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Development of an Active Variable Anti-Roll Bar

A Product Development Project including Design Optimization and Material Selection

Master's thesis in Product development and Materials engineering

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Department of Industrial and Materials Science
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2020

MASTER'S THESIS 2020

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Cover: Timber truck, a typical usage area for the active anti-roll bar.

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Abstract

When a truck is driving through a corner, the body of the truck rolls with regards to the frame. An anti-roll bar, ARB, exist on vehicles to control this rotation by contributing to the truck roll stiffness. In the current trucks, the roll stiffness from the ARB is constant and even though it exists variable ARBs on passenger cars there is no known solution used on trucks.

This thesis project attempts to fill this gap and develop a variable ARB adapted for a heavy-duty truck. The aim of the project was to develop a concept of an active variable ARB that can change its contribution to the roll stiffness of the truck.

The main approach with the methodology was a product development process performed in four iterations. In addition to this method, engineering optimization is applied and material selected.

The project work has found a way to vary the contributing roll stiffness from the ARB on a heavy-duty truck. This solution moves the attachment point where the stabilizer stay is connected to the frame towards the centre of the truck. By doing this, the contributing roll stiffness from the ARB is lowered without changing the characteristics of the existing ARB. The best-suited material for the selected concept is ADI for high performance, low weight and cost.

Keywords: Anti-roll bar, optimization, ADI, Volvo

Acknowledgements

This section is dedicated to expressing our gratitude to the many people who kindly have provided essential assistance throughout this master's thesis. We would like to thank our supervisor at Volvo, Anders Olsson. Thanks for giving us the opportunity to do this project and providing such an interesting topic for a product development project. We have learned much about the complexity of a development project for a new component and also a deeper understanding of product development regarding trucks. Further, we would like to thank all co-workers at Volvo GTT for all the valuable input and feedback on the concepts during the meetings.

We would also like to show gratitude to our supervisor and examiner at Chalmers, Gauti Asbörnsson for all the input from an academic perspective. Thanks for following up on the project and keeping us on the right track.

Max Gålnander, Pontus Karlsson, Gothenburg, June 2020

Contents

List of Abbreviations

ADI Austempered ductile cast iron

ARB Anti-Roll Bar

DFA Design For Assembly

DFM Design For Manufacturing

DI Ductile cast iron

HAZ Heat Affected Zone

MF modeFRONTIER

Nomenclature

δ Deflection, [mm]

ϕ Roll angle

ρ Density [kg/m^3]

σ_u Fatigue limit, [MPa]

C Stiffness of an ARB, [kN/mm]

C_m Cost per mass, [SEK/kg]

C_s Roll stiffness of an ARB, [kNm/rad]

E Young's modulus, [GPa]

S_{max} Max stresses, [MPa]

1

Introduction

This chapter will cover the background and purpose of the conducted research. Furthermore, the aim and scope of the project are described and the issue under investigation alongside the limitations for the project is specified.

1.1 Background

The roll stiffness is one of many important aspects to ensure safe and efficient transportation with heavy-duty trucks. The Anti-Roll Bar (ARB) stands for the main contribution to the roll stiffness in the rear suspension of a truck with air-spring suspension.

The ARB connects the frame with the suspension on the left and right side and acts as a torsion spring when the body wants to roll. The ARB prevent the suspension of the wheels on the left- and right-hand side of the vehicle to function independently from each other. A high torsion stiffness in the ARB results in reduced body roll of the vehicle, leading to increased traction in high-speed cornering but is less comfortable for the driver on bumpy roads. Low torsion stiffness is wanted on straight and bumpy roads to allow the wheels to always be in contact with the road but is negative at high-speed cornering. Because of this, there is a trade-off between stiff and soft ARBs depending on the anticipated area of usage for the vehicle [1].

In certain heavy-duty truck applications, there could be an advantage to actively vary the roll stiffness of the ARB. This is a typical situation for a timber truck. A full load of timber results in a high centre of gravity that is causing high roll moments when driving at curvy roads. In this case, the truck needs an ARB with high stiffness to prevent the truck from rolling over in corners. When the timber truck is unloaded, the truck needs to reach the timber loading site, often located at the end of a rough forest road. The high stiffness of the ARB, in this case, can prevent the truck from getting the needed traction and ride comfort.

Active variable ARBs are known to be used on cars on the market [1, 2]. In the current trucks, the roll stiffness of the ARB is constant and there is no known solution for a variable ARB used on trucks. This thesis project attempts to fill this gap and develop a variable ARB adapted for a heavy-duty truck.

1.2 Aim

The project aims to develop a concept of an ARB where the roll stiffness can be actively selected between two levels. The project will use construction optimization to find the best solution on how to change between two stiffness levels. The project will produce a CAD model of the concept for the actively varied ARB with a selected material.

1.3 Scope

The market should be examined and interviews with stakeholders will be conducted to establish knowledge about the product and what is expected of the new product. New concepts will be generated to cover the gap between the current state and the target. The concepts should solve how to shift between high and low level of roll stiffness and how this can be actively varied on an ARB. A design optimization software will be used to find the best concept on how to change the stiffness between a high and a low level on the ARB.

Using the right type of material is of importance to achieve desired properties in the ARB. The product should withstand the harsh environment under the truck and the material will, therefore, be selected accordingly. The materials in the final concept will be selected in such a way that both fulfils requirements for performance and cost. The material in the final concept should be manufactured and shaped with existing production methods. Since safety is important at Volvo the material used will withstand the safety requirements at Volvo regarding crash safety, fatigue and fracture.

1.4 Specification of Issue Under Investigation

The project is intended to investigate the following questions:

- How should the anti-roll bar be designed to allow the stiffness to be actively selected between one low and one high stiffness level?
- What material should be used for the best performance, low cost and low environmental impact?
- What parameters on the anti-roll bar is impacting the stiffness the most?
- What is the optimal solution with the selected design?

1.5 Limitations

- The project is not going to produce detailed drawings with tolerances.
- The project will not consider developing the concept to make it ready for production.
- There will be no detailed investigation on the cost for a full production of the final concept.
- The project is limited to the resources and computer programs provided by Volvo and Chalmers.
- The project will avoid altering the existing layout of the truck.
- The project will not investigate how the electrical control system for the stiffness change concept should function.

2

Theory

The theory chapter cover information about anti-roll bars which is important for the report. It also treats a description of the material theory and construction optimization used for the project.

2.1 Anti-Roll Bar

An ARB has the purpose to limit the roll on the vehicle. Roll is defined as the angle between the body of the truck and the axle of the truck, see Figure 2.1. By using a stiffer ARB the roll can be lowered and thereby allow for higher speed in curves on a levelled surface. The problem with a stiff ARB is that when driving on a rough road one wheel can lose traction due to that the ARB is trying to keep the axle levelled with the body.

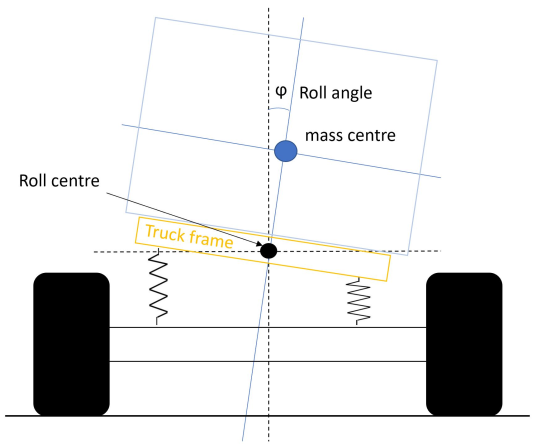


Figure 2.1: A Schematic figure of the trucks roll angle, the springs represent the ARB that counteract the roll moment.

Figure 2.2 shows a side view on the rear axis of a truck where the ARBs are highlighted in blue. In this figure, two ARBs are displayed and oriented towards each other. Each ARB is attached to one separate axle. Meaning if the truck has one rear axle there is one ARB and if the truck has two rear axles it has at least two ARBs. It exists examples where a truck has more ARBs than axles and this is to have an even further increased roll stiffness. This is especially needed in applications when a dumper truck is loading a large amount of gravel. When the dumper tilts the bed to dump the gravel, the centre of gravity is moved upwards and backwards. In such applications, a higher level of roll stiffness is needed to avoid the truck falling over to the side.

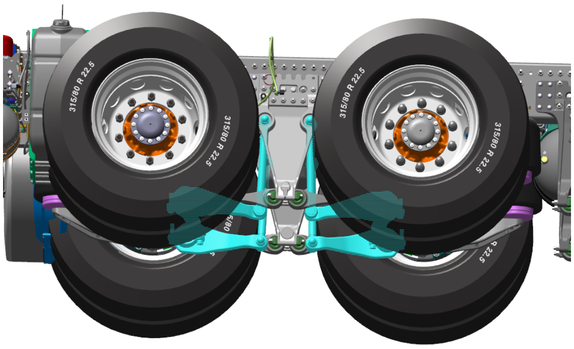


Figure 2.2: Side view of a rear axle on a 6x4 truck and the ARB highlighted in light blue colour

Figure 2.3 shows the most commonly used ARB on the rear axle of Volvo trucks. This ARB consist of two stabilizer stays, two arms and one torsion bar. When the truck rolls one stabilizer stay is pushed down and one is pushed up. This means that the torsion bar is twisted around its own axis and this is the basis of the roll stiffness. The rotation angle in the bar is the same as the roll angle, ϕ, from Figure 2.1.

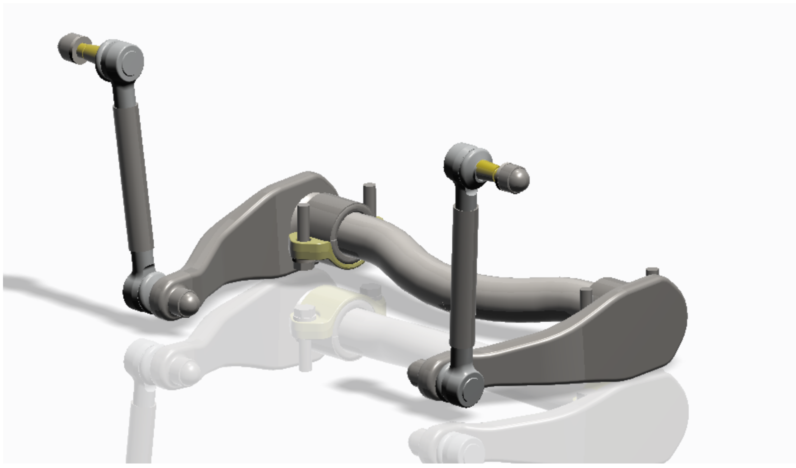


Figure 2.3: Today's ARB from Volvo, called GRAS 2.7

2.1.1 Parts on and Around the ARB

Figure 2.4 shows the parts that are included on a rear axle of a truck. The specific truck on the picture is a 6x2, meaning that it has 6 wheels, 2 on the front and 4 on the rear. Out of these 6 wheels, 2 are driven. The driven pair of wheels is located on the rear axis and has the suspension unit as seen in the figure. The important parts for the project are tagged with a number and have their respective name in the figure. These names for the different parts will be used in the report.

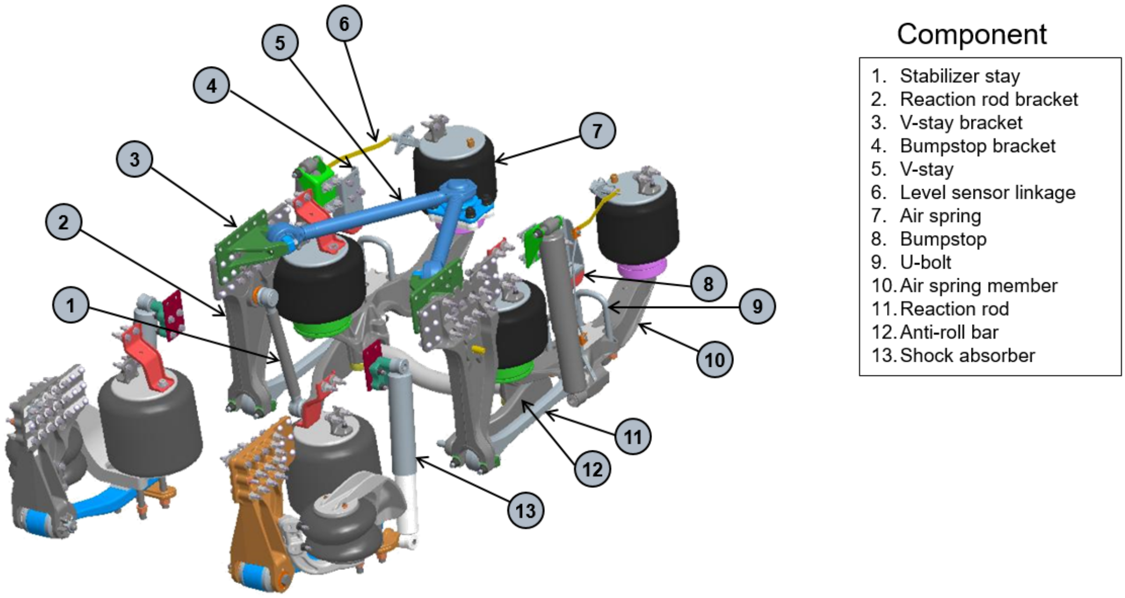


Figure 2.4: Component view of the rear suspension on a 6x2 truck

2.1.2 General Requirements on an ARB

The ARB is supposed to have a lifetime equivalent to the lifecycle of the truck. This corresponds to around $1,000,000 - 2,000,000$ km in road life. Therefore the ARB is subjected to repeated torsion and deflection which means that fatigue strength is a major requirement for the ARB. Since the ARB is located underneath the truck it has to resist water and dust. A critical point on the ARB is where the arm and stabilizer stay connect, which in some truck configurations is the lowest part of the vehicle. This means that the ARB is exposed to hits from rocks on the ground and needs toughness to avoid cracking.

