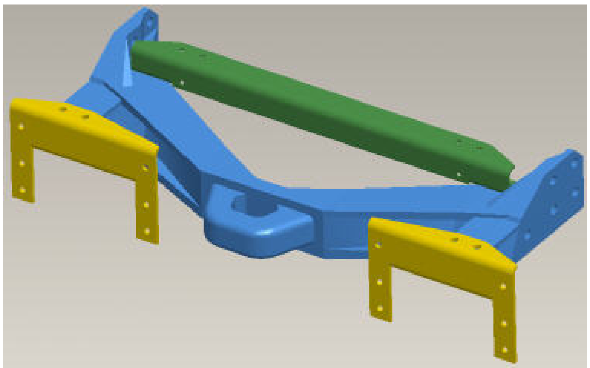


Figure 6.27 Bumper plates and hood mount for the Single-Ring and Single-Pin.

Table 6-2 Single cast design parts.

Parts:	**Parts/design**	**Weight**
Single-Ring	1	63.9 kg
Single-Pin	1	64.3 kg
Hood mount	1	4.4 kg
Bumper plate Standard	2	3.9 kg
Design:	**Parts**	**Weight**
Single-Ring Standard bumper	4	76.1 kg
Single-Pin Standard bumper	4	76.5 kg

6.4.2 Topology Optimization Three Pieces Cast Design

The three pieces cast design concept consists of two side pieces and one cast center piece. The center piece had the two towing options, the one with the ring (here referred to as option 3-Ring) and the one with a pin (option 3-Pin). Figure 6.28 shows the available space that was used in the analysis for the two center section options.

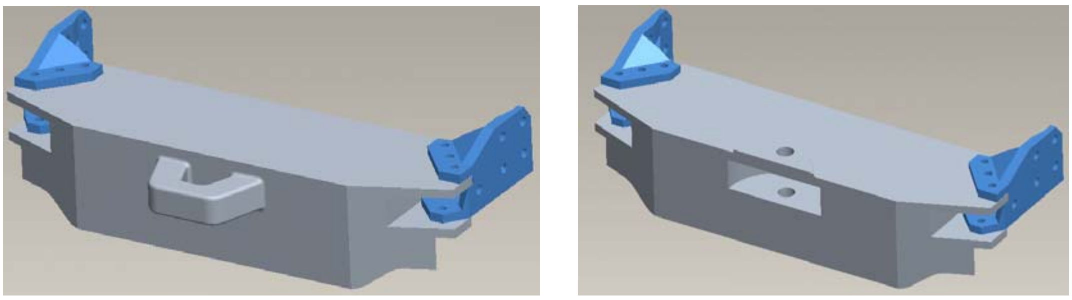


Figure 6.28 CAD-models of side casts and center section for option 3-Ring and 3-Pin.

The side casts were the same in both options, Figure 6.29. Since all forces from TMC's requirements are located in the middle in the front of the center piece, effort was made to try to align the bolted joint connecting the sidepieces to the center piece normally to the direction of the tow point. This would help distribute the force equally on all bolts. The tow point is differently aligned in the ring and pin option. Since the side piece was the same in both options, the bolts could not be exactly normal to either options but lays in between, Figure 6.30. The bolt connection was made with the surfaces parted 3 mm and filled with cylinders, as described in chapter 5.3.

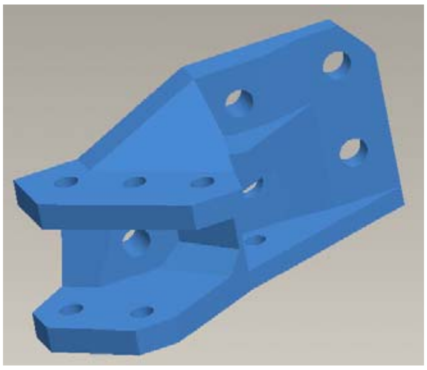


Figure 6.29 Cast sidepiece, same for both options.

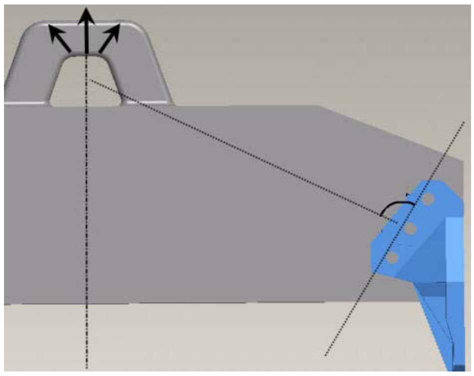


Figure 6.30 Bolt joint normal.

The mesh was created. The volumes consisted of 127 640 elements for the 3-Ring resp. 141 930 for the 3-Pin. The whole volume of the sidepiece and the connection between the side and the centre piece was defined as 'non-design' to keep the bolt interface in the analyses. Part of the tow ring and the volume around the tow pin was also defined as 'non-design' to keep the two tow designs,
Figure 6.31.

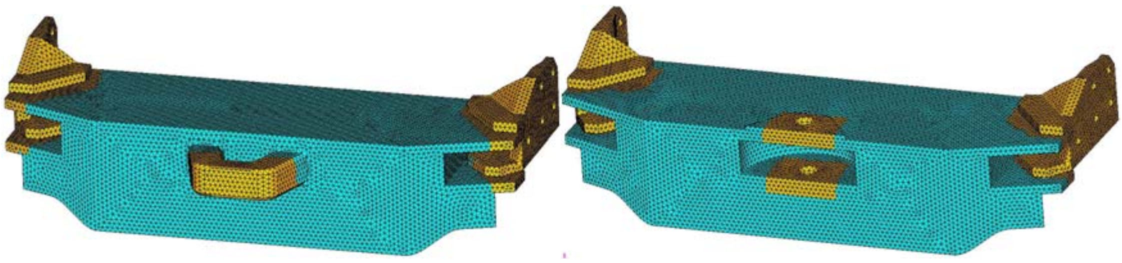


Figure 6.31 Option 3-Ring and 3-Pin. Mesh with yellow 'non-design' and blue 'design' volumes.

As boundary conditions, the nodes in the six holes connecting to the frame rail were locked in the x, y, and z-direction. The five forces defined from TMC were distributed on nodes on the inner surface of the tow ring and on the nodes on the surfaces of the tow hole. The program was set to keep 10-20% of the 'design' volume in the topology optimization and to minimize compliance. A split draw direction was defined.

The results of the analyses in Optistruct are shown in Figure 6.32 and Figure 6.33. For the isosufaces, the density limit chosen is 0.3.

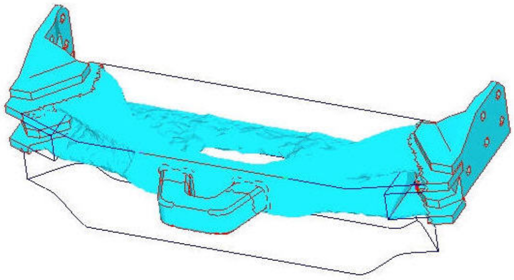


Figure 6.32 Optimization results for 3-Ring.

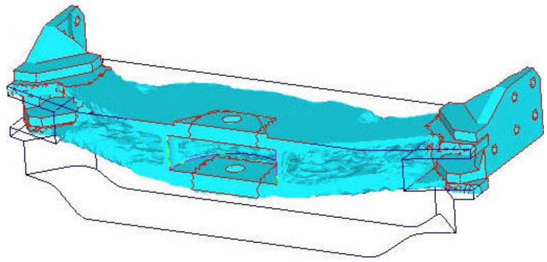


Figure 6.33 Optimization results for 3-Pin.

The CAD-models based on the results are shown in Figure 6.34 and Figure 6.35.

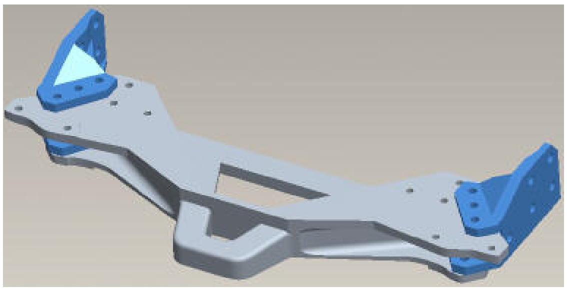


Figure 6.34 CAD-model for 3-Ring.

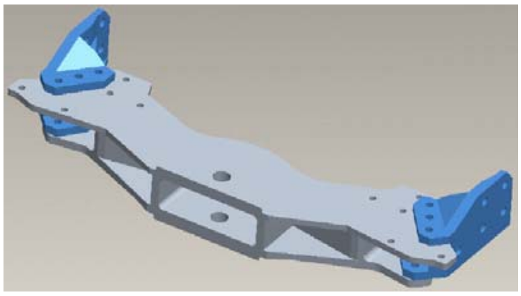


Figure 6.35 CAD-model for 3-Pin.

The hood mount and the bumper plates need to attach to the design. The hood mount is symmetric so that it can be used both on the left and the right side. This will save manufacturing cost since only one mould is needed. Three of the four bolt holes are used when attaching to the large centre cast piece.

There are two bumper plate options to show alternatives for the both bumper options, the standard and the stylized. Support gussets are placed in the bend of the bumper plate to assure the 90º angle. This is of importance to achieve a straight bumper mounting. The gussets are welded to the bumper plates. Both bumper plates are of 7.9 mm sheet metal plates. The placement and the design of the hood mount and the bumper plates are shown in Figure 6.36.