2.1.3 Stiffness of an ARB

The stiffness on an ARB-system depends on several parameters: The length of the arms, stays and bar, the cross-section area of the bar and arm, the material, the position on where the ARB is mounted onto the axle and the frame respectively.

The stiffness of an ARB can be calculated with Brüninghaus formula, Equation 2.1. To get the stiffness per radian of rotation or roll stiffness, one can apply formula 2.2 [3]. The lengths used in the equations can be found in Figure 2.5.

$$C_s = \frac{3 \cdot E \cdot \pi \cdot d^4}{32 \cdot \left(2 \cdot l_0^3 + l_5^2 \cdot L_s + 2 \cdot l_4^3 + 3 \cdot \frac{E}{G} \cdot (l_2 \cdot l_7^2)\right)} \qquad [{}^{kN}/_{mm}] \qquad (2.1)$$

$$C_{s,rad} = \frac{C_s \cdot L_s^2}{2} \qquad [{}^{kNm}/_{rad}] \qquad (2.2)$$

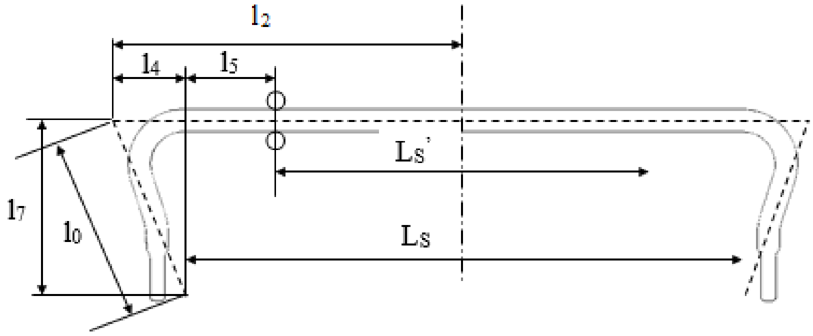


Figure 2.5: Lengths used to calculate the stiffness of an ARB

The correlation between the parameters in Equation 2.1 can be seen in Figure 2.6. In the correlation matrix, k is the stiffness of the ARB and as shown in the figure the diameter, d, is the variable that correlates the strongest with the stiffness. Variable m is the mass and does not contribute directly to the stiffness. The other variables contribute approximately the same to the stiffness.

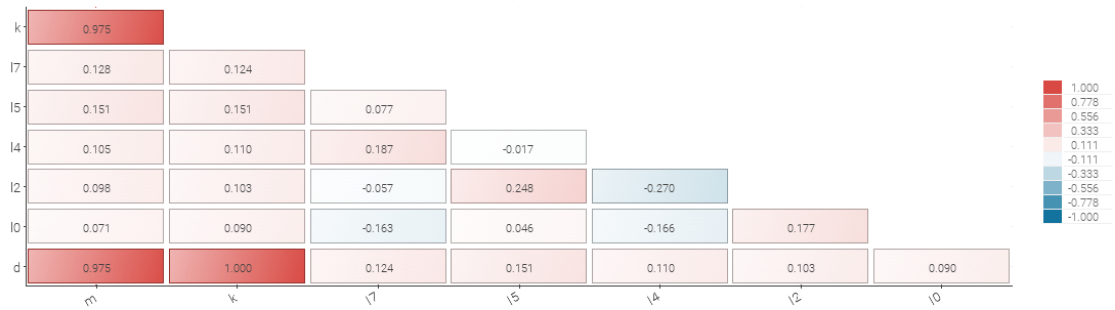


Figure 2.6: A correlation matrix of the different parameters in Brüninghaus formula

2.2 Problem Description

An active variable ARB will increase the traction on rough roads and increase the handling at high speeds [4]. The desired stiffness of the ARB is to be varied between a high and a low setting depending on the driving conditions. The stakeholder, Volvo, wants a solution that can change stiffness level by a factor of 2. The stiffness can either be varied in two stiffness levels or be varied actively between several stages. The active stiffness variation can be step-less and connected to sensors that scan the driving conditions and automatically adapt the setting of the ARB stiffer or softer. The latter is more a technically advanced solution that might be more up to date since it can interact with all advanced sensors in the vehicle. The former one, that only can vary between two stiffness settings is an older type of solution but can also be more robust since there is less technology that interacts and therefore less probability for things to break.

2.3 Existing Solutions on the Market

Today there exists no active variable ARB for a heavy-duty truck according to Volvo. However, it does exist some solutions for passenger cars. Some examples are the Active Roll Control from Schaeffler and the Active Stabilizer System from BWI [2, 5]. There also exist other solutions on the market, especially for rally cars that allow a change in stiffness by adjusting the length of the arms or turning the arms 90 degrees to adjust the moment of inertia. Figure 2.7 shows these two designs. These designs are not active though, instead, they need to stop the vehicle and adjust the ARB manually.

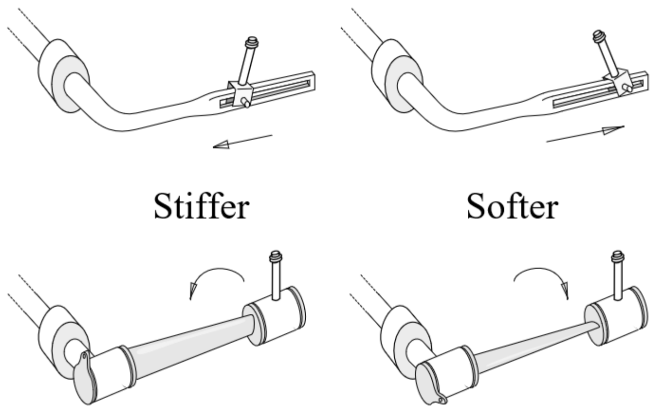


Figure 2.7: Two types of adjusting the stiffness of an ARB [6]

It exists some patents on how to achieve a variable ARB on a heavy-duty vehicle but none have been applied to a truck on the market. This indicates that many manufactures want to solve the same problem but none have found a good enough solution. One example of an investigation in variable stabilizers for trucks is the master's thesis by A. Carlsson and E. Svensson [7]. They investigated the possibility to change the stiffness of an ARB by applying different diameters of the bar on the ARB. The thickness is varied by having several tubes inside each other that can be connected with splines. They concluded that the splines intended to use in the design are difficult to design and the risk for failure is impending.

2.4 Todays ARB and Material Theory

The current two types of ARB at Volvo trucks are the newer GRAS 2.7, see Figure 2.3, and the older GRAS PC04. The GRAS PC04 ARB consists of a solid bar, being bent to shape the back bar and two lever arms, similar to the ARB that can be seen in Figure 2.5. The GRAS 2.7 ARB instead consist of one hollow tube that acts as the torsion bar of the ARB. On each side of the torsion bar lever arms are joined to the bar with welds. The newer version, GRAS 2.7, was developed because it was not possible to bend a solid bar with such small radius so that it both fitted between the reaction rods and could be fastened in the mounting bushings, see Figure 2.8 of the GRAS 2.7 ARB's position on the truck to understand the difficulty of fitting the ARB in the available packaging space on the truck. The packaging space is also limited by ground clearance, as indicated in Figure

2.2, the ARB is one of the components closest to the ground. Therefore, the ARB cannot be located much closer to the ground because of the required ground clearance.

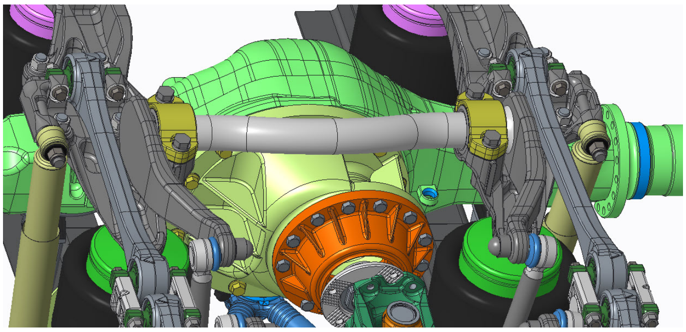


Figure 2.8: The packaging space of the GRAS 2.7 ARB underneath the truck. The ARB is located between the reaction rods and underneath the green wheel axle. The mounting points of the ARB to the wheel axle are yellow.

The arms of the GRAS 2.7 ARB are wrought to shape in a high strength steel, the steel has good forming properties in its annealed state and also has the ability to be hot forged. The result after finished processing is a material with a high yield- and ultimate tensile strength while still retaining ductility and elongation. The selected steel also has good weldability so that the arms can be joined together with the bar of the ARB.

The torsion bar of the ARB is a hollow tube that is manufactured from very high strength steel. The very high strength of the material is required because of the relatively short length of the torsion bar and the high loads it is subjected to.

The forging process of the arms increase the overall mechanical properties compared to if the arms would have been produced from casting or machined to shape [8]. Forging increase the strength, ductility and toughness of the material by deformation hardening and grain size refinement [9]. Porosities are closed and defects in the material are refined and aligned from the forging process, this generates a flow structure in the material which result in directional properties in the finished product [8, 9]. The flow structure can also act as a crack delay mechanism if produced properly [9].

The arms and bar of the GRAS 2.7 ARB are joined by welding. In general, welds and heat-affected zone (HAZ) are sensitive to cracks and can act as initiation places for fractures and fatigue [9, 10]. When welding, a HAZ is produced in-between the weld and the base metal [10]. The thermal cycle of welding can alter the microstructure in the HAZ in such way that hardness is both higher and lower across the HAZ profile compared to the base metal [10]. The grain size also often varies across the HAZ because the welding causes both recrystallization and grain growth [10]. The effect of the altered microstructure makes the HAZ a weak spot but the effects can be mitigated by preheating before welding and/or post heat-treatment of the welded area depending on material [9, 10]. Welds should be placed in areas with low stresses in dynamically loaded structures to avoid the risk of failure. To avoid welds and HAZ on the ARB in the area with the highest stress, in the connection between the torsion bar and the arms, the welds are placed with additional distance out from the centre of the bar to decrease the risk for crack initiation and failure.

It can be concluded that the process of developing the GRAS 2.7 ARB to fit in the available packaging space was a complex procedure of combining design, materials, processes and testing to verify the that the ARB would survive a lifetime on a truck with the high loads it is subjected to.

2.5 Design Optimization

Optimization means finding the best solution within the available means [11]. Design optimization depends on mathematical principles where you try to find the optimum solution to a specific problem. First, you have to set up the problem by stating the problem. The problem statement includes the objective function with parameters, design variables and constraints with parameters. The objective should preferably be written on negative null form, which means that the objective is to minimize towards zero. A typical optimization problem can be seen in Equation 2.3. In this equation $f(x,p)$ is the function to be minimized where x is the variable and p are the parameters, $g(x,p)$ and $h(x,p)$ are inequality respective equality constraint.

$$\begin{aligned} \underset{x}{\text{minimize}} \quad & f(x,p) \\ \text{subject to} \quad & g(x,p) \leq 0 \\ & h(x,p) = 0 \end{aligned} \tag{2.3}$$

If the problem contains two objectives that are corresponding to each other. The optimal solutions will be spread out alongside a Pareto front. A solution on the Pareto front means that there does not exist any solution that is better in one objective that does not make the other worse. Figure 2.9 illustrates a Pareto front on a problem where the objective is to minimize both objective 1 and objective 2. Each circle represent one solution to the problem and the ones that are the optimal solutions are the ones on the Pareto front. If choosing one of these you cant find a solution that is better on any objective without the other objective is impaired. [11]

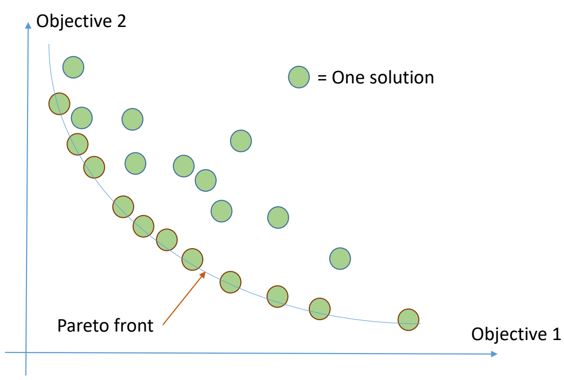


Figure 2.9: A schematic illustration of a Pareto front on a min min problem

3
Method

In the method chapter, the research methodology is described. The main approach is a product development process described by Ulrich and Eppinger in "Product Design and Development" [12] and Johannesson, Persson, and Pettersson in "Produktutveckling, effektiva metoder för konstruktion och design" [13]. In addition to this method engineering optimization is applied as described in Principles of optimal design: modeling and computation [11]. The concept development is performed in four iterations and close contact with the engineers at Volvo is kept to make sure the product will suit its purpose. The flow chart in Figure 3.1 illustrates the outline of the methodology in the report.

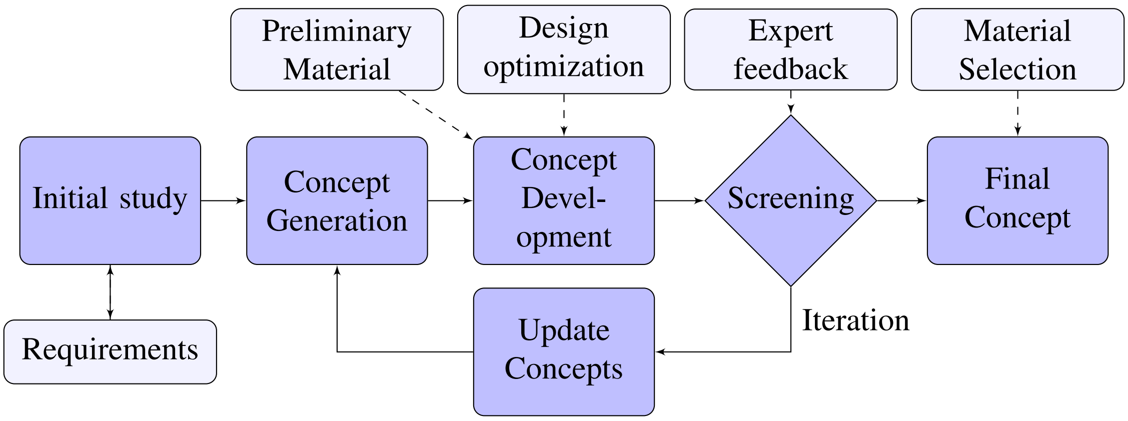


Figure 3.1: Flow chart of the methodology outline of the report

3.1 Requirement Chart

The purpose to begin with creating a requirement chart was to get information about the product that was to be designed. The requirement chart was made based on the methods provided in the book by Johannesson, Persson, and Pettersson [13] and in Product Design and Development by Ulrich and Eppinger [12]. The requirement chart consists of the product specifications that describe what the product has to do and are based on customer needs.

To understand the customer needs, meetings with employees at Volvo were conducted. The information from these meetings was combined with requirements on existing parts on the truck, especially the existing GRAS 2.7 ARB developed by Volvo together with Kongsberg. The requirement chart is a document that was constantly revised and developed throughout the project. The requirements that were set at the beginning of the project have to be updated when the product is tested. However, the scope of the project does not include to make the product ready for production and therefore the final product will not be tested in this project.

3.2 Concept Generation

The concept generation phase was iterated several times to generate new concepts when new information had emerged, this is done to gather as many feasible solutions as possible.

3.2.1 Brainstorming

The brainstorming was an idea generation of concepts on how to change the level of stiffness in the ARB. The brainstorming was conducted both in groups and individually. The brainstorming was conducted on two different occasions, named "Brainstorming 1" and "Brainstorming 2". Common for these two sessions was that ideas were sketched down on paper to quick and easy visualize the ideas.

Brainstorming 1
For the first iteration, the brainstorming began with sketching down as many solutions as possible. This part was done by individually sketching on paper for about five to ten minutes. Thereafter the ideas were explained in the group, and the solutions were compiled for further development. The other person in the group could then come with ideas on how to further develop the concept, both by combination with existing concepts and adding something to the concept. The group members were not to criticize the concept, this to create a creative environment and stimulate creative problem solving [13].

Brainstorming 2
For the second iteration, different categories on how to achieve a stiffness change on an ARB were created. Each category was then designated with five minutes each where the group members individually tried to brainstorm and come up with as many ideas on different ways to achieve a stiffness change within this category. The purpose of this is to try to find new ways to change the stiffness of the ARB without implying a specific technology [12]. It can sometimes be easier to think of solutions to a problem within a given framework. By narrowing the problem down the intention was to find more focused solutions with a higher level of detail and technical contribution [13].

The new solutions in brainstorming 2 were combined with the previously created sketches in brainstorming 1 to summarize the concepts generated in the brainstorming process. The concepts were then named and listed.

3.2.2 Elimination

All the concepts generated were summarized and screened in an elimination matrix. The elimination matrix used in this project is described by Johannesson in "Produktutveckling" [13]. This allowed for a first selection on the concept. The concepts were evaluated in the following categories: Solves the main problem, meets the requirement on twice the stiffness, realizable, within the cost aspects and safe which means that there is always a roll stiffness even if the variable function breaks.

3.2.3 Concept Sketching

The remaining concepts were sketched more thorough in order to get a better understanding of how the concepts would work. According to Ulrich and Eppinger sketching is a fast and inexpensive method to express ideas and evaluating possibilities [12]. The sketching was divided up within the group and each concept. When the sketches were done they were scanned and printed out on large paper to visualize the solutions and the key features for every concept.

3.2.4 Construction Meeting, Concept Feedback

The construction meeting was a meeting with experts at Volvo to evaluate existing concepts and create new ones. The experts are a group of experienced engineers within the field. The engineers were presented with the problem and the sketches of the exiting concepts. With this information, they were encouraged to give feedback on generated concepts and asked if they could have any new ideas. With each concept, there was time to discuss the pros and cons. The purpose of this meeting was to get feedback on the existing concepts and possibly ideas on new ones to continue with the concept development. [12]

3.3 Concept Development - Iteration 1

The first development iteration consisted of one development phase and one screening phase. Information about how to optimize the way of changing stiffness was gathered and the alternatives were narrowed down.

3.3.1 Concept Optimization with modeFRONTIER

For the optimization, the program modeFRONTIER was used to get insight into what parameters affect the stiffness of an ARB the most. The purpose of this information was to use it to develop the concepts. This was done by evaluating the existing ARB in a beam model in modeFRONTIER together with the pre-processor ANSA. The base of the analysis was set up in ANSA and can be seen in Figure 3.2. The objective of the simulation was to maximize deflection with as little variable change as possible. Torque was applied to the roll centre and the direction of the torque around the x-axis was varied, this simulates the roll moment of the body and makes it possible to monitor how the ARB behaves when the load direction change. The goal is that the stiffness of the ARB will be independent of the direction of the force. The rotation in the roll centre was to be maximized, this rotation simulates the roll angle, ϕ, of the truck. The method to calculate ϕ used the following process: With the model pre-processed in ANSA, the result was calculated by Abaqus, written to a text file and read by modeFRONTIER. Then modeFRONTIER changes the parameters and Abaqus performs another calculation, the loop was iterated 1000 times.

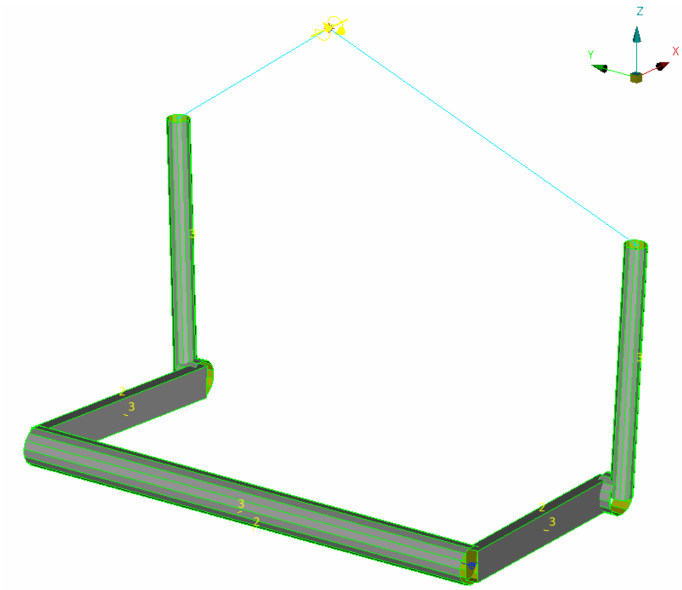


Figure 3.2: The base of the analysis in ANSA built up by variable beam elements. The roll centre is the yellow node connected with light blue rigid links

The optimization problem for this case means was set up with modeFRONTIER. The model is as previously described pre-processed in ANSA, the calculations are performed in Abaqus and the optimizer is in modeFRONTIER. In ANSA, the model puts on a load, load A. Then change to the soft setting and puts on another load, load B. The rotation around the x-axis is calculated by Abaqus is read by modeFRONTIER. The objective is to minimize the distance and the number of variables that change between the soft and stiff settings. The mathematical expression for the optimization is described in equation 3.1.

$$\begin{aligned} \underset{x_1,x_2,x_3,x_4,x_5,x_6}{\text{minimize}} \quad & f_1(x_1,x_2,x_3,x_4,x_5,x_6) = \sum_{n=1}^{6} |x_n| \\ & f_2(x_1,x_2,x_3,x_4,x_5,x_6) = -\phi(x_n) \\ \text{subject to} \quad & S_{max} - 500\,MPa \leq 0 \\ & |\delta_A| - |\delta_B| = 0 \\ & |x_4| - |x_5| = 0 \end{aligned} \tag{3.1}$$

The following parameters and variables are used in equation 3.1

Parameters:
δ_{pos} = Deflection in load case A
δ_{neg} = Deflection in load case B
ϕ = Roll angle
S_{max} = Max stress in load case A or B

Variables:
x_1 = Distance in X-direction of the connection between the stabilizer stay and the frame

x_2 = Distance in Y-direction of the connection between the stabilizer stay and the frame
x_3 = Distance in Z-direction of the connection between the stabilizer stay and the frame
x_4 = Length of right arm
x_5 = Length of left arm
x_6 = ARB arm rotation around bar

All the variables that change in the generated concepts were added to the model, to see which variable affect the stiffness the most and to evaluate which concepts that will be able to solve the requirement that the stiffness has to be changed by a factor of 2. The input variables that change the model in Figure 3.2 and the referred coordinate system in the table can be seen in the top right corner of Figure 3.2. An objective to minimize the variable change and an objective to maximize the rotation was added. This was done to find the least stiff solution with as little variable change as possible. The maximal stress in the ARB was constrained to be lower than 500 MPa to avoid plastic deformation during peak loads.

Figure 3.3 shows how the workflow in modeFRONTIER was set up for the analysis. The green squares are the input variables and the blue squares are output variables. The optimization algorithm runs horizontally in the figure. The optimization algorithm used is pilOPT which is suited for multi-objective optimization [14]. In the analysis, there are both input constraints and output constraints. This to accommodate how modeFRONTIER will read the optimization. The goal was to come as close to the mathematical expression in equation 3.1 as possible. The problem with setting constraints as an input parameter is that the algorithm will not know until after the calculation if it fulfils the constraint or not.

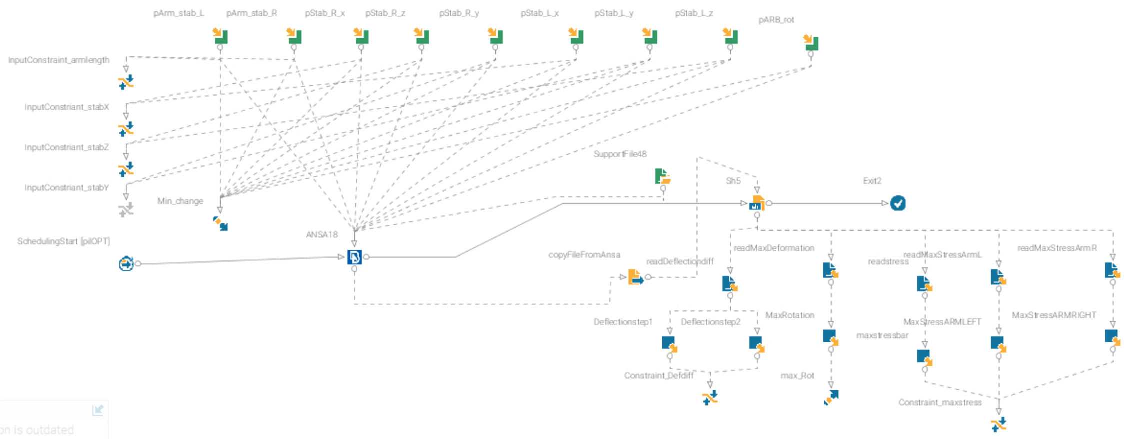


Figure 3.3: Workflow setup in modeFRONTIER

3.3.2 Concept Screening

To select concepts for the continued development, a Pugh matrix was performed as described by Stuart Pugh in the book Total design: integrated methods for successful product engineering [15]. The matrix ranks the concepts based on some predetermined criteria. All concepts are compared to one concept which is set as a reference. The matrix was iterated several times with a different concept as a reference for each iteration. The iterations stopped when the outcome from the matrices converged into the same result.

3.4 Concept Development - Iteration 2

To move on, more information about the concepts were needed. The purpose of this iteration was to learn more about the positive and negative of the concepts and to eliminate the least promising ones. This, in turn, will allocate more time to develop more promising concepts.

3.4.1 FE-analysis of the Concepts

An structural FE-analysis was done to get more information about the dimensions and how well the concepts will cope with the task. The concepts were analysed by a quick beam model to verify if it is possible to accomplish the stiffness change with the concept solution. Each concept was evaluated on stresses and displacements. The target was to find out how much each concept had to change to decrease the roll stiffness by 50%.

Every concept, except one, was modelled in the pre-processor ANSA with a beam structure that represents the geometry of each concept. A rotational load was applied at an approximation of the roll centre on the truck. This load was applied in both directions to make sure that the concept behaves in the same way when the truck turns to the right and left.

The Pencil, that use length change of the arms to adjust the stiffness, was evaluated by changing the original GRAS 2.7 ARB's arm length and bar diameter in different combinations. The evaluation was done in Creo Parametrics, using the built-in FEM application, Simulate. Creo Simulate uses the P-method to calculate the FEM results. The reason for using Creo Simulate instead of the ANSA beam models was because a more detailed result of the stress distribution was wanted since it was anticipated that the shorter arm length would result in high stresses at the radius between arm and torsion bar. This would result in permanent deformation and a fatigue fracture within a few cycles of loading.