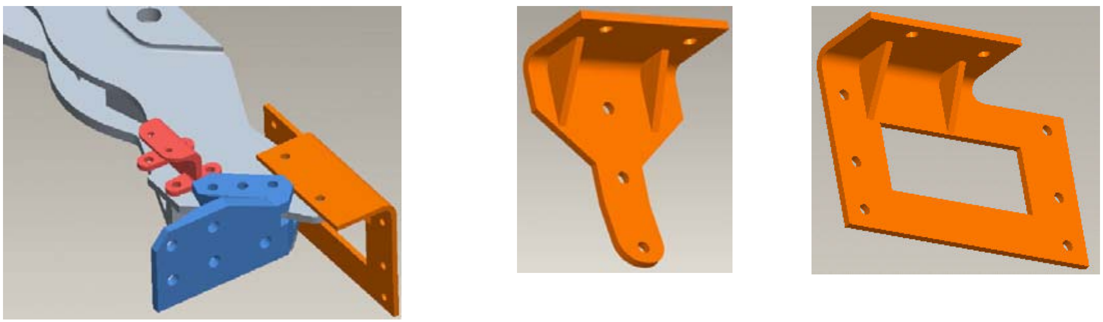


Figure 6.36 Hood mount cast piece and bumper plates for standard and stylized bumper. Can be used for 3-Pin and 3-Ring.

Table 6-3 Three piece cast design parts.

Parts:	**Parts/design**	**Weight**
Side cast	2	10.1 kg
Center section 3-Ring	1	56.8 kg
Center section 3-Pin	1	57.1 kg
Hood mount	2	1.2 kg
Bumper plate Standard	2	6.2 kg
Bumper plate Stylized	2	3.6 kg
Design:	**Parts**	**Weight**
3-Ring Standard bumper	7	93.0 kg
3-Pin Standard bumper	7	93.3 kg
3-Ring Stylized bumper	7	87.8 kg
3-Pin Stylized bumper	7	88.1 kg

6.4.3 Result of Stress Analyses

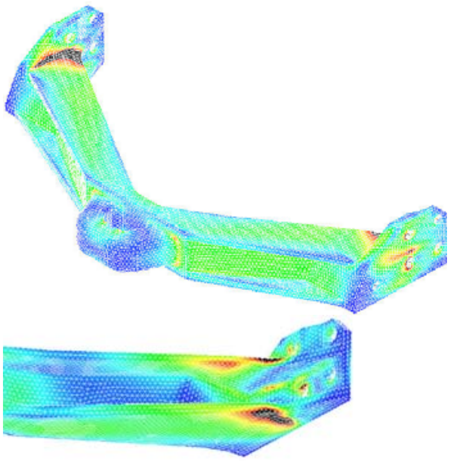
FE-analyses in I-Deas with the five TMC load cases was made on the four CAD-models presented in the chapter above; Single-Ring, Single-Pin, 3-Ring and 3-Pin. For each option the worst case result is shown in Figure 6.37 to Figure 6.40. The remaining analyses can be found in Appendix 1.

The analyses show black areas above the yield limit that are unacceptable. In the Single-Ring and the Single-Pin design the stress concentration appears around the bolt interface. For the Single-Ring it is the Vertical 90° that is critical, because the moment it creates around the bolt interface. For the Single-Pin it is the Forward Pull instead that is the worst load case.

Looking at the 3-Ring the same problem with the large moment can been seen in Figure 6.39. Here, Cone Up 45° is the worst load case. The rib that spans in the back of the middle section seems to works very well. It takes up much of the load as can be seen on its yellow and orange color. For the 3-Pin the Forward Pull is the worst. The bolt connection between the centre section and the side pieces for the both 3 piece solutions shows high stresses.

Even though there are areas with unacceptable stresses, forces are well distributed. On all worst case analyses there are only small dark blue areas that does not contribute to handle the stress. Large areas with light blue, green and yellow are leading forces in a very efficient way.

Because of limited time, the designs are not improved after studying the results from the analyses. Neither is draft angles added to the surfaces.



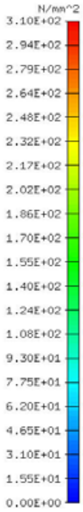


Figure 6.37 Single-Ring, TMC's Vertical 90º (112 580 N). Top and back bottom view.

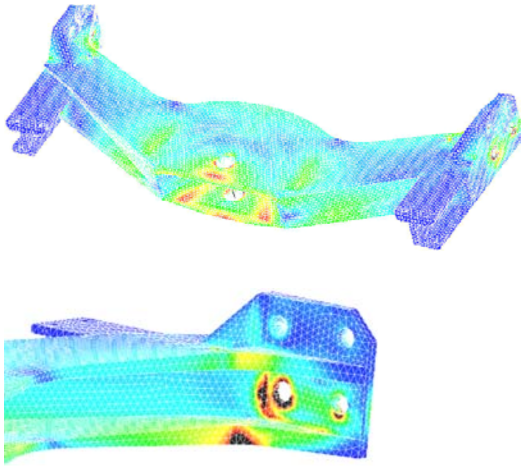


Figure 6.38 Single-Pin, TMC's Forward pull (430 742 N). Top and back/top view.

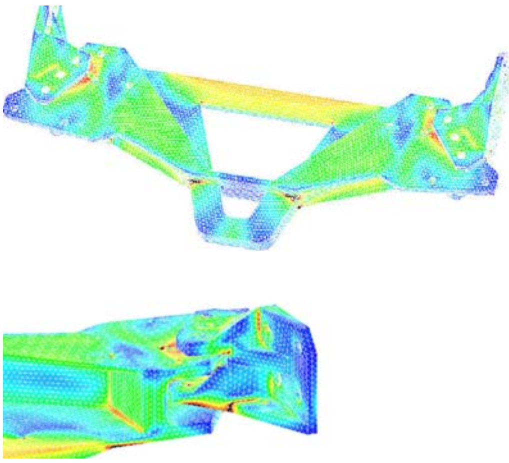


Figure 6.39 3-Ring, TMC's Cone Up 45º (157 612 N). Top and back/top view.

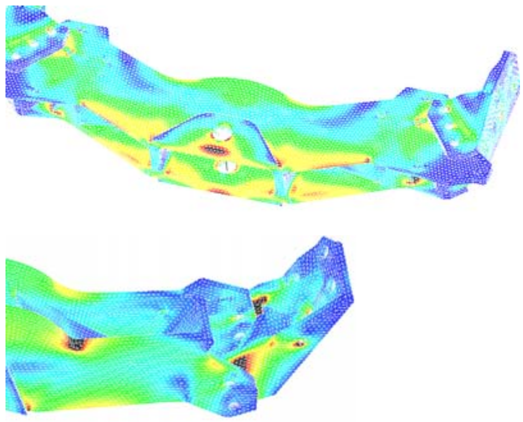


Figure 6.40 3-Pin, TMC's Forward pull (430 742 N). Top and back/bottom view.

6.5 Further Development of Modular Design

The presented in the mentioned in chapter 6 is used to further develop the modular design. Many different ideas are tried, among them the idea of having the same sheet metal plate to support both the bumper and hood mount. The way to bolt the plate that support the hood mount to the side bracket were changed many times before settling for what came to be the final design. With this solution all parts can be assembled in any order preferred by the assembly plant.

The stiffening ribs shown in Figure 6.44 are brought back all the way to the back of the side brackets to distribute the forces as evenly as possible over all six bolts. The front pocket made by core is made deeper to save material and the countersinks for the front four bolts are included in the core to save machining time.

6.5.1 Prepare for manufacturing

The modular design is the only one that is prepared for manufacturing. The design has been analyzed with good results in both I-Deas and ProMechanica. After that draft angles and rounding were added. Since the space is very limited, both the outer dimensions and around the bolt heads, adding draft angles means redrawing the models from the beginning. The draft angles were included in the new models and the adding of machined surfaces and rounding were considered from the start. The draft angle vary from 3 degrees up to ten except for one place where it goes down to one degree to save machining cost in a small area. There are also raised bosses for the heads of the Huck bolts.

6.6 Final Design

The final modular design consists of cast side brackets that attach to the frame and parts made of sheet metal in between that supports the bumper and the hood. There are two versions, single and dual tow. In the single tow version the sheet metal center section also support the tow pin placed in the middle. In the dual tow the pins are located in the side brackets. Both versions share the same castings but they are machined differently.

The design can be assembled in any order. It can be mounted on the frame as one piece or the side brackets can be mounted first and the sheet metal parts afterwards. The part that supports the hood is one piece to ensure proper alignment of the hood mounts.

Table 6.4 Data for the Final Design

Name	Weight	Cast	Sheet Metal	Thickness	Parts
Cast Side Bracket	17 kg	X		-	1
Hood Mount Plate	4 kg		X	5 mm	1
Centerpiece (single tow)	57 kg		X	7.9-12.7 mm	4
Centerpiece (dual tow)	8 kg		X	5 mm	1
Design	**Weight**	**Cast parts**	**Sheet Metal Parts**	**Pins**	**Parts**
Single Tow	95 kg	2	5	1	8
Dual Tow	46 kg	2	2	2	6

6.6.1 Shared Parts

The side brackets are mirror images of each other and are made of cast iron and bolted to the frame rail by five 20 mm Huck bolts and a 14 mm flange bolt. They use the existing interface of bolts also used by the spring bracket on the outside of the frame in axel forward configuration. The draft direction is chosen so that the surface against the side of the frame rail doesn't need machining, Figure 6.41. A core is needed to create the front pocket.

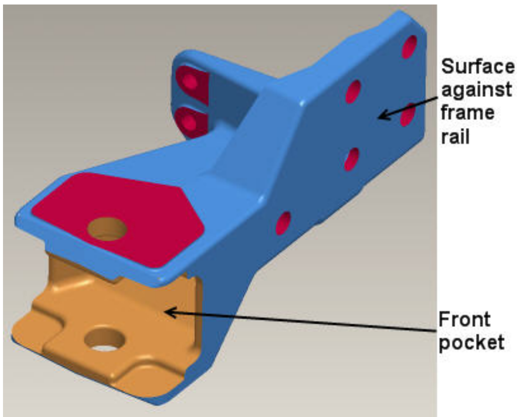


Figure 6.41 Left side bracket.

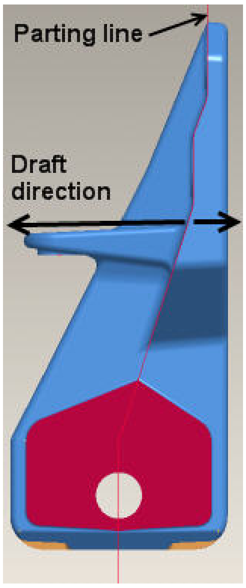


Figure 6.42 Top view.

The parting line and draw direction is as shown in Figure 6.42 and Figure 6.43. The draft angels on the side brackets vary between three and five degrees with one exception, the pocket for the 14 mm nut shown in Figure 6.44. To save machining in a very difficult spot the draft angle is two degrees going down to one degree where the nut sits. That takes away the need for machining since a one-degree draft angle is flat enough for the nut.

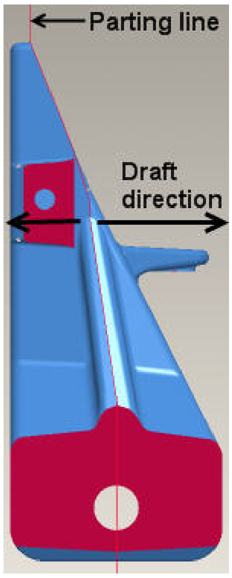


Figure 6.43 Bottom view.

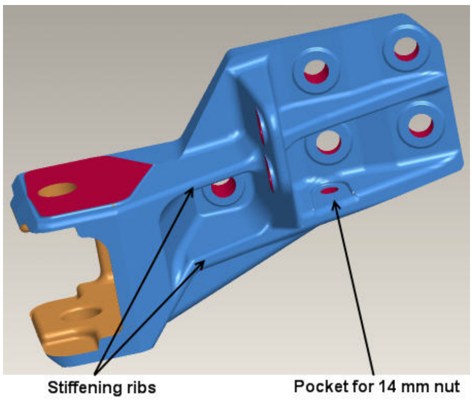


Figure 6.44 Right side bracket.

Thickness on the wall against the frame rail differs so the Huck bolts have to be of different length. This is to fit the bolt heads and transmit force to the rear fasteners. The raised buss for the Huck bolt shown in Figure 6.45 is smaller than the others due to the lack of space in the side pocket. The stiffening ribs shown in Figure 6.44 and Figure 6.46 distribute force to the centre of the bolted joint with the frame rail. Technical data for the cast side brackets are presented in Table 6.4.

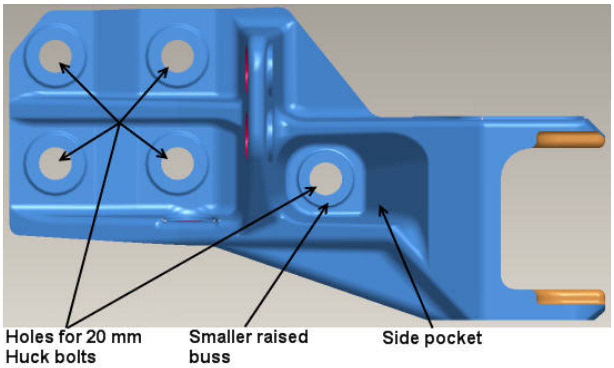


Figure 6.45 Left side bracket, side view.

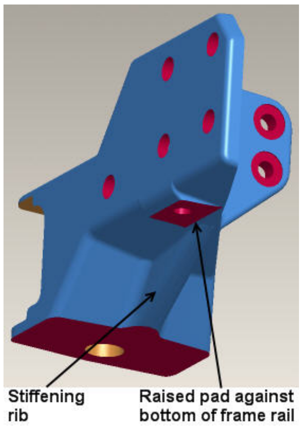


Figure 6.46 Bottom left view.

The plate that support the hood mounts is also same for both single and dual tow. It is made of 5 mm sheet metal and attaches to the side brackets with two 14 mm bolts, Figure 6.47. Technical data for the plate is presented in Table 6.4 .

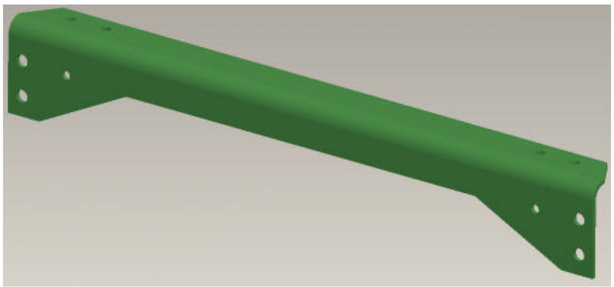


Figure 6.47 Plate to support hood mount.

6.6.2 Single Tow

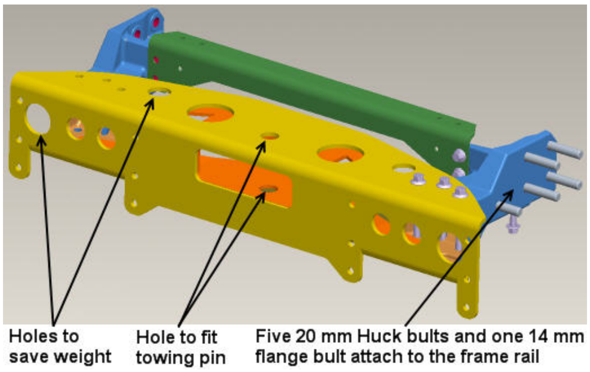


Figure 6.48 Single tow, top front view.

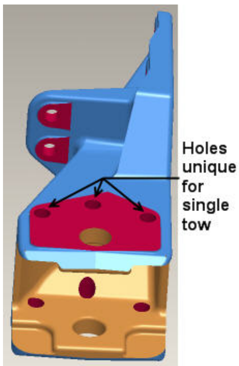


Figure 6.49 Single tow, side bracket.

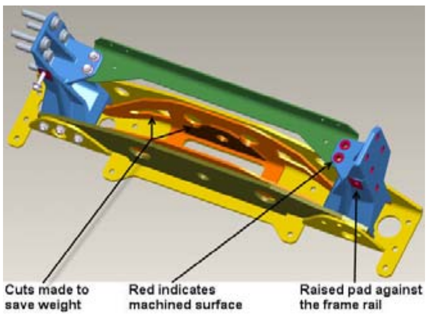
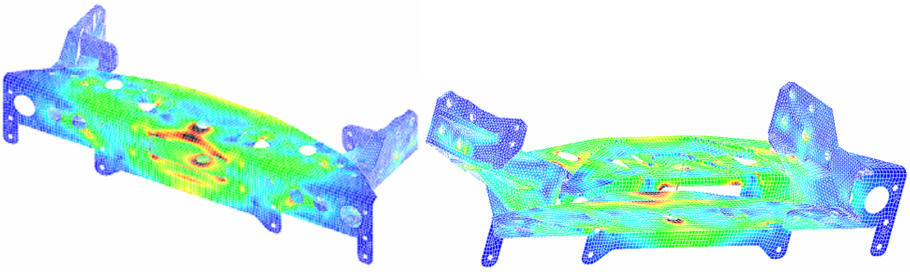


Figure 6.50 Single tow, bottom back view.

In the single tow version the towing pin is located in the middle of the centre section. That means the sheet metal in the centre must be strong enough to lead the forces to the cast side brackets. To achieve this four sheet metal parts are welder together to form the centre section. The thickness of the sheet metal varies from 7.9 mm to 12.7 mm. The holes in the metal visible in Figure 6.48

are made to save weight. The centre section is attached to the side castings with three bolts on each side. The difference in machining of the side brackets is limited to three holes drilled through the front pocket to fit the bolts holding the centre section. Pictures of all the sheet metal parts are in Appendix 3

6.6.3 **Result of Stress Analysis, Single Tow**



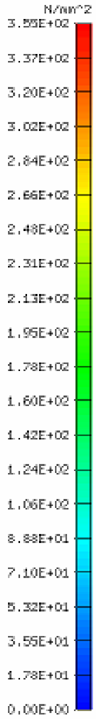


Figure 6.51 Single tow, Forward pull 430 kN.

With exception for the plate that support the hood mounts the whole single tow structure were analyzed. The most demanding load conditions were the Forward pull and the pull Up 45º. As seen in Figure 6.51 the critical areas for Forward pull is around the pin and in the back of the lower plates. No yielding in the cast parts (even though they have lower yield strength). In the load case Up 45º the most critical part is in front of the towing pin and around the bolts, Figure 6.52. The pin goes through five layers of sheet metal, three of them over the pocket for the towing chain. For both load cases the second and third plate from the top is not yielding and can take up more load if necessary. Appendix 3 includes more analyses.

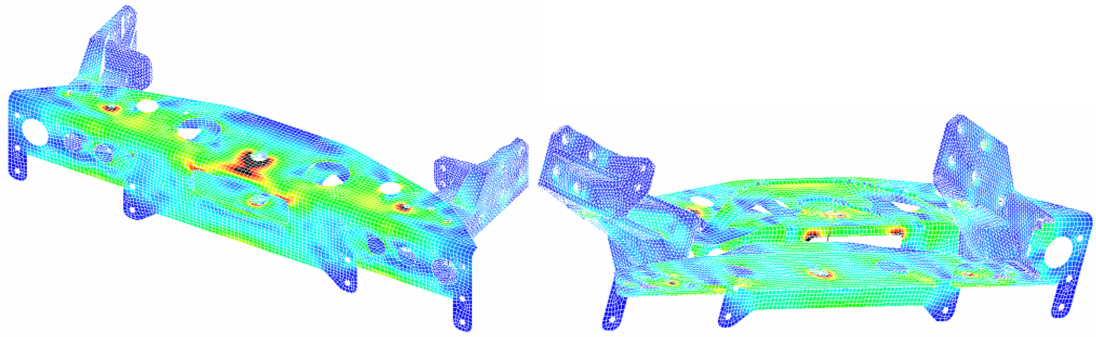


Figure 6.52 Single tow, Up 45º 158 kN .