The arm length and the bar outer diameter was varied between two different levels for *the Pencil*. The effective length of the long arms were 380 *mm* and the short arms 280 *mm*. The different bars were 70 *mm* for a large bar and 60 *mm* for a small bar. One arm was fixed and the other arm was subjected to a load of 70 *kN* to represent the roll moment of the truck that the ARB should be able to handle.

3.4.2 Screening of Concepts, Pros and Cons List

To collect the information about the concept and be able to eliminate concepts, a Pros and Cons List was created based on information gained from the FE-analysis. What further needs to be investigated is also listed in the Pros and Cons List. The idea is that the list will help to eliminate the least promising concepts. [16]

3.5 Concept Development - Iteration 3

To develop the concepts even further it was decided to design CAD models of each concept in Creo Parametrics to easily communicate the ideas at a new construction meeting and after that be able to eliminate even further.

3.5.1 Construction of the Selected Concepts

The construction began with a more careful discussion about how to design details, what the approximate dimension should be, and package the concepts on the Truck. This was done with a directed brainstorming on how to develop the concepts.

With the brainstorming as background, each remaining concepts were designed in the CAD-program Creo Parametrics. Compared to a sketch of the concepts like previously, a CAD model takes more time to create but consist of more information and will better visualize the ideas. With the new information about how the concepts will work it was considered positive to investigate the time to construct all remaining concepts to receive valuable feedback from the construction meeting with the experienced engineers. [12]

3.5.2 Construction Meeting 2, Concept Feedback

Each concept was presented shortly to the whole group, this followed by a discussion dedicated to each concept where both pros and cons were treated. The intention of the conduction of the meeting was to get information on which concepts to eliminate. With the concept designed in CAD, it enabled a better discussion with the experts compared to the previous construction meeting.

3.6 Concept Development - Iteration 4

The fourth iteration of concept development involved CAD-construction of the two remaining concepts. This time more focus was given to investigate packaging space, materials and stresses than before. The fourth iteration ended with a final decision on which concept to choose.

3.6.1 Detailed CAD Design of Selected Concepts

The feedback from the construction meeting enabled further construction of the selected concepts. As previous construction, the concepts were modelled in the CAD program Creo Parametrics. This time more focus was invested into some details and quick simulations were conducted to ensure that the concepts would not break.

3.6.2 Preliminary Material Selection

The preliminary selection and investigation for material and processing of for the concepts in this phase of the project were conducted with consultation from experts in the field at Volvo and Trelleborg. As described in the literature for material selection, a material selection for a product can be made based on evaluating what material is used in similar successful products [9, 13]. This approach is less time consuming compared to a thorough material selection for a new component where all possible materials are screened and ranked to get the best material candidate [9, 17]. The thorough method for material selection was used for the final concept in chapter 3.7.1.

Material Selection for the Y-Change Concept
In this preliminary material selection for the concept, the material was selected based on that the product should be manufactured using manufacturing techniques that are available for Volvo today. Expert competence at Volvo was consulted to evaluate how the concept Y-change should be manufactured the best way.

Material Selection for the Spring Strut Concept
Inspiration for *the Spring Strut* was taken from different dampening structures, the idea was to use some sort of dampening material in the strut. Literature suggests that elastomers are among the best materials for light springs [17]. It is possible to store around eight times more elastic energy per unit of weight in rubber compared to a spring steel [17]. Rubber has a dampening capacity that means that rubber is dampening resonance in application with dynamic loads [18]. Because there was a gap in knowledge about designing with rubber in the project group and at Volvo, contacts with Trelleborg was made. Trelleborg develops and manufactures rubber springs and dampeners for various applications, which is why they can be seen as experts in rubber [18]. The experts at Trelleborg were handed information about the loads, desired deformation, stiffness and packaging space requirements. The idea was to first investigate if rubber as a material can allow the wanted deformation with the given loads. Thereafter look into if there is a current product that could be used on *the Spring Strut* or if a new product needed to be developed.

3.6.3 Final Concept Selection

After design in CAD and feedback from Trelleborg and Volvo, enough information was gathered to make the final selection. This enabled a final selection to be made in consensus with the engineers at Volvo.

To gain knowledge about the best solution for the *Y-Change* concept, a topology optimization for minimizing compliance was prepared. The load applied is two times the peak load and the direction is changed to have one load up and one down to see how the model perform should look like with changed load case. The result from the topology optimization was interpreted into *the Y-Change* shape. One other requirement that was implemented into the part at this time is modularisation, the bracket has to be able to change for different ride heights. Since one of the main challenges with the construction is to handle all the stresses in the bracket, the CAD-model was constantly evaluated in the program Creo Simulate so that the stresses in the model did not exceed the yield point.

Change in Condition
Based on new information about how the forces impact the truck that was discovered during a consultation, the *Y-change* concept had to be redesigned. Due to time restrictions, no new topology optimization could be implemented, instead, the focus was located to design and analyze stresses in the new *Y-Change bracket*.

3.7 Design of the Final Concept, the Y-Change

The approach for the embodiment design process was to use existing parts as much as possible. If a part is not previously used by Volvo another manufacturer may have a similar product to the one that is desired. This means that a lot of existing parts can be used and by that cost reduced. Some parts had to be constructed from the beginning and made to fit this specific application.

The final solution was designed for a 6x4 truck. The reason to target 6x4 trucks is that most of the timber and gravel trucks have 6 axles. This is a segment that will have great use of an active ARB since the condition of roads they operate on varies a lot. A timber truck customer is also less cost dependent and is more prone to have extra accessories compared to a regular long haulage customer. The rear axle installation on a 6x4 is also the same on an 8x4 meaning the solution can be implemented directly on those vehicles as well. The idea is to later implement the same solution for all other chassis variants.

3.7.1 Final Material Selection

The methodology for selecting material for a product with conflicting objectives proposed by Ashby in Material Selection in Mechanical Design, [17], is used to decrease the material alternatives and end up with fewer plausible candidates.

To narrow down the types of materials available and have a manageable selection of material candidates, it was decided to use the software CES EduPack combined with different material indices [17]. Volvo desires a material that is both light, stiff, strong and has a low cost for this application. This posed an optimization and trade-off problem where CES EduPack with database level 3 was used to select material candidates that could live up to the requirements. The material indices were evaluated in the same procedure as described by Ashby [17]. The derivation of the material indices can be seen in Appendix A. The objective was to minimize mass and cost for the component. The constraints were both stiffness limited and fatigue strength limited and the free variable was the area. Equation 3.2 and 3.3 shows the material indices used for fatigue strength limited design and equation 3.4 and 3.5 shows the material indices used to for displacement limited design.

$$\text{Minimize mass } \Rightarrow M_1 = \frac{\rho}{\sigma_u^{2/3}} \tag{3.2}$$

$$\text{Minimize cost } \Rightarrow M_{1C} = \frac{C_m \cdot \rho}{\sigma_u^{2/3}} \tag{3.3}$$

$$\text{Minimize mass } \Rightarrow M_2 = \frac{\rho}{E^{1/2}} \tag{3.4}$$

$$\text{Minimize cost } \Rightarrow M_{2C} = \frac{C_m \cdot \rho}{E^{1/2}} \tag{3.5}$$

The material indices were derived from the assumption that *the Y-Change Bracket* is a 200 *mm* long beam fixed in one end and free in the other, where the free end is subjected to a bending force. The free variable in the equations was the cross-section area of the beam. The assumption that the area is free needs to be considered when evaluating the material candidates since this means that in theory the area can be very small or large but in reality, there is a limitation from the surrounding parts and interfaces. The derivation of these indexes can be seen in Appendix A.1 and A.2. The two most important factors for the part was a deflection less than $1-2$ mm when subjected to a load of 70 kN, that is the stiffness of the part. The other factor was that the material should have sufficient fatigue strength to survive the whole life of the truck. Therefore fatigue strength at N $= 10^7$ cycles is considered as the other important factor for the material index. These became the two constraints that the indexes were derived from. The free variable, section area was removed from the objective equation for mass and cost. It was replaced with an expression for section area using the equations for stiffness and fatigue strength. The whole derivation of the equations can be seen in Appendix A.1 and A.2.

A screening was set up to screen out any materials that could not live up to the requirements set by Volvo before any ranking or trade-off plots were made. These requirements were: Elongation of minimum 7 % strain, fracture toughness of minimum $K_{1c} = 15$ $MPa \cdot m^{0.5}$, durability in a saltwater environment (roads are salted on winter), a fatigue strength at 10^7 cycles were set of minimum 100 MPa, maximum price were set to 1000 $^{SEK}/_{kg}$ to screen out the most expensive materials. Because of the production volumes and the desire to achieve a near-net shape in few process steps, the casting was selected as a viable production process for the material. A summary of all the screening criteria can be seen in table 3.1 below.

Setting fracture toughness to above $K_{1c} = 15$ $MPa \cdot m^{0.5}$ is a rule of thumb for materials that should handle some mishandling and survive an impact [17]. The reason to use a limit on 7 % elongation was to allow the material to deform before fracture when subjected to extreme peak loads. This deformation allows for some energy absorption in the part without fracture and leaving the truck in an undrivable condition in case of a crash or driving down into a ditch. Breaking would leave the truck in a state where it potentially has no roll stiffness and would risk tipping over. After a severe impact, even if the truck is not fully functional, it should at least be able to get to a safe place.

CES screening criteria	**Limits**
Material price	max 1000 $^{SEK}/_{kg}$
Elongation	min 7 %
Fracture toughness	min 15 $MPa \cdot m^{0.5}$
Fatigue strength at 10^7 cycles	min 100 MPa
Metal casting ability	Acceptable, Excellent
Durability in salt water (seawater)	Limited use, Acceptable, Excellent

Table 3.1: First stage screening criteria for CES

The first plot was a coupling line plot to evaluate the dimensioning constraint [17]. First calculating the coupling constant. Five different coupling lines were created, the lines were calculated for maximum deflections of the beam between $1-5$ mm, the calculations and exact values on the coupling lines can be found in Appendix A.3.

The next step was to evaluate the most cost-efficient materials for the dimensioning constraints. After that, the trade-off plots were constructed where performance and cost for the material were the trade-off [17]. A line with a slope equal to -1 was plotted to evaluate the best materials where both cost savings and performance were considered as equally important. Then lines for exchange constants were plotted in the trade-off plots, these lines represented how much Volvo could be prepared to add in material cost if they were to save 1 kg of material weight on the final product [17]. The exchange constant is generally calculated from fuel-saving value and the fact that more pay-load can be added to the truck, but other factors such as how much the customer can be expected to pay for the weight saving also play a large role [17]. In this case a conservative estimate is done based on values from, Ashby - Material Selection in Mechanical Design [17]. These two lines are plotted for an exchange constant of 30 $^{SEK}/_{kg}$ and 10 $^{SEK}/_{kg}$, in this case, it means that a customer can be expected to pay between $10-30$ SEK per kilogram of weight savings [17]. The two exchange constants are plotted in the same trade-off graph to compare the materials they result in. The exchange constants also make it possible to evaluate if there is a non-dominated solution on the Pareto front that could be better than the reference material for similar products. In this case, Volvo uses ductile cast iron, GJS-500-7.

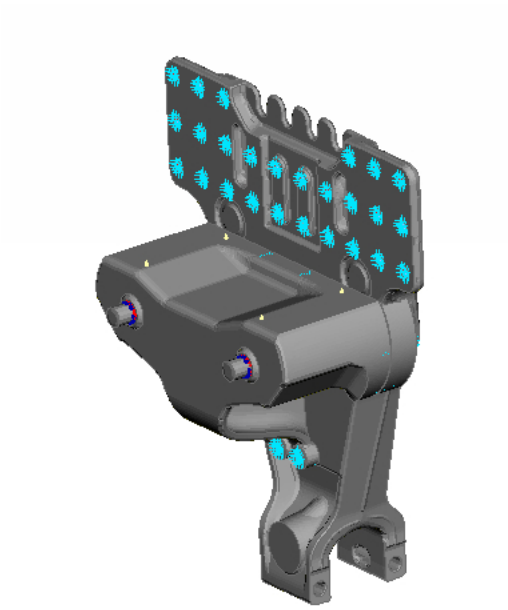
After the screening and ranking were done in CES, documentation was searched for to evaluate if the candidates would be better than the current reference material, GJS-500-7, that Volvo use for similar components to *the Y-Change Bracket*. The search for documentation was done by searching for information on the internet, in articles and literature. Consultation with Volvo materials technology was done to confirm the material selection and evaluate the best solution regarding cost and processing for the material.

3.7.2 Detailed Design of the Final Concept

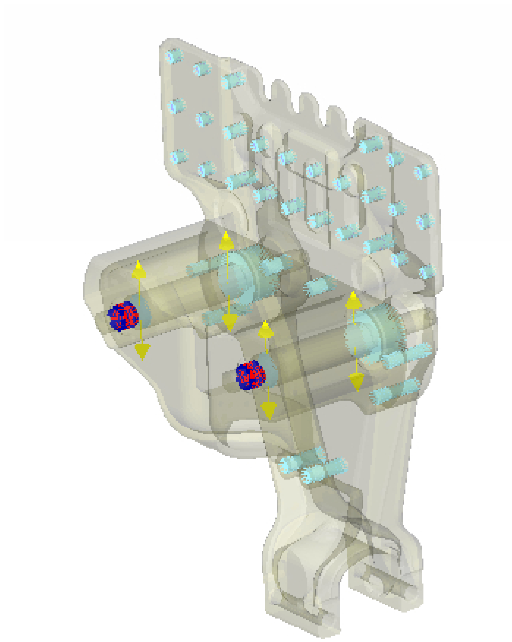
The CAD-model of the *Y-Change bracket* was created in Creo Parametric. Based on the bracket layout, the other parts could be constructed. Focus when the choosing parts were to ensure that the parts would fit together with Volvo's existing system. The detailed construction also included searching for existing parts and consolidation with other companies to investigate what they could offer. Contacts were made with Bosch and SKF. The challenge here was to give the right requirements to the contacts so that the quality of the received product is sufficient. Various internet sites were also scanned for products that could meet the requirements.