6.6.4 **Dual tow**

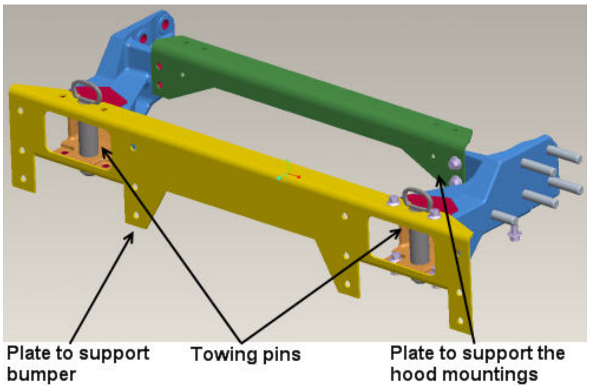


Figure 6.53 Dual tow, top front view.

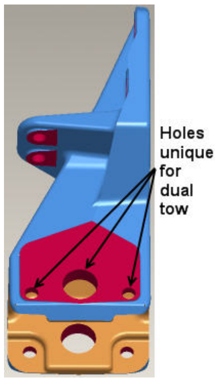


Figure 6.54 Dual tow, side bracket.

In the dual tow version the towing pins are placed in the two side brackets. This means a lot of weight has been saved in the centre section compared with the single tow version. The centre section consists of a single piece of sheet metal of 5 mm thickness and support only the bumper. The plate that support the hood mount is the same as for the single tow version. Appendix 4 includes more detailed pictures of the design.

6.6.5 **Result of Stress Analysis, Dual Tow**

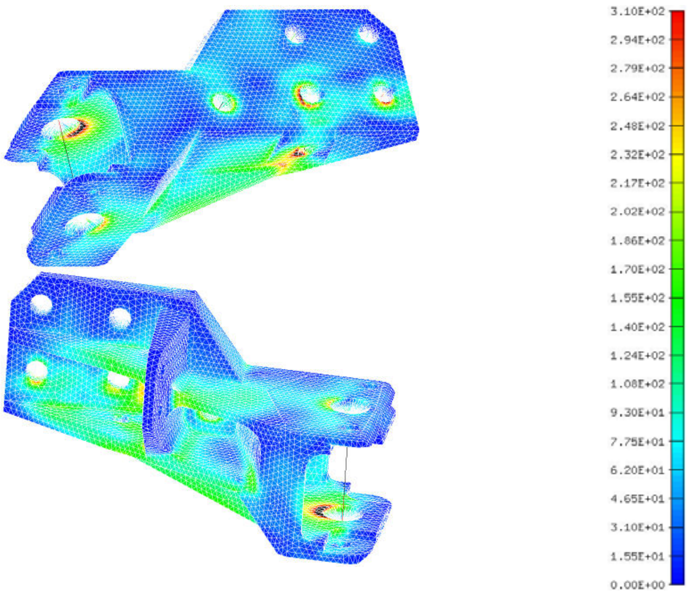


Figure 6.55 Dual tow, Forward pull 215 kN.

To save computing time only one side bracket were analysed. The sheet metal sections don't take up much load in this case so the results should be accurate enough. For further details regarding the analysis see chapter 5.2. The most critical case was the Forward pull shown in Figure 6.55. The structure is yielding in the areas around some of the boundary condition but is otherwise well below. The stresses are distributed over large parts and even the back to fasteners takes some load. More analyses are presented in Appendix 4 where a comparison between von Mises stress and Maximal Principle stress also can be seen.

7 Solution

The final solution is presented with hard core facts compared to the existing sheet metal solution.

The modular design solution using combined cast side brackets and sheet metal solves all the goals that were set up:

- **Modular design.**

  Only the center section is different between option A and B, single and dual tow, the other parts are the same. Both options works for axle forward and axle back, and for all bumper options.
- **Keeping a low part cost.**

  Castings have been used instead of sheet metal in side brackets to reduce part costs. Since the weight of the designs is kept to a minimum, material costs are minimized. The same moulds are used for the side brackets for both towing options to cut tooling costs.
- **Keeping a low assembly time when mounting to surrounding parts.**

  The modular system improves assembly issues since it makes it possible to assemble part by part or as one pre-assembly depending on surrounding parts and assembly line preferences:

  ◊ Assembling part by part keeps the number of fasteners to line up to a minimum and independent of the tolerances of the frame width. Fewer fasteners to line up make it easier when there are many parts using the same fasteners. By mounting the center section and hood mount support afterwards the width of the frame is locked.

  ◊ One pre-assembly means a single assembly step that locks the width of the frame.
- **Reduces number of parts.**

  The single tow contains 8 parts in total (2 cast side brackets, 4 sheet metal plates in center section, 1 hood mount support and 1 tow pin).

  The dual tow contains 6 parts (2 cast sides, 1 sheet metal plate in center section, 1 hood mount support and 2 tow pins).
- **Fitting within the available space.**

  No changes on surrounding parts have been made and sufficient clearance has been added.
- **Keeping the interface with frame, bumper and hood mountings.**

  Only existing fasteners are used when mounting design option A or B to the frame. Interface with bumper and hood mounts are maintained.
- **Fulfilling the TMC:s towing requirements for both single point and dual point towing.**

  FE Analysis in I-Deas has shown that both designs have acceptable stresses for the given loads. For the cast side brackets Spherodial Graphite Iron 0722 (STD 1107,22) was used and for the sheet metal plates Steel 2132 (STD 1121,32).

The modular design was sent to the cast manufacturer for quoting. The answer for the cost of the single tow option is not yet final and can not be given. Table 7.1 summarizes the result of the modular design compared to the existing design in sheet metal for the US07 project.

Table 7.1 Modular design compared to existing soultion.

Hard Core Facts for Modular Design Solution			
	Weight	**Parts**	**Price**
Dual Tow			
Modular	46 kg	6	$ 215
Existing	64 kg	13	$ 414
Single Tow			
Modular	95 kg	8	$ 435
Existing	96 kg	13	-

Note that the dual tow has reduced the weight with 18 kg, equal to 28%. The cost is cut with $199, equal to 48%.

The single tow is not reducing weight more than 1 kg. Comparing the figures to the existing sheet metal design it is important to remember that the existing single tow has not yet passed FEA. The modular single tow has.

8 Discussion

The discussion is split into two parts, one for the modular design and another about the project in general. The assembly plant LPP gave their comments on our modular design solution and they are presented in the first part.

8.1 Discussion of Modular Design Solution

The modular design solution was presented to LPP, the Chassis assembly plant in Lancaster. They gave the following comments [7]:

- One problem found was that if the bumper extension already is an assembly, the 5th frame rail hack bolt can not be fastened as the hood mount as it was designed on the presented suggestion.
  (Our comment: The original thought of the design was that the pieces would be assembled at LPP, and then the hood mount could have been attached at a later step. This was adjusted in the last version so that the pieces can be put together in any order, as preferred by the assembly plant.)

- There does not seem to be any problems with the tooling.

- Good that the two rear holes at the lower frame side are taken away on our solution compared with today's. This gives the possibility to fasten the space plates in the rear holes and the spring hanger and then afterwards fit in the bumper extension. Then the space plate will not slide. As it is now they have big problems to get the space plates in the right position since it has to be fastened with the rest.

- Since the bumper extension would not be put together at LPP they did not see any problem in having one big cast design or a 3 pieces cast design. What they prefer now is a ready subassembly of the front end extension that locks the frame width when mounted.

The modular design with both single and dual option has passed FEA in I-Deas. It should be noticed that they are compared to the FEA for the existing designs that are made in the computer program Ansys. Mr Joel Bassani at the Chassis Structure Group has checked the set up of our analyses and approved them.

The modular design could be patentable. Mack Trucks are currently looking into the question.

8.2 General discussion

A strength problem that was found in most designs was the area around one of the frontal lower holes in the vertical side of the frame rail and the hole in the horizontal side. We were not allowed to change the interface with the surrounding parts but it is worth a notice for future designs, Figure 8.1.

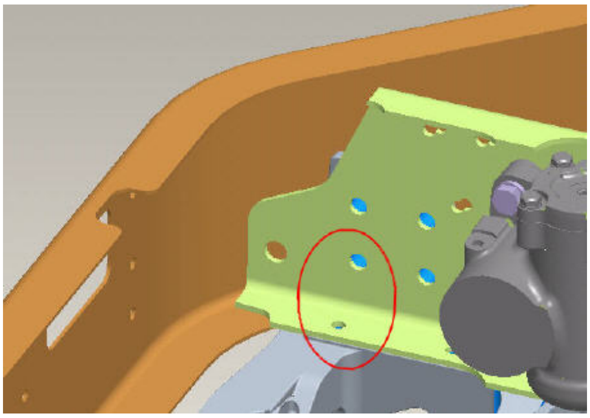


Figure 8.1 Problem area in the bolt interface of the frame rail.

Time limited us from continuing with further analyses and design adjustments on the single and 3 piece designs that was analyzed with material distribution in Optistruct. From what we have seen there should be possibilities to find lighter and stronger solutions. Again it is the bolt interface to the frame rail that is crucial for the single and 3 piece suggestions. The ring seems to be a difficult design to solve since it has the load center further out to the front than the pin which creates a larger moment on the bolt interface to the frame rail. Even though the designs are not yet finished there seems to be possibilities to find a single cast solution that reduces weight. On the single piece suggestions of this report the weight is around 20 kg lighter.

If continuing with optimization analyses size and shape optimization could be performed for fine-tuning. There are several applications in the software that can improve structures. Because of limited time, we did not get a chance to look into them.

If it would show in prototype testing of for example modular design that the side parts are not strong enough there could be a possibility to austemper the parts, i.e. increasing the temperature to create austenite. This would improve the quality. Speaking with Mr Arthur Fowler at the Development and Test Center at Mack Trucks in Greensboro, he estimated the cost to be $30 per casting with the size we have on our side parts. For the dual tow option that has $199 lower cost than existing designs this could be an alternative if it would be needed.

The hood load that was listed in chapter 3 showed a very little influence on the designs. Including the load in the analyses of the early designs we could see that it only gave minor stresses, nothing that needed to be analyzed further. If Mack would like to look into reducing more weight for the design suggestions of this report, the hood mount plate could be redesigned. Therefore, the hood load is left in the report even if they are not treated further.

9 Conclusion

The problem of this project has been solved with the modular solution that meets all the goals of the project. If Mack Trucks do not see a need for further FEA, the modular design would be ready for prototype testing.

For the design suggestions that was developed with the help of Optistruct, the result of the two single piece designs, Single-Ring or Single-Pin, shows so far that weight can be reduced. With the expressed preference on fewer parts and one subassembly to lock the frame width (LPP) we think it is worth to develop the idea of a single cast further. Looking at the FEA results, the most promising seems to be Single-Pin.

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Appendix 1

Single-Ring

Results of von Mises stress in I-Deas analyses:

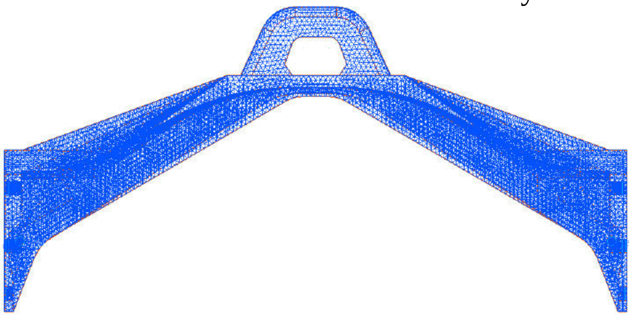


Figure A 1 Mesh. Single-Ring.

Number of elements: 20 333

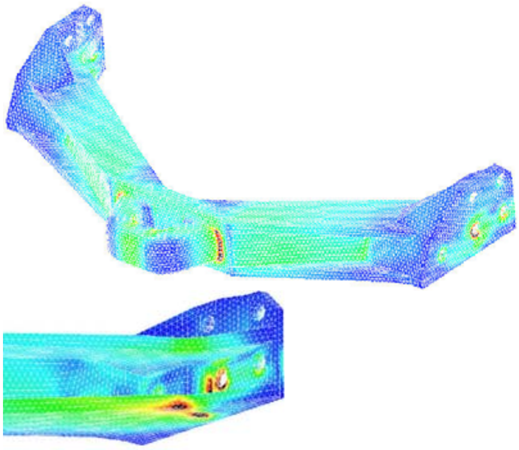


Figure A 2 Single-Ring, TMC's Forward pull (430 742 N). Top and back bottom view.

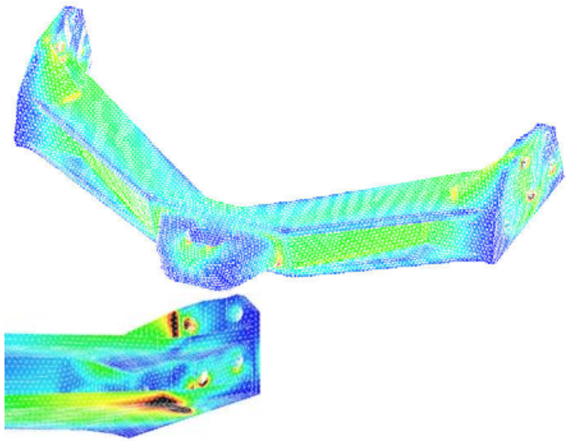


Figure A 3 Single-Ring, TMC's Cone Up 45º (157 612 N). Top and back bottom view.

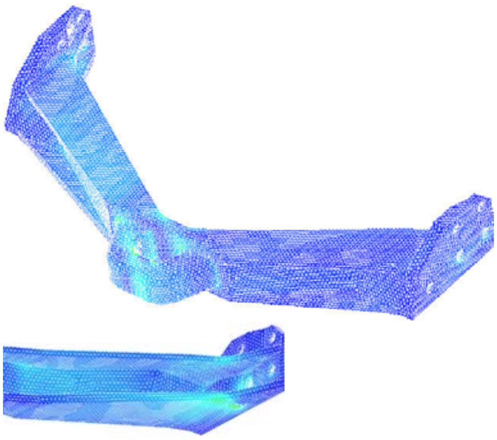
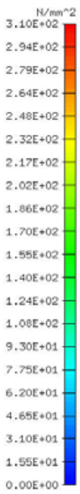


Figure A 4 Single-Ring, TMC's Cone Left 45º (157 612 N). Top and back bottom view.



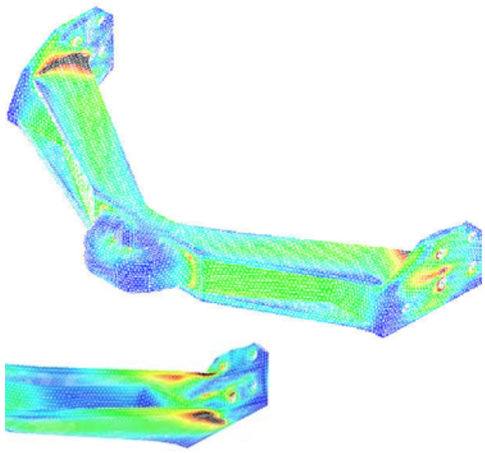


Figure A 5 Single-Ring, TMC's Vertical 90º (112 580 N). Top and back bottom view.

Single-Pin

Results of von Mises stress in I-Deas analyses:

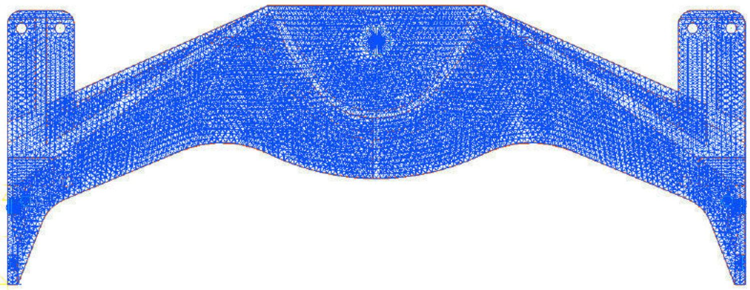


Figure A 6 Mesh. Single-Pin. **Number of elements: 40 553**

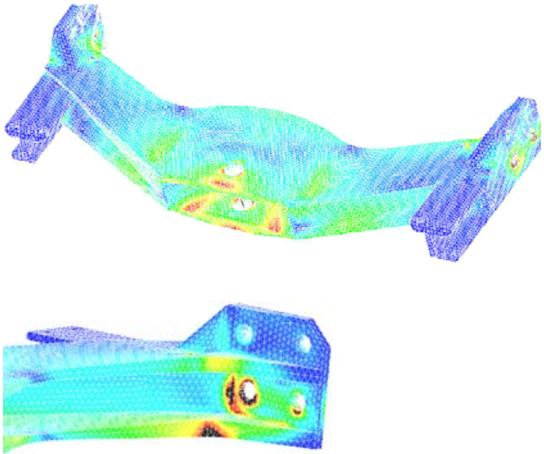


Figure A 7 Single-Pin, TMC's Forward pull (430 742 N). Top and back/top view.

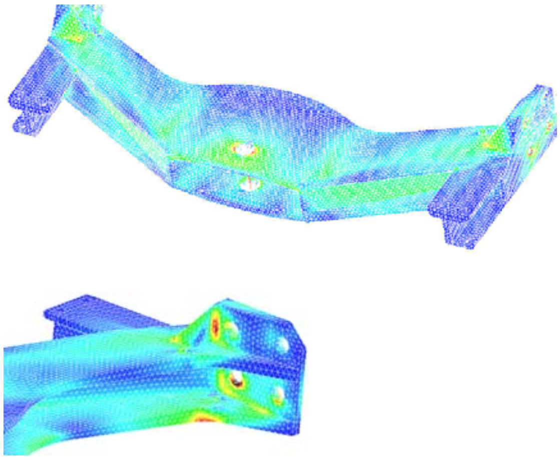


Figure A 8 Single-Pin, TMC's Cone Up 45º (157 612 N). Top and back/top view.

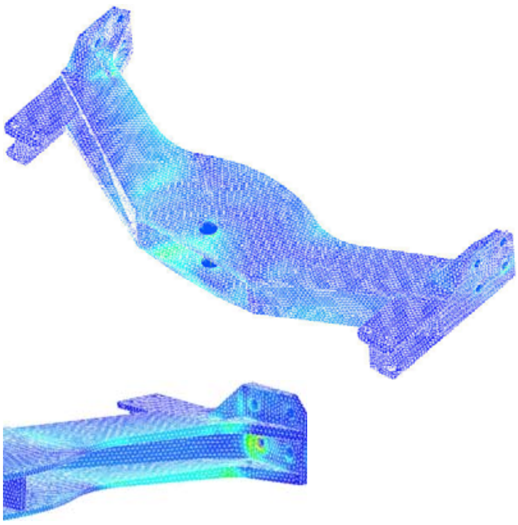
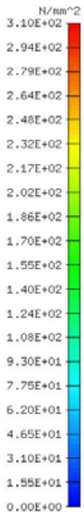


Figure A 9 Single-Pin, TMC's Cone Left 45º (157 612 N). Top and back/top view.