3.7.3 Structural FE-Analysis of Y-Change Bracket

ANSA is used to pre-process the model. Included in the analysis are the *Y-Change bracket*, the axles, and the reaction rod bracket. The materials that is applied to the model have the same properties as the selected material from the material selection in CES. The material data is loaded from the CES database into the ANSA model. The model was meshed with the batch mesh method in ANSA. Since there are two different stiffness settings that the concept should be able to handle the analysis is build up in different sets, one set represents one load case. This means that there are 6 different sets, three on each stiffness setting, and on each stiffness setting the load is applied in a combination of negative and positive direction as well as the only negative and only positive direction, the load vectors are indicated in Figure 3.4b. The load applied in the high stiffness setting, closest to the reaction rod bracket, is 70 kN and in the low stiffness setting the load applied is 42 kN this due to that the reaction forces are lowered to 60% when the stabilizer stays are moved 100 mm towards the centre of the truck. These values are based on the results from a beam analysis where the difference in the section forces was analyzed by using a controlled rotation around the roll centre for the two different stiffness settings, that is the two different placements of the stabilizer stays. The result of this analysis can be seen in Appendix H. Figure 3.4a below shows the meshed model in ANSA and Figure 3.4b shows where the loads are applied to the model.



(a) Meshed model in ANSA



(b) Loads applied to the model, the loads are divided into different sets

Figure 3.4: The set up of the model in ANSA

The calculations were performed in Abaqus and the results from the analysis are interpreted in the post-processor META.

3.7.4 Weight Calculations

A calculation was made to be able to compare the final concept with the existing ARB. The weight for the cast parts is based on data from CES Edupack and the weight of the parts that will be bought externally has been analysed from the CAD model [19]. Some assumptions had to be made to be able to compare the weight of the new product and an existing ARB. The weight for the new parts are summarized together with the existing ARB and by comparing this with the new ARB system it will indicate how much extra weight this variable ARB add to the truck.

4

Results

This chapter will present the results generated during the project. The outline of the chapter will follow that of the methodology chapter. The results and figures presented are necessary for following the progress of the project in the report. Additional results, like concept sketches, material selection plots and concepts screening matrices are presented in the Appendix.

4.1 Requirement Chart

The complete requirement chart can be seen in Appendix B. In the requirement chart there are both requirements and desires. Requirements are marked with an "R" and desires with a "D". A summary of the most important desires and requirements can be seen in table 4.1. These are the ones that the concepts will be screened against in the concept screening phase of the project.

R	Fit inside available packaging area
D	Alter the existing layout as little as possible
D	Be able to scale down for the front roll bar. Scaleable
D	Compatible with several chassis variants
R	Not impact the overall safety of the truck. Failsafe
R	95 % of the ARB's should last longer than the lifetime of a truck
R	Should be able to handle maximum peak loads without plastic deformation or buckling
R	Stiffness equals to 1250 $^{N}/_{mm} \pm 10\%$, in high setting
R	Stiffness equals to $50\% \ \pm 10\%$ of high stiffness, in low setting
D	Be able to change stepless between different stiffness levels
D	Be able to change stiffness at any time, not just at $0°$ roll
R	Withstand the environment under a truck
D	Cost less than 5 $\times$ the existing ARB
D	Weight less than 2 $\times$ the existing ARB

Table 4.1: An excerpt of the most important specifications from the requirement chart

The product has to fit inside the available packaging area and the solution should not alter the existing layout on the truck since the cost and complexity of changing the existing solution is too high. Since there also is an ARB on the front axle it would be beneficial if the same concept could be used there. The solution might not be directly transferable to all chassis since the different chassis have different setups regarding the ARB. If the solution would fit all it will be an advantage. Since safety on the truck is regarded as important for Volvo, the active ARB should not impact the overall safety of the truck. Meaning that if the active function on the product would break there should always exist a roll stiffness from the ARB that prevents the truck from rolling over. The desired change in stiffness is as previously mentioned 50 % of the high stiffness setting. It would be an advantage if the solution could change stiffness stepless and at all times. Since the environment underneath the truck is harsh because of the truck driving in varying climates and on different roads the ARB should be able to withstand it all, meaning that no dirt, water, or snow should be able to impact the solution. The cost of the solution should not exceed five times the cost of the existing ARB, this is an estimate from Volvo based on how much a customer is willing to pay for this solution. The weight should not exceed the double of the existing ARB to not impact fuel consumption too much.

4.2 Concept Generation

The concept generation consists of an iterative process with brainstorming, elimination, and sketching. This chapter includes a description of each sketched concept. In total 10 concepts were sketched and brought to a construction meeting for feedback. The feedback from the engineers made it possible to eliminate two more concepts.

4.2.1 Brainstorming

The brainstorming resulted in numerous concepts with a focus on solving the problem: varying the roll stiffness of the ARB. All of them might not be realisable. But at this stage, the focus is on developing ideas that can be further developed and inspire to non-obvious technical solutions. The brainstorming also gave insight into how complex the problem is. Solutions were invented that had two different degrees of stiffness but not all of these could change between these two levels. Only a few were invented that both were able to set to different stiffness levels and also change between these levels. The concepts that were developed during the brainstorming were named and taken further into an elimination matrix.

4.2.2 Elimination of Generated Concepts

The elimination matrix can be seen in table 4.2. The concept that ended up with a "+" will be developed further and the ones ended up with a "?" will be sketched in more detail to get a better understanding in how the concept works. The reference concept in table 4.2 is the ARB evaluated by A. Carlsson and E. Svensson in their master's thesis [7]. The elimination matrix resulted in that *the Cradle*, *the Composite strut*, *the Crane*, *Double Pipes* and *the W* are removed. The rest of the concept are developed further.

Concept	Solves problem	Stiffness times two	Realizable	Within cost frame	Safe	Comment	Decision
Reference	+	+	-			Not as it is now, develop?	-
The Extra strut	+	+	+	+	+		+
The Leaf spring	+	+	?	+	+	Develop further	+
The Waist	+	+	+	+	+		+
Chebyshev	+	+	+	+	+	In combination with another concept	+
The Cradle	+	-				Not as it is now, develop	-
Bike spring	+	+	+	+	+		+
Rotating disc	+	+	+	+	+	Develop to combining with other concept	+
The Piston	+	+	+	+	+	Combine with Chebyshev	+
Double pipes	+	?	?	+	+		-
Crane	?	?	+	+	+		-
Pencil	+	+	+	+	+		+
I-Beam	+	+	+	+	+		+
Composite strut	?	+	?	-			-
Engine	+	+	?	+	+	Calculate and search internet	?
The W	+	?	+	+	+		-
Double arm	+	+	+	+	+		+
The wire	?	+	?	+	+	Develop	?
The rotating arm	?	+	?	+	+	Fatigue problem? Develop	?

Table 4.2: Elimination matrix

4.2.3 Description of Concepts

Below follows a short description of every concept, sketches of every concept can be seen in Appendix C. The sketched concepts resulted in a deeper understanding of how the concept work, what part that moves to reach the desired change in stiffness.

The Extra Strut

This concept builds on the idea of adding one extra strut on the arm that can be turned on and off. When the extra strut is turned on it is stiff, this means the arm on the ARB is shortened and thereby the stiffness is increased. When *the Extra strut* is turned off, meaning the strut is soft, the ARB depends on the existing stabilizer stay which is attached further out on the arm. This means that the ARB will have a softer setting. A sketch of *the Extra strut* can be seen in Figure C.1.

The Spring Strut
This concept has one stabilizer stay on each side and these stabilizer stays can be turned on and off, when they are turned on the ARB acts like a normal one but when the stabilizer stay is turned off and is soft the load goes through a spring instead that can take up all the load. This will lead to that the ARB will be softer. Figure 4.1 shows a sketch on the idea of the selected concept.

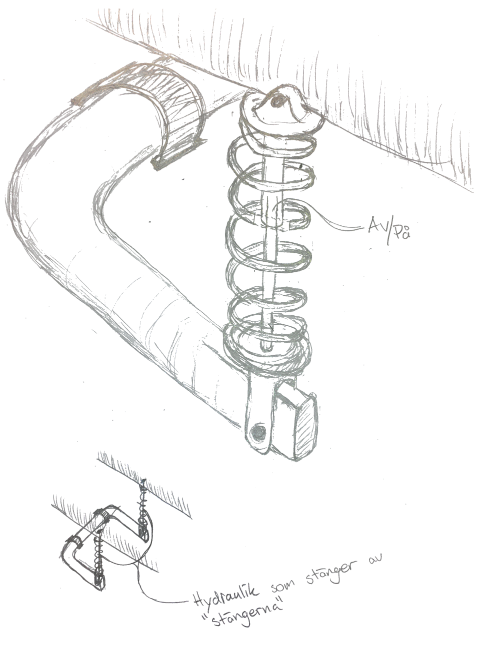


Figure 4.1: A sketch of the Spring Strut

The Spinning Disc
This concept moves the position where the stabilizer stay is attached to the arm. By moving the connection point, the arm will get shorter and this will lead to shorter leverage and therefore less movement in the bar. There is a disc located at the base of the arm, a linkage is attached to this disc and when the disc rotates the linkage moves the point where the stabilizer stay is attached to the arm. Figure C.3 shows this concept.

The Pencil 1 - Chebyshev
The name pencil comes from that the movement mechanism in this concepts has similarities with the clicking mechanism a pencil, that locks itself in the outer and inner position. Similar to *the Spinning Disc* this moves the point where the arm and stabilizer stay is connected. One actuator pushes a cylinder horizontally on the arm. This cylinder is connected to a linkage that has replaced the stabilizer stay. Figure C.4 shows the principle of the linkage, it is inspired by a Chebyshev linkage [20], which enables movement along a straight line.

The Pencil 2
Similar movement mechanism as in *the Pencil 1* but instead of having a linkage as a stabilizer stay this one have just one standard stabilizer stay on each side but the attachment to

the arm is curved so that the ARB does not need to rotate around the bar when changing stiffness. Figure 4.2 shows a sketch on the concept.

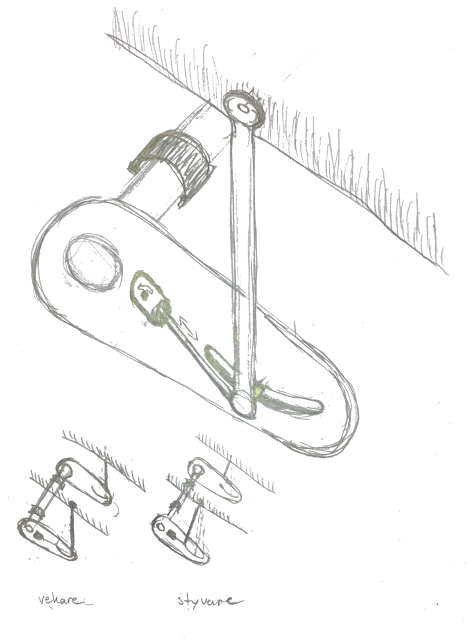


Figure 4.2: A sketch of the Pencil 2

The Extra arm

The Extra arm consists of one regular ARB but alongside the arm, there is another arm attached to the bar. This extra arm has a higher stiffness than the existing arm and can be connected with the existing arm. By connecting it with the existing arm the two arms will work together and create an even stiffer arm. Figure 4.3 shows an idea of the principle with a pin that locks the arms together.

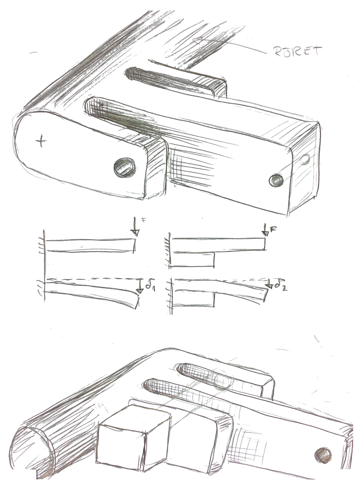


Figure 4.3: A sketch of the Extra arm

The Leaf Spring
The Leaf spring concept build on the idea that combining two leaf springs that are angled toward each other will create a progressive stiffness for the ARB. Figure C.7 is a sketch on how the system will work. When the spring deflects, the endpoints of the leaf springs will travel on two different rotation curves. Initially, a low force will create displacement, but since the leaf springs counteract each other, further displacement will require a substantially higher force.

The I-Beam
The idea is that the arm can change its second moment of inertia. This is achieved with placing two plates with an air "pillow" between. When air is inflated to the pillow the distance between the two plates will increase and thereby also the second moment of inertia. How the arm is supposed to take up shear stresses is a problem that has to be investigated further. A sketch on the idea of the concept can be seen in Figure C.8.

The Waist
The Waist has two parallel leaf springs instead of a regular arm. The stabilizer stay is attached to the one at the top. These two springs are connected with a waist that can slide alongside them. By sliding further out on the arm the waist allows the arm to work together to create a stiff arm. When the waist is as far towards the bar as possible only the top leaf spring carry the load and by this the ARB is soft. The Waist can be seen in Figure C.9.