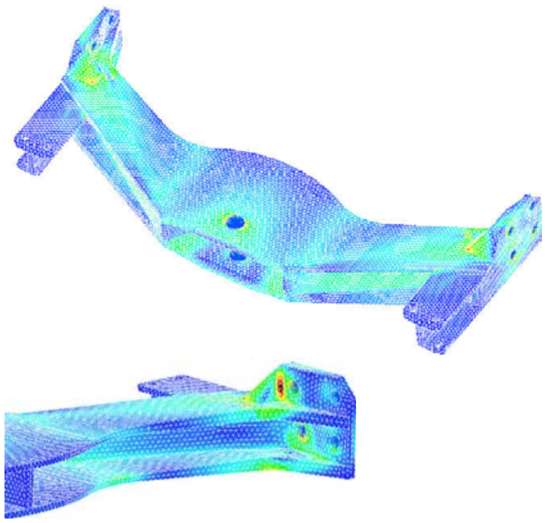


Figure A 10 Single-Pin, TMC's Vertical 90º (112 580 N). Top and back/top view.

3-Ring

Results of von Mises stress in I-Deas analyses:

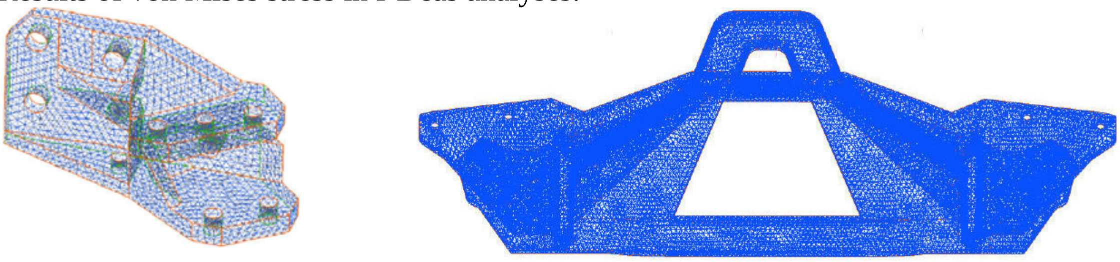


Figure A 11 Mesh. Side and center section in 3-Ring. Number of elements: 5 505 and 31 938

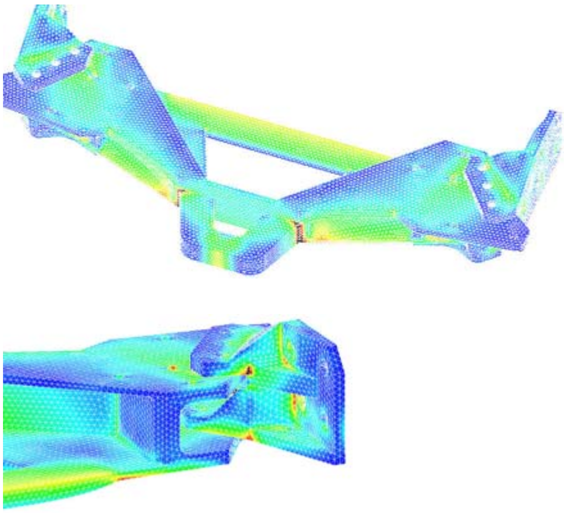


Figure A 12 3-Ring, TMC's Forward pull (430 742 N). Top and back/top view.

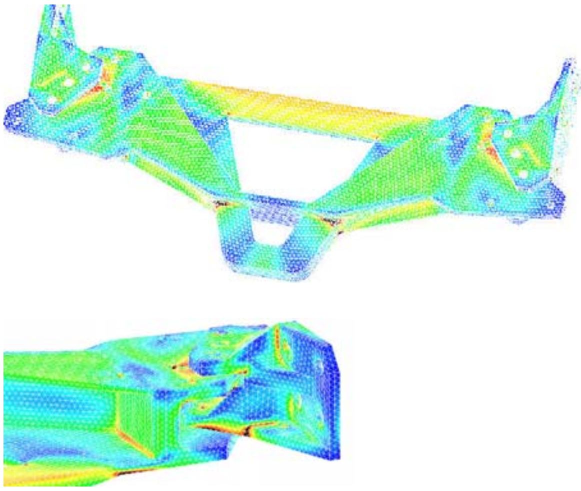
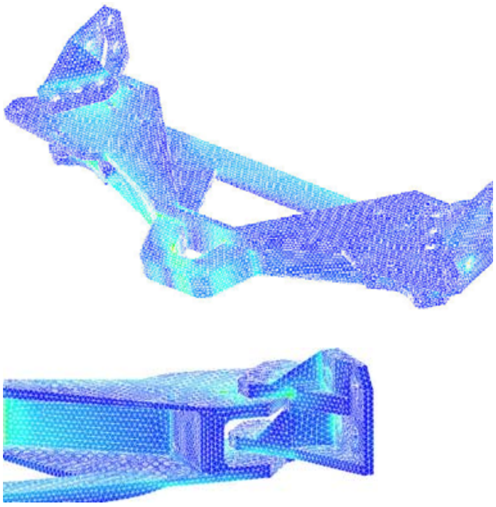


Figure A 13 3-Ring, TMC's Cone Up 45º (157 612 N). Top and back/top view.



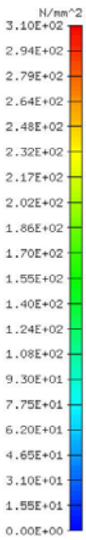


Figure A 14 3-Ring, TMC's Cone Left 45º (157 612 N). Top and back/top view.

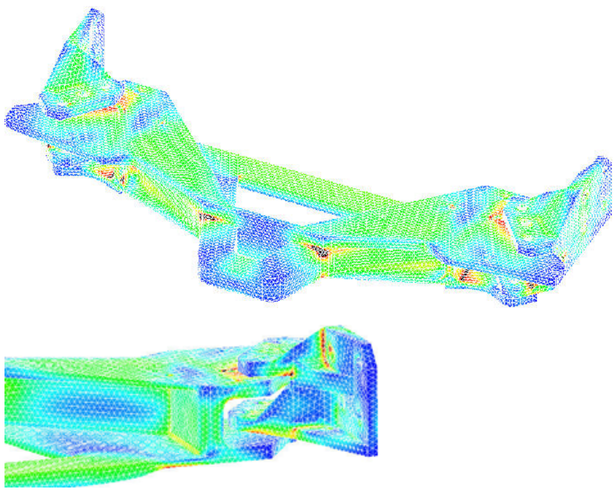


Figure A 15 3-Ring, TMC's Vertical 90º (112 580 N). Top and back/top view.

3-Pin

Results of von Mises stress in I-Deas analyses:

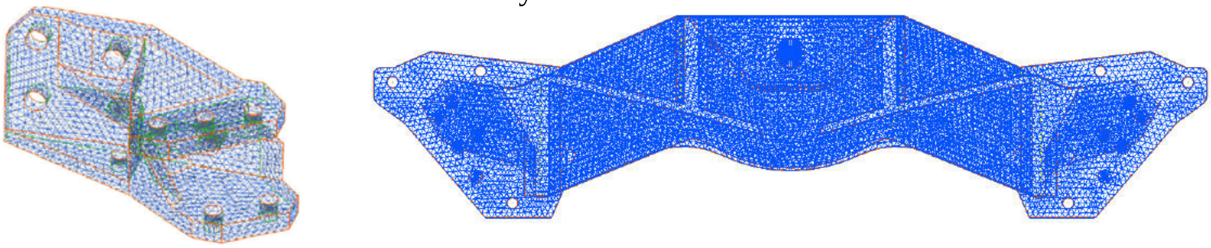


Figure A 16 Mesh. Side and center section for 3-Pin. Number of elements: 5 505 and 30 988

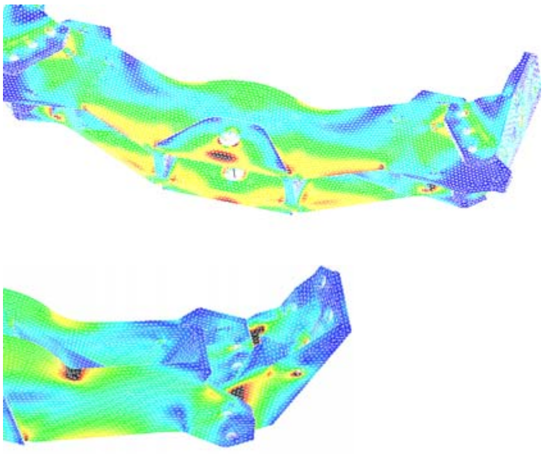


Figure A 17 3-Pin, TMC's Forward pull (430 742 N). Top and back/bottom view.

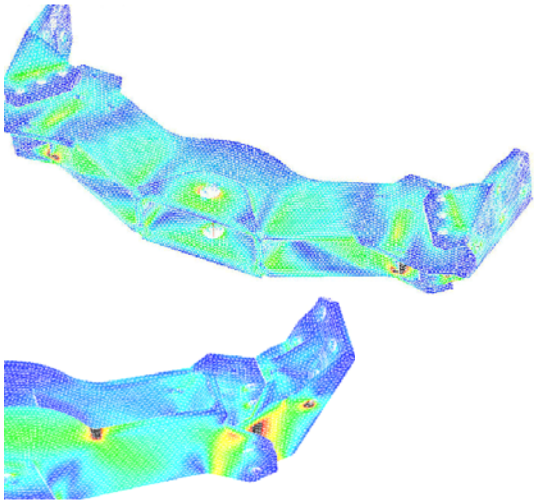


Figure A 18 3-Pin, TMC's Cone Up 45º (157 612 N). Top and back/bottom view.

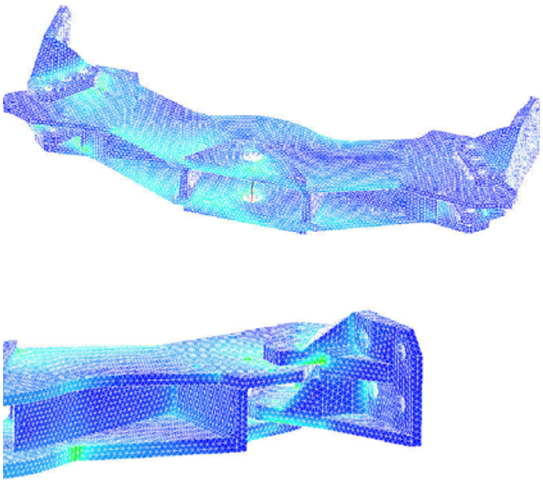
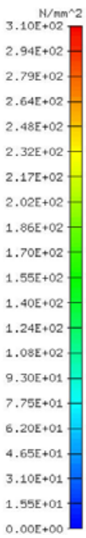


Figure A 19 3-Pin, TMC's Cone Left 45º (157 612 N). Top and back/top view.



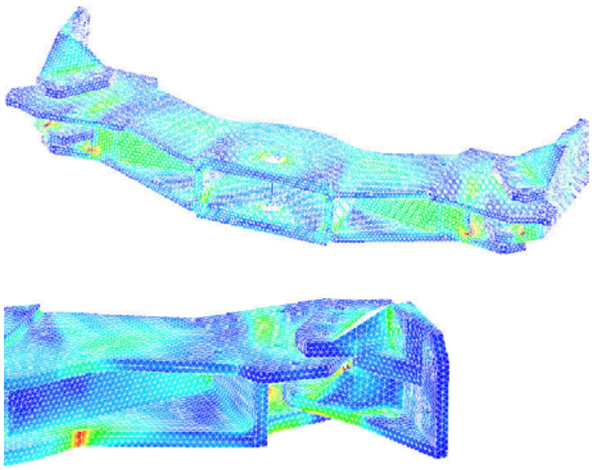
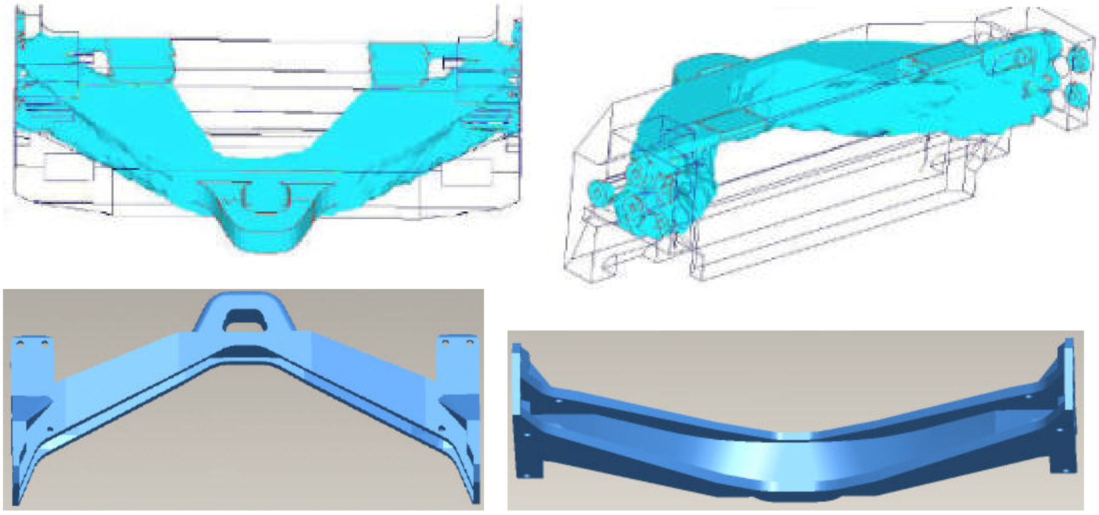


Figure A 20 3-Ring, TMC's Vertical 90º (112 580 N). Top and back/top view.

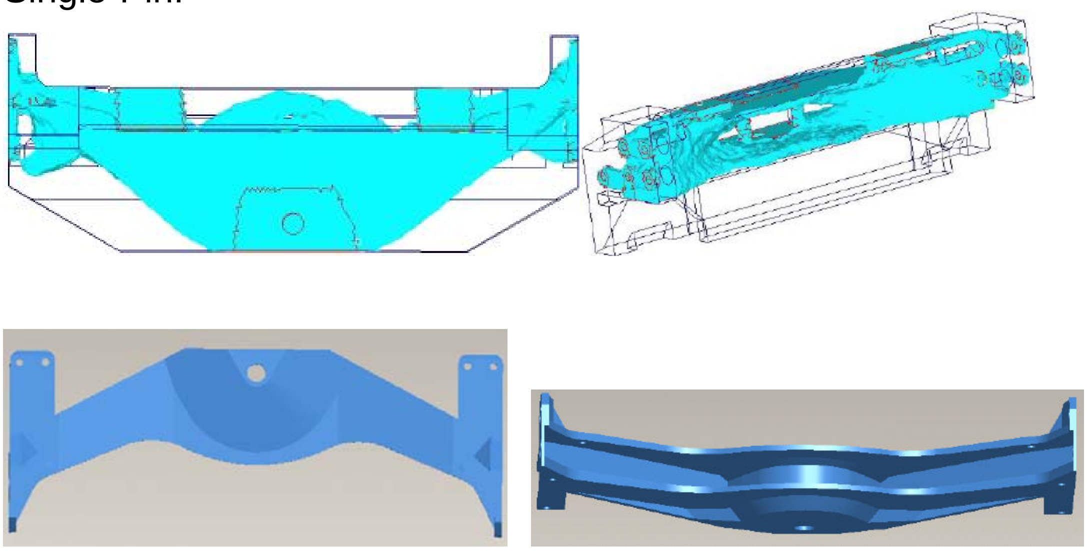
Appendix 2

Additional views of Optistruct Results and CAD-models from ProEngineer.

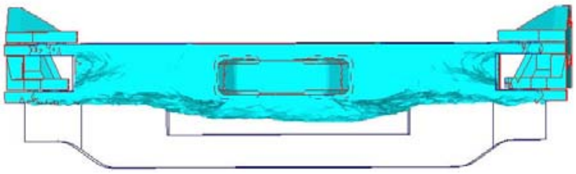
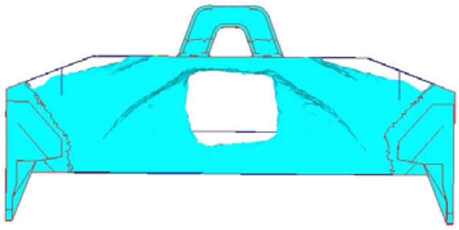
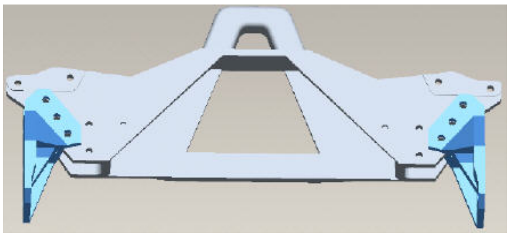
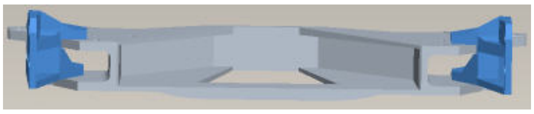
Single-Ring:



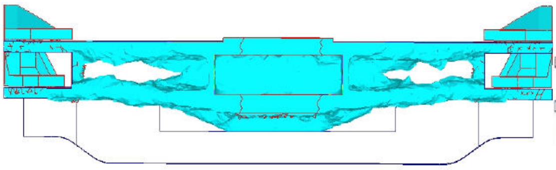
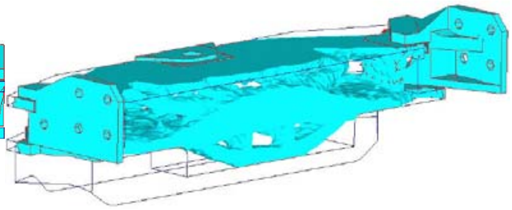
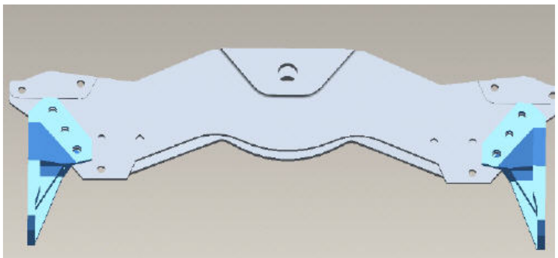
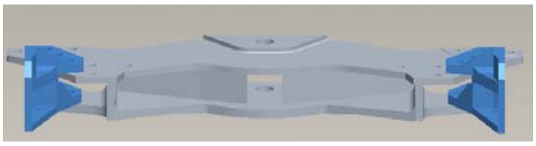
Single-Pin:



3-Ring:

3-Pin:

Appendix 3

Modular design, single tow.