The Wire
The Wire concept has the idea that steel wires are good at handling tensile stress. The concept was inspired by the way wires are used in sporting equipment, such as cycling shoes or skiing boots, to stiffen the structure when tensioning the wires. By using a more ductile base structure and adding wires that can be tensioned and stiffen the structure the arms of the ARB can be set to be stiff or soft depending on the tension in the wires. It has to be one wire on the top and one wire on the bottom of the arm to handle stresses in both directions since wires only can take stress in tension. Figure C.10 illustrates the idea.

4.2.4 Construction Meeting, Concept Feedback

The session with the engineers resulted in new insight into the robustness needed in the concepts. The concept called *The Extra strut*, see C.1, was considered interesting for development but insight about the behaviour from parallel stabilizer stays in deflection was gained from the group. The stabilizer stays will deform by different amounts because of the way they are mounted on the arm. The structure will self-lock, act as a truss and become very stiff. This was noted for further development of *The Extra strut*. The meeting also resulted in one new concept that moves the points where the stabilizer stays are attached to the frame. These points move in a lateral direction towards the centre of the truck to reduce the vertical deflection going into the ARB. This concept was interesting to investigate in modeFRONTIER and will be further described in the following section 4.3.1. This idea set the foundation for the concept that eventually would become *Y-Change*.

The construction meeting also generated further development of the concept *Extra arm*, the concept was idea was to use brake caliper and brake pads on the arms to easier attach and detach the added stiffness from the second arm. Another idea was to combine *the Extra arm* concept with additional torsion bars, the diameter of the torsion bar affect the stiffness, and using a drum brake to attach or detach the tubes from each other. These concepts are described in section 4.3.2 and more sketches can be found in Appendix D.

The I-Beam as it was described in the previous section 4.2.3, was regarded as a not realizable concept because it consisted of many different parts and would have problems being subjected to shear forces. The concept mentioned in the Theory 2 with a rotating arm also use the moment of inertia to change the stiffness by rotating an I-beam. Therefore *the Rotating Arm* is merged with *the I-Beam* concept and replace the previous *I-Beam*.

The group of engineers raised a concern with some of the concepts. A desire from the engineers was to keep any alterations of the ARB stiffness away from the area with the highest torque, near the torsion bar of the ARB. The area where the arms connect to the bar is the area with the highest stress levels and therefore this spot is a weak link and it will be tough to keep the solution intact.

4.2.5 Concept Elimination

Two of the concepts were eliminated before further development. The elimination was based in the on the feedback from the construction meeting. *The Wire* concept was perceived as too complicated and consisting of too many moving parts to make it robust. To make *the Wire* concept robust it was decided that too much time would be needed to develop it.

The Leaf spring concept would be interesting to investigate if it would be possible to activate and deactivate the stiffness in the leaf springs. The progressive anti-roll behaviour of the concept was perceived as unwanted from the perspective of predictability of the vehicle behaviour.

4.3 Concept Development - Iteration 1

The first iteration of the concept development explores the design space with optimization. This creates one new concept, *the Y-Change*. The iteration continues with developing the existing concepts and merge them with the newly created ones. The iteration ends with a screening of the concepts where two concepts are removed and the insight that more information about the concepts is needed to continue the development.

4.3.1 Concept Optimization with modeFRONTIER

Figure 4.4 shows all the feasible designs from the optimization in one bubble-chart. The x-axis in the graph indicates how large the roll angle is with the given change and the y-axis indicates how much the variables have changed from the original design. The colour on each bubble represents the stresses in the designs. In the figure the Pareto front with the optimum solutions is clear, meaning that the rotation and amount of change are contradicting each other. There is no clear best solution in the diagram, this is because all the solutions are along with one even curve without a knee. This means another chart, more constraint, or more data is needed to choose the best design.

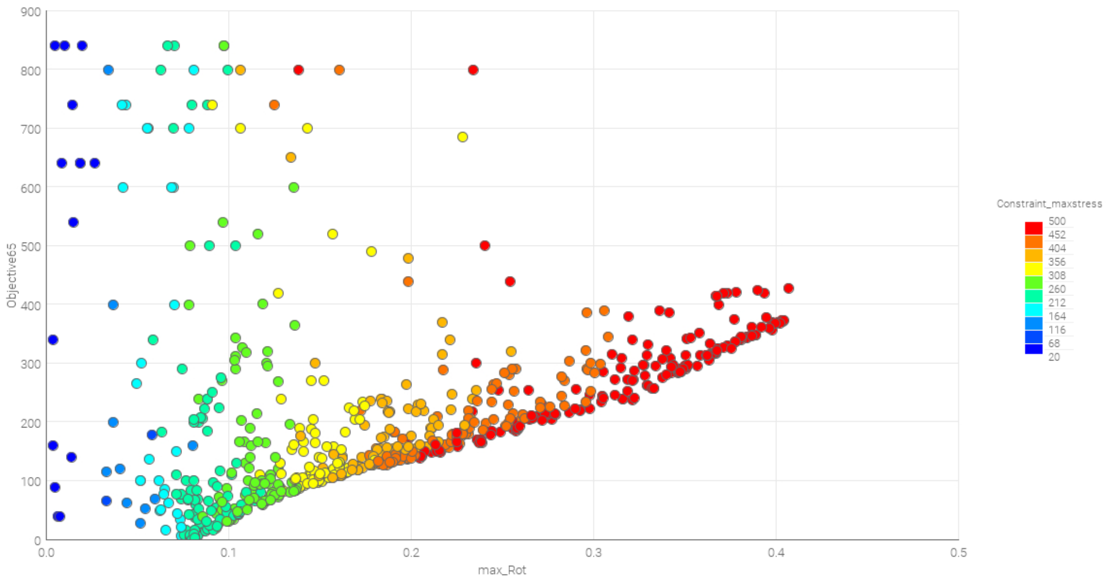


Figure 4.4: Bubble diagram with all feasible solutions, the x-axis is the roll angle ϕ, and the y-axis is how much the concept have changed, the colour indicates the stresses in the solution

Figure 4.5 shows all the solutions on the Pareto front with all the input parameters, objectives, and constraints. "Objective65" in this graph is the objective to minimize the change as much as possible. The colour on each line represents how much the ARB rotates around the roll centre. As a reference, the base model, that is an unchanged model, have a rotation of 0.078 radians. This shows that to get the least stiff design with the minimum change on all parameters one should have a design that moves the upper mounting position, where the stabilizer stays connects with the frame of the truck. The direction of movement is towards the centre of the truck, that is the Y-direction. The analysis also shows that the arm length is important for the stiffness of the ARB.

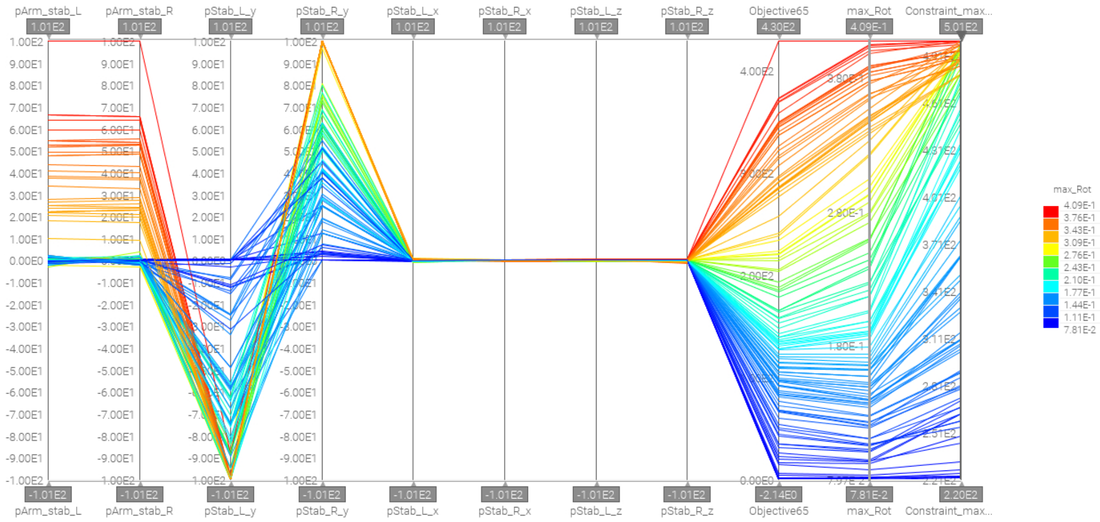


Figure 4.5: The Pareto designs in a parallel coordinates-chart, each line shows the values of one solution on the Pareto front

Based on this result and previously gathered information from the construction meeting it was decided to generate a concept where the stabilizer stays upper connection moves in the Y-direction toward the centre of the truck.

4.3.2 Concept Development, Generation 2

Table 4.3 shows a summary of the remaining and new concepts after the elimination and optimization. Further below each concepts are described in detail.

Concept	Status
The Spring strut	Developed
The Extra arm	Developed
The Extra strut	Developed
The Waist	Unchanged
The Rotating arm	New
The Y-Change	New
The Pencil	Developed from Pencil 1 and 2 and Rotating disc
The Engine	New
The Planetary gear	New

Table 4.3: The remaining concepts after the first screening and development

The Spring strut
Two vertical stays and several parallel glass fibre leaf springs connected between the stays, instead of one coil-spring and one strut, this is illustrated in Figure 4.6. It is inspired from the bicycle suspension from Lauf Cycling [21]. The leaf spring stays can be locked together with an actuator and by that be stiff and act as a regular solid stabilizer stay. When the lock is released the force goes trough the glass fibre springs and *the Spring strut* is soft and affect the ARB system to become soft.

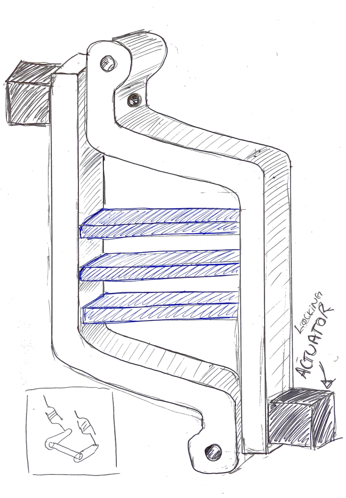


Figure 4.6: New solution of the Spring strut with glass fibre springs in blue

The Extra arm

The Extra arm has several options, one extra arm, or to have an extra torsion bar. Meaning one extra bar over the existing bar that connects to the arms, the larger diameter of the torsion bar increases the stiffness of the ARB. The extra bar can be locked together with the existing bar to increase the stiffness. The idea was to use a drum brake to attach or detach the tubes from each other. There are also several options on how to connect the arms. It can be done by a locking pin or a brake. The concept idea was to use brake calliper and brake pads on the arms to easier attach and detach the added stiffness from the second arm. Figure 4.7 shows a sketch on how the locking mechanism could look like with a pin. More sketches on the concepts using extra bars and brake callipers can be found in Appendix D.

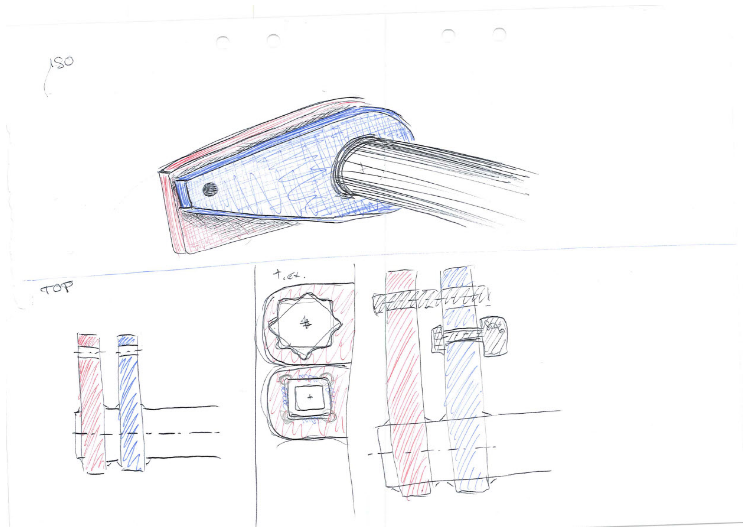


Figure 4.7: The further developed Extra arm concept with a locking pin between the stiff red arm and the softer blue arm. The stays attachment point is in the softer blue arm

The Extra strut
The Extra strut is developed to have two lockable stabilizer stays on each side instead of just one. In this new version, only one strut at each side of the ARB is activated at the same time. Either in the soft or the stiff setting. By adding one more flexible strut, the system is no longer able to become self-locking and thereby become very stiff as was pointed out at the construction meeting.

The Waist
The Waist is unchanged since the previous development, see section 4.2.3 for a description of how the concept works.

The Rotating arm
The Rotating arm have replaced the *I-Beam*. The concept builds on the principle of changing the moment of inertia on the arm. This is done by having an I-beam as the arm and to achieve a softer setting rotate the arm 90° axially. It is similar to the variable ARB's found on rally cars which can be seen in Figure 2.7 in chapter 2. One main uncertainty is how well this concept will handle repeatedly high loads in the soft setting without failing from fatigue.

The Y-Change
The Y-Change has got its name from that the stabilizer stays connection to the frame change in Y-direction, that is they move towards the middle of the truck to lower the stiffness of the design. Figure 4.8 shows a schematic sketch of the concept.

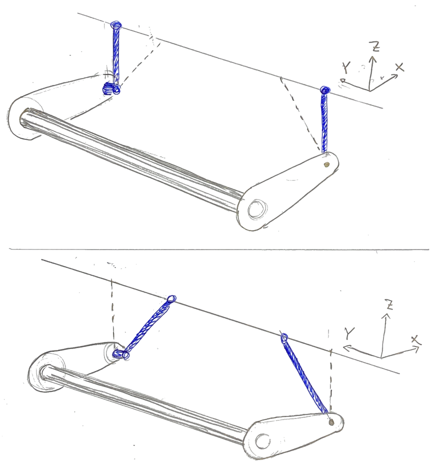


Figure 4.8: A sketch of the Y-Change concept. The thick blue lines indicate the position of the stays and dotted lines indicate the alternative position of the stays

The Pencil
The Pencil is a combination of the previously developed concepts that were similar to each other, that is *the Pencil 1*, *the Pencil 2*, and *the Rotating disc*. All these concepts build on the principle of changing the length of the arm by moving the attachment of the stay to the arm. Therefore a combination of these concepts was developed to ease the development process.

The Engine
Borrows the concept from Schaeffler where they have put an electrical motor at the middle of the bar on the ARB [2]. This would be interesting to investigate since it is the most active system of all the concepts. The main question is how large the motor and gearbox have to be to handle maximum peak loads.

Planetary gear
The idea is similar to *the Engine* but without an electrical motor, that is just to have a planetary gear on each side of the bar that can shift down or up the movement that's inputted into the ARB bar from the arms, a sketch of the idea can be seen in Appendix D, Figure D.6. There is however an uncertainty if and how exactly this will work.

4.3.3 Screening of Concepts

The Pugh matrices 1 and 2 can be seen in Figure E.1 and Figure E.2. Pugh 1 resulted in that *the Engine* and *the Planetary gear* were removed. This mostly due to a high price and problems with packaging space to allow for ground clearance. This means that the improvements an electric motor can give with improved active stiffness do not count for the negatives. The *Extra Arm* concept with torsion bars connected with a drum brake was eliminated because of the curvature of the torsion bar. It was regarded as difficult to assembly and package in the available space. The *Extra Arm* with brake callipers were also considered difficult to package. Pugh iteration 2 resulted in lots of 0's and therefore the conclusion was made that more development is needed to make a decision which concepts to continue development on.

This means that the remaining concepts that after the concept screening are:
- The Spring strut
- The Waist
- The Y-Change
- The Rotating arm
- The Extra arm
- The Extra strut
- The Pencil

4.4 Concept Development - Iteration 2

For the second concept development iteration, an FE-analysis of the concepts was made and this analysis gave a better insight into what the concepts are capable of and how much stiffness change could be anticipated. The result is summarized in a pros and cons list that screens out the least promising concepts.

4.4.1 Structural FE-Analysis of the Concepts

In the section below the result from the beam analysis is presented. Figures with the results from the analysis can be seen in Appendix F.

The Spring strut
The spring needs to have a stiffness of around 1000 $^{N}/_{mm}$ for the ARB to be stiff enough. The total displacement will then be approximately doubled. One question that remains to be answered with the concept is if the spring should have a deflection to accommodate the whole roll angle of the truck, or if it could be allowed to flex 50 *mm* and then act as a solid stabilizer stay if the spring bottoms out and let the ARB take the remaining deflection.

The Waist
The Waist have a problem with achieving the stiffness change according to the beam analysis in ANSA. It is also questioned whether the concept is strong enough because of the lack of material between the two bars. This will have to be investigated further.

The Y-Change
The stabilizer stays need to move around 100 *mm* in Y-direction to achieve the desired change in the ARB. The positive thing with *Y-Change* is that the stress is decreased which means that there will not be any problem for the ARB to manage the loads. The stiffness change also takes advantage of the new roll angle that the smaller distance between the two stabilizer stays create.

The Rotating arm
Regarding displacement, the I-Beam can handle the needed change by having the dimensions of 20 *mm* $\times$ 40 *mm* on the arm. This provided that the bending is taking place the whole arm. The analysis shows that to minimize stresses in the arm, the arms should be rotated in opposite directions.

The Extra arm
With the extra ARB attached close to the stabilizer stay and arm connection, the stiffness can be doubled. The analysis shows that by adding a stiffer arm to the weaker arm there is a lot of added stress at the end of the weak arm. With repeated loads, there could be a problem to make the weak arm manage the fatigue stresses that will occur at the end of the arm.

The Extra strut
The analysis shows that the concept could work with the extra stabilizer stay attached at the same place in the frame as the existing but attached further in on the arm. The maximum stress appears in the arm. However, there still is a lot of torsional stresses in the bar in the low stiffness setting compared to the high stiffness setting. This could depend on that the triangle created between the two stabilizer stays and the arm has stresses between each other and together they work to stiffen up the construction. This triangle might inhibit the functionality of the ARB.

The Pencil
The original stiff ARB has long arms, 380 *mm*, and a large bar diameter of 70 *mm*. If the arms are shorter on this configuration the ARB becomes even stiffer, which not is desired here. Although, an analysis was performed on these configurations and the results can be seen Figure F.13 that represent the high stiffness and in Figure F.14 that represent the

even stiffer setting with less deflection.

The 70 *mm* bar with long arms analysis is done as a reference since this configuration is used on the truck and this configuration is working. The long arm analysis, in Figure F.13, shows that the stress in the highest spot was 1390 *MPa* and around $800-900$ *MPa* in general over the bar. The deformation in the arm is 60 *mm*.

A smaller diameter 60 *mm* bar was needed to get a combination between arms and bar that have the high stiffness setting with the short arms and the possibility to half this stiffness by increasing arm length to long arms.

Short arms and small bar diameter represent the high stiffness setting. The result of the analysis can be seen in Figure F.15 In this setting the stress in the bar reaches 1200 *MPa* in the maximum spots and around 900 *MPa* on average. This is around the same as for the 70 *mm* bar and long-arm configuration in Figure F.13. The deflection is 54 *mm* in the arm.

Long arms and small bar diameter represent the low stiffness settings. The result of the analysis can be seen in Figure F.16. In this setting, the stress in the bar reaches 1500 *MPa* in the maximum spots and around 1100 *MPa* on average over the torsion bar. The deflection is 100 *mm* in the arm and is around the double of the short arm configuration.

Compared to the 70 *mm* bar and long-arm configuration the stresses are in general around 300 *MPa* higher, or around 20 % higher. So this solution needs to be evaluated in a better FEM software with a more accurate model to evaluate the severity of the stress but it is concluded that it has some potential since it achieves to change the stiffness by half and is not completely deemed to fail.

4.4.2 Screening of Concepts, Pros and Cons List

This screening is based on the Pros and Cons List that can be seen in Appendix G. The result from the Pros and Cons List is that *the Waist*, *the Extra Strut* and *the Rotating arm* are removed. The reason to remove *the Waist* is that according to the analysis it is hard to achieve the desired stiffness change. *The Extra strut* is removed because of uncertainties on the functionality and also because it is another way of changing the length of the arm. Meaning that the solution can be included in *the Pencil*. *The Rotating arm* is removed because of the uncertainty in fatigue and problems to fit within available packaging space. The packaging space problem for the *the Rotating arm* depends on that all the moving parts are located at the place where the stresses are high, and the available packaging space in this area of the truck is also very limited. It is anticipated that the mechanism that allows for the arms to rotate will clash with the reaction rods on the sides.

This means that the remaining concepts after the Pros and Cons List are:

- The Pencil
- The Spring Strut
- The Y-Change
- The Extra Arm

4.5 Concept Development - Iteration 3

The third iteration with concept development. CAD models of the concepts are created and presented at the construction meeting. The feedback from the construction meeting results in an elimination of rejected concepts.

4.5.1 Detailed CAD Design of Selected Concepts

Figure 4.9, 4.10, 4.11 and 4.12 shows the constructed CAD models of the concepts. *The Pencil* concept have its actuators that moves the stabilizer stays connection to the arm. The idea is that the stabilizer stay will be able to slide inside the track in the arm.

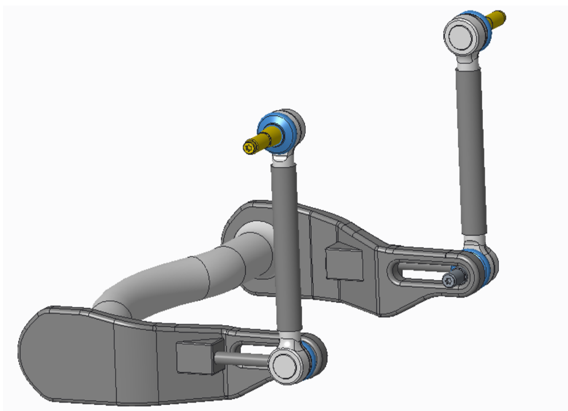


Figure 4.9: The Pencil concept, the stabilizer stay can be moved on the arm to change the stiffness.

The Spring strut was brainstormed one more time since it was not clear on how the final *Spring strut* should look like. The spring function was moved around in the area around the stabilizer stay and the reaction rod bracket. Inspiration was taken from Lauf bicycles glass fibre leaf spring bicycle fork suspension [21]. Another concept was inspired by a so-called springer fork from old motorcycles. The last idea was to use a torsional spring, that is created by using a lever arm and a torsional tube and attach it in the reaction rod bracket. The concept with the torsional tube could also be used with a rubber bushing instead of a steel torsional tube, this a suspension configuration used on some car trailers. This was decided as the concept with the most potential and can be seen in 4.15.

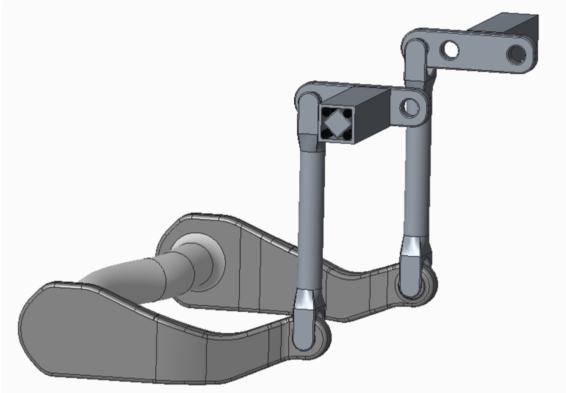


Figure 4.10: The Spring strut concept, a torsional rubber spring that can be locked by a locking pin in one of the holes.

The Y-Change concept have a ball screw that can translate the attachment of the stabilizer stay in Y-direction. The idea is that an electric motor will turn both ball screws at the same time. Figure 4.11 shows the concept mounted on the cross beam of a truck.

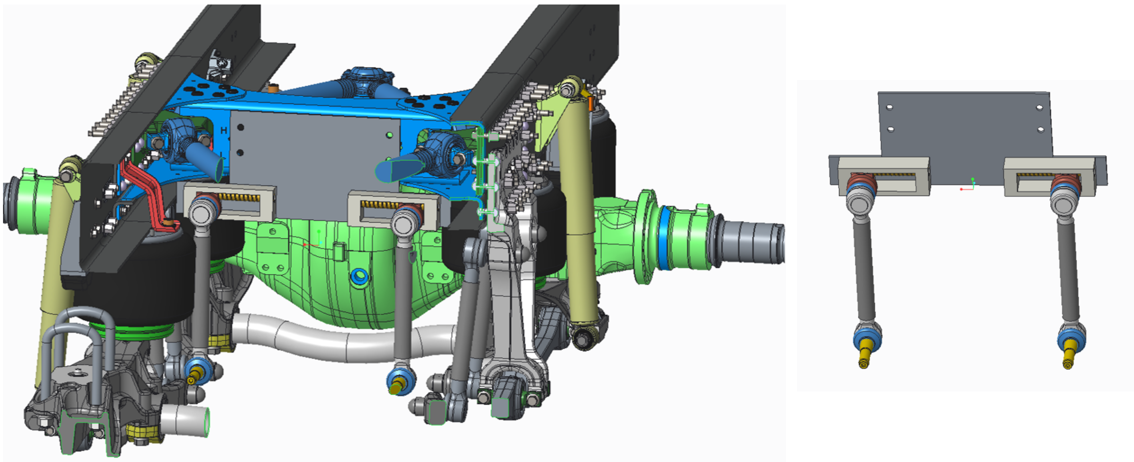


Figure 4.11: The Y-Change concept, a ball screw moves the struts towards the centre of the truck to decrease the roll stiffness.

The Extra arm concept, seen in Figure 4.12, has an insert that is riveted in a slot on the arm close to the bar. The insert can be locked together with the arm by a locking pin to increase the stiffness of the arm. Because of packaging space reasons, it was not possible to fit a second arm on the torsion bar and still have sufficient clearance to the reaction rods this is why the extra arm had to be inserted in a slot.

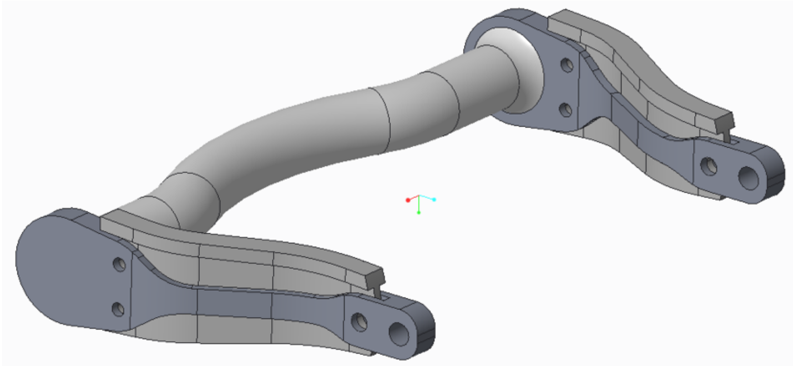


Figure 4.12: The Extra arm concept, a locking pin can connect the two arms and thereby stiffen up the arm

4.5.2 Construction Meeting 2, Concept Feedback

The feedback from the meeting was that the engineering team liked the concept with *the Y-Change* and how it changed the stiffness, the team also appreciated that the solution was not located close to the ground of the truck. The feedback was that *the Y-Change* concept that was presented, see Figure 4.11, had the upper mounting points too far inside of the truck. The upper mounting points for the stabilizer stays should be further out against the sides, toward the reaction rod bracket where they are located originally and can also be seen in Figure 4.11. There was also concern that the middle bracket where the ball screws are mounted was going to interfere with the prop shaft between the first and second axle on 6x4 trucks.