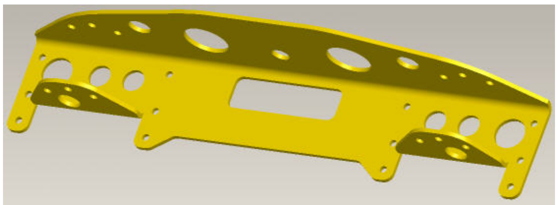


Figure A 21 Single tow, plate to support the bumper, 9.5 mm thick.

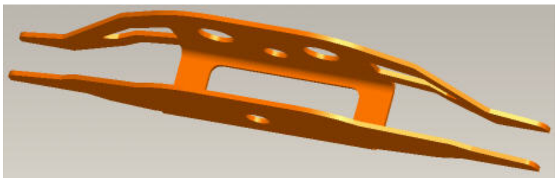


Figure A 22 Single tow, pin reinforcement 1. 7.9 mm thick.

Figure A 23 Single tow, pin reinforcement 2. 12.7 mm thick.

Figure A 24 Single tow, pin reinforcement 2, 12.7 mm thick.

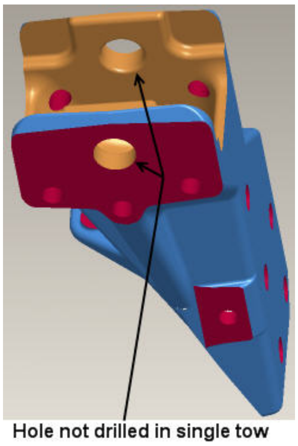


Figure A 25 Single tow, side bracket

Results of von Mises stress in I-Deas analyses.

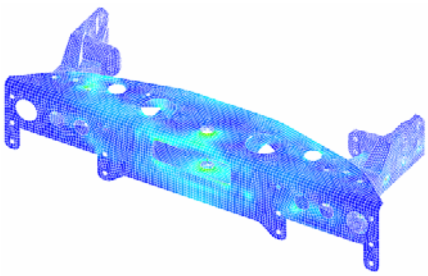


Figure A 26 Single tow, Pull 45° left

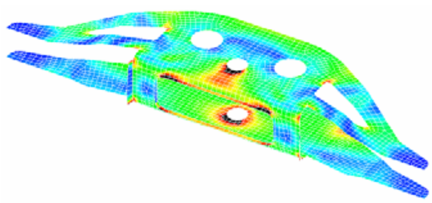
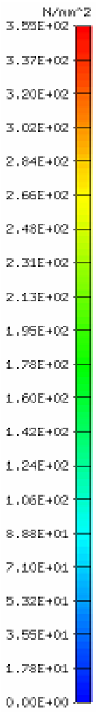


Figure A 27 Single tow, reinforcement 1, forward pull



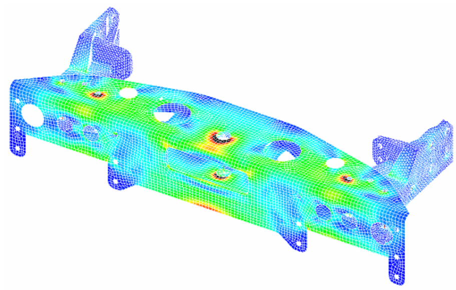


Figure A 28 Single tow, up 90° pull

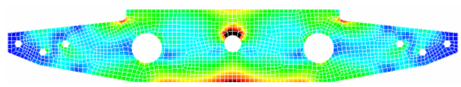


Figure A 29 Single tow, reinforcement 2, forward pull

Modular design, dual tow.

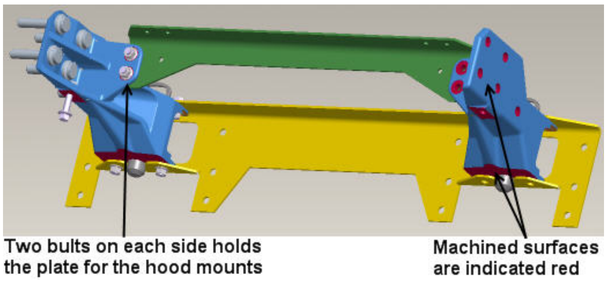


Figure A 30 Dual tow, bottom back view

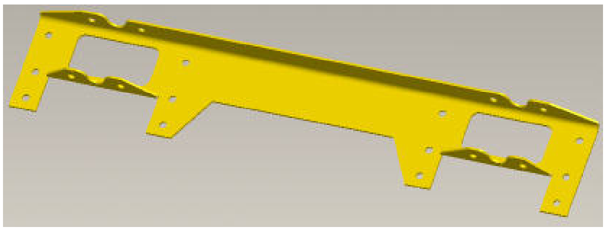


Figure A 31 Dual tow, plate to support the bumper

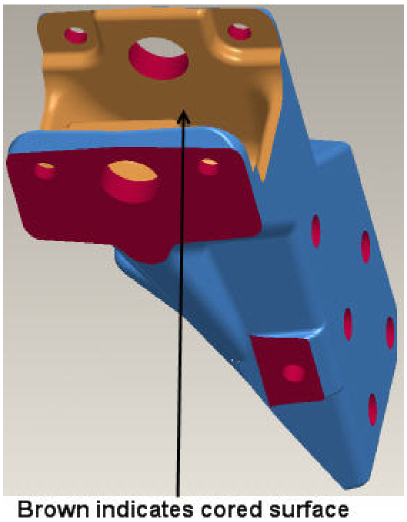


Figure A 32 Dual tow, side bracket

Results of von Mises stress in I-Deas analyses.

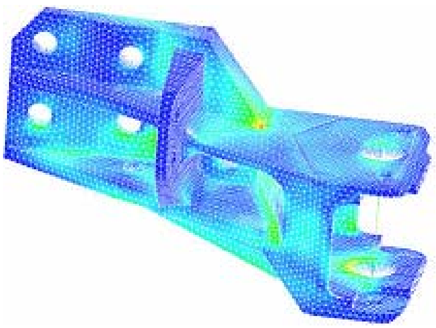


Figure A 33 Dual tow, up 45° pull

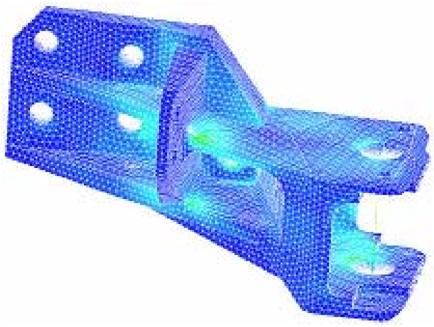
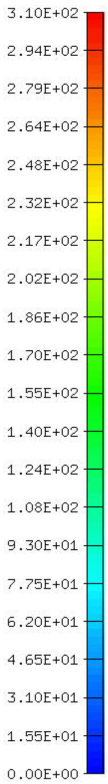


Figure A 34 Dual tow, in 45° pull



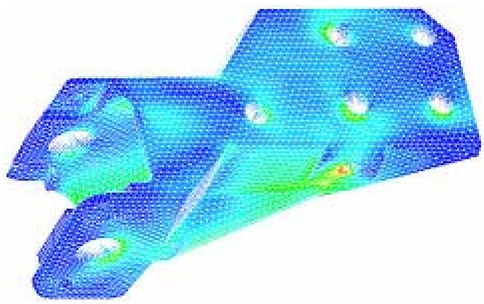


Figure A 35 Dual tow, up 45° pull

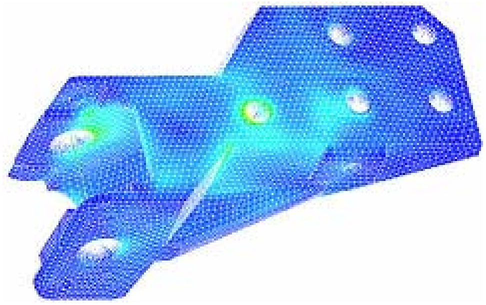


Figure A 36 Dual tow, in 45° pull

Comparison between von Mises stress and Maximal/Minimal Principle stress for dual tow, side bracket.

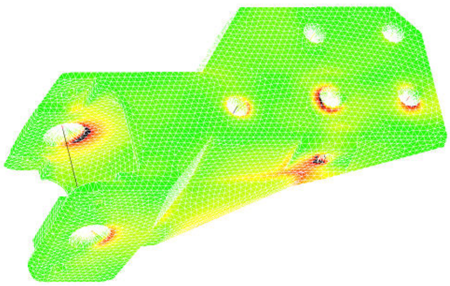


Figure A 37 Forward pull, von Mises stress

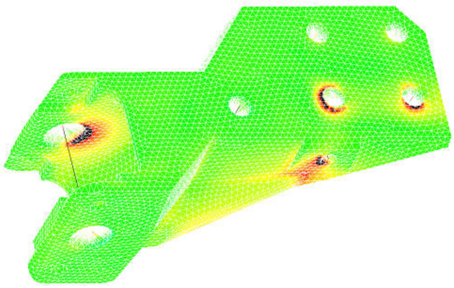
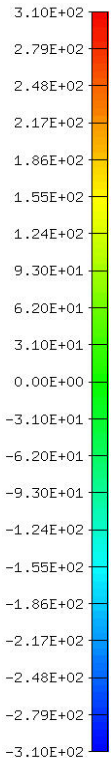


Figure A 38 Forward pull, maximum principle stress



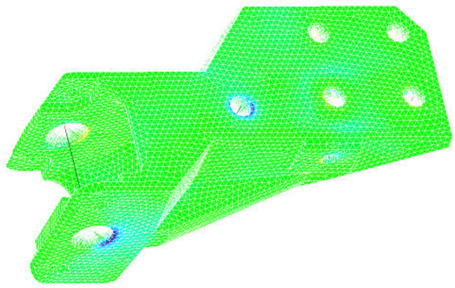


Figure A 39 Forward pull, minimum principle stress