The group were concerned that it would be too costly to solve the tolerances that *The Extra arm* need to have when manufacturing a slot in the arm and to fit an extra piece of metal there that also require tight manufacturing tolerances, see Figure 4.12. Also, making the extra stiffening part of steel in the middle to be activated and deactivated with a pin was regarded as undesired. The general feedback was that there is an imminent risk that water, dirt and salt from the road can build up in the slot and cause corrosion. Concerns about the fatigue strength of the arms were raised for when the middle piece of steel is de-activated in the soft setting.

Remaining Concepts after Construction Meeting 2

The feedback from the construction meeting and previously known about the concepts resulted in that *the Pencil* and *The Extra arm* is removed. This means that the remaining two concepts are:

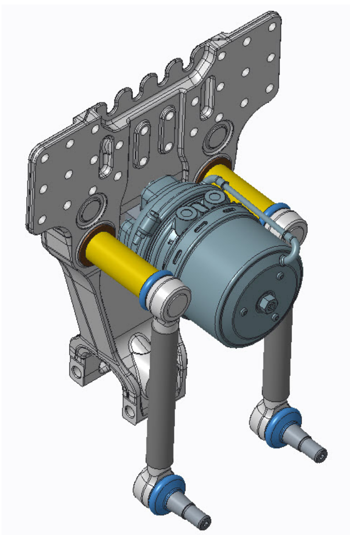
- The Y-Change
- The Spring strut

4.6 Concept Development - Iteration 4

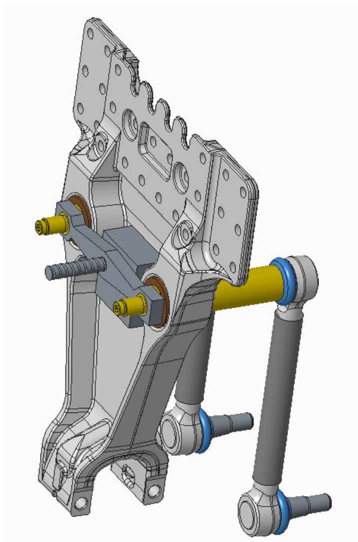
This chapter shows CAD models of the two reaming concepts, *the Y-Change* and *the Spring strut*. This iteration also describes the conclusions that were drawn from contacts with external experts. The iteration ends with a description of the result from the final concept selection.

4.6.1 Design of Y-Change and Spring strut

During the brainstorming session on how to develop *the Y-Change* concept, an idea that was generated is to mount the axles directly in the reaction rod bracket. The reaction rod bracket could support a sliding mechanism and at the same time support, the stabilizer stays. The idea is that the axles are mounted on linear bearings so that the axle itself can slide in and out of the reaction rod bracket to change the y position of the stabilizer stays. Both solutions can be seen in Figure 4.13. The brake cylinder in Figure 4.13a was an idea of how *the Y-Change* could be activated with parts available in-house to start testing. It was anticipated that the brake cylinder could provide the necessary force required to move the mechanism. However, the cylinder could not fit in the limited packaging space when the mechanism was fitted in the truck assembly. As another solution to change y position, a mock-up ball screw and electrical motor were fitted as seen in Figure 4.13b. The solution was FEM simulated in Creo Simulate and it was found that with a vertical load of 70 kN the axle deflected with 2 mm and had stresses of 700 MPa in the most affected areas. The deflection means there is an impending risk for the yellow axle to seize in the brown bearing bushing in Figure 4.13b.



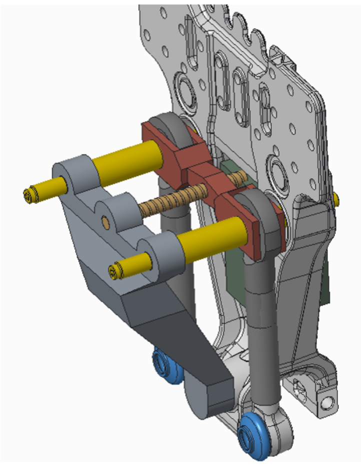
(a) The solution with a brake cylinder as linear drive.



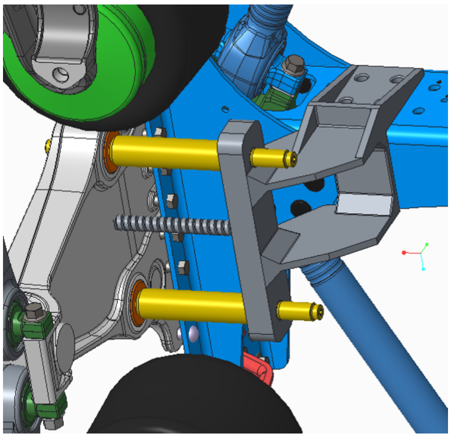
(b) The solution with a electric motor and ball screw as linear drive.

Figure 4.13: Both versions of the Y-Change solution without a support but from different angles. The axles slide in and out linear bushings in and out of the reaction rod bracket.

To decrease the stress and deflection extra support was needed, two concepts were created to fix this problem. Figure 4.14 shows the two alternatives of *the Y-Change* concept. The alternative in Figure 4.14a have attached the bracket to the reaction rod bracket and whilst the alternative in Figure 4.14a have attached the bracket to the cross beam instead.



(a) Y-Change alt. 1, the bracket is attached to the bottom of the reaction rod bracket.



(b) Y-Change alt. 2, the bracket is attached to the cross beam.

Figure 4.14: Both versions of the Y-Change with a bracket concept. The axles are fixed and the stays slide on the axles to the new position.

Figure 4.15 shows two alternatives of *the Spring strut* concept. Both versions have rubber inserts that will work as a spring. The left stabilizer stay has a lever with a rubber insert, the idea is that it will be fixed by an axle in the rear hole and with a locking pin, it will be able to lock the spring in the other hole. The right stabilizer stay can flex in the stay.

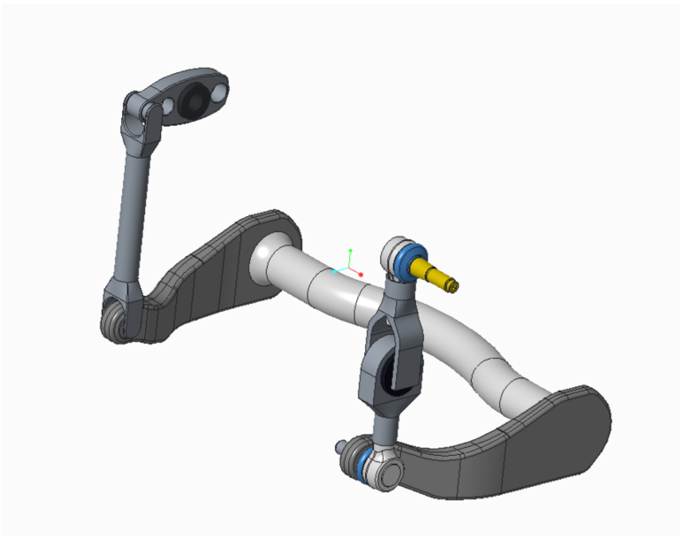


Figure 4.15: The Spring strut concept with two alternatives, one stay has a rubber spring on it and one have a rubber spring connected with a lever

4.6.2 Preliminary Material Selection

Material Selection for the Y-Change Concept

Volvo wants to reduce the number of components in the assembly, this is in accordance with DFA and to manufacture the part to as low cost as possible. For complex-shaped products, casting is a commonly used production method at Volvo. Since *the Y-Change* is large but could be made slimmer with reinforcements, casting is consequently regarded as the best production method. Alternatives such as machining and welding components together are less cost-effective because of the added production steps. Welds are also, as mentioned in the theory chapter 2, problematic in dynamically loaded structures. Since the microstructure is affected, they are susceptible to cracks and failure [10]. Welds are hence wanted to be avoided. Metal casting is, therefore, a cost-efficient alternative at the expected production volumes, according to Volvo.

The material often used in cast components similar to the concept *Y-Change* is a ductile cast iron (DI) material, GJS-EN-500-7 this is therefore selected as a preliminary material for *Y-change* and is used as a reference in the future thorough material selection for the final concept in section 3.7.1.

Material Selection for the Spring Strut Concept

After contacts with experts in rubber suspension at Trelleborg, *the Spring strut* was written of. The main takeaway from the consultation was that the available space is too small to design a rubber bushing that can handle all the loads and at the same time have the required stiffness and the given deflection.

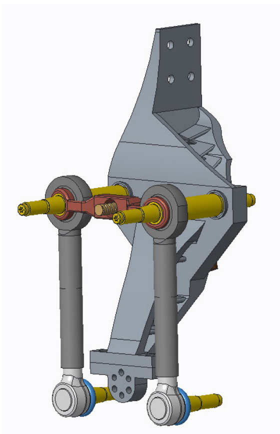
Although, rubber is a material that can withstand long elongation before fracture and much deformation in compression. The conclusion from Trelleborg was that the part could be produced in rubber but that a larger section area was needed in order to allow for the deflection and to have the specified stiffness of 1000 kN/mm. Since the elastic modulus of rubber is around 100,000 times lower compared to the elastic modulus of steel [18]. *The Spring strut* has to be compensated with more section area to have the required stiffness. Consequently, the available packaging space limited *The Spring strut* to be constructed with the required stiffness and it was eliminated from further development.

4.6.3 Final Concept Selection

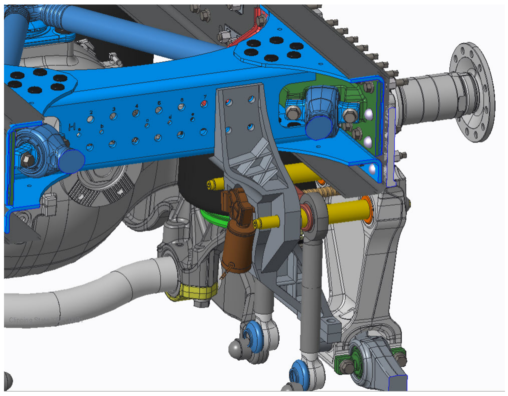
The material investigation in previous section 4.6.2 resulted in the elimination of *the Spring strut*. This means that the remaining concepts are three different variants of *Y-Change*. *Y-Change* with a new bracket in the cross beam, *Y-Change* with a new bracket in the reaction rod bracket and *Y-Change* without any bracket. After consultation with Volvo, it was decided to create a combination of *Y-Change* in the cross beam and *Y-Change* in the reaction rod bracket. This is motivated by connecting the new bracket into two places the loads on the attachments are spread out and this leads to lower forces on the cross beam. The plain bearings in the solution without a bracket have to be able to handle large torques and forces and also enable the axle to slide. None such application today can be found and this is the reason to eliminate this solution.

The Combination

The combination of the two *Y-Change* concepts can be seen in Figure 4.16. The layout of the new concept is called *the Y-Change Combination* and is a result of topology optimization. The result from the topology optimization shows where to place the material to have the best combination of a lightweight and stiff construction. With this information, a CAD model of *the Y-Change Combination* bracket could be designed in Creo Parametrics. This model of the concept, see Figure 4.16a, is designed to be manufactured in cast iron since that is the preliminary material that is selected for the product. To make the part adapted for casting, its designed not to have any hidden surfaces and the parting line of the casting mould is located in the middle, between the axles from the top to the bottom.



(a) The Y-Change combination bracket in a sub-assembly with the stabilizer stays.



(b) The Y-Change combination bracket mounted on the truck between the cross beam and bottom of the reaction rod bracket

Figure 4.16: The Y-Change combination bracket in sub-assembly and assembled on the truck.

Change in Condition

After a meeting with Volvo, that took place after the Covid-19 lay-off was resolved, new information about the load case came up that meant that the bracket had to be redesigned. The theory was that the concept in Figure 4.16b would stiffen the whole frame structure in an undesired way, by connecting the reaction rod bracket and the cross beam. If the bracket would be mounted like in Figure 4.16b, there is a risk that when the truck rolls, all forces will travel through the new bracket to the cross beam. This is not desired from Volvo since the cross beam is not designed to handle such loads. Due to time restrictions, no new topology optimization could be implemented, instead, the focus was located to design and analyze stresses in the new bracket.

A major change in the conditions for the concept development limitations also changed. Earlier it was desired from Volvo not to alter existing components or alter as little as possible, therefore only 3 bolt holes and 3 pinholes were made in the bottom of the reaction rod bracket to mount *the Y-Change Combination* bracket, see 4.16a. This was the only place with enough material thickness to support the structure. But because of that, the 6 holes were made in the reaction rod bracket the limitations set by Volvo did not apply any more. It was said from Volvo that because of the bolting holes, even more alterations of the reaction rod bracket were allowed. This changed the conditions for the project in such way that the new bracket was completely re-designed in combination with re-design of the reaction rod bracket to allow better joining and assembly. This new concept is described in the following chapter.

4.7 Design of the Final Concept, the Y-Change

Figure 4.17 shows the final solution on the rear axis of a truck. This solution is mounted on a 6x4 truck with two driven rear axles.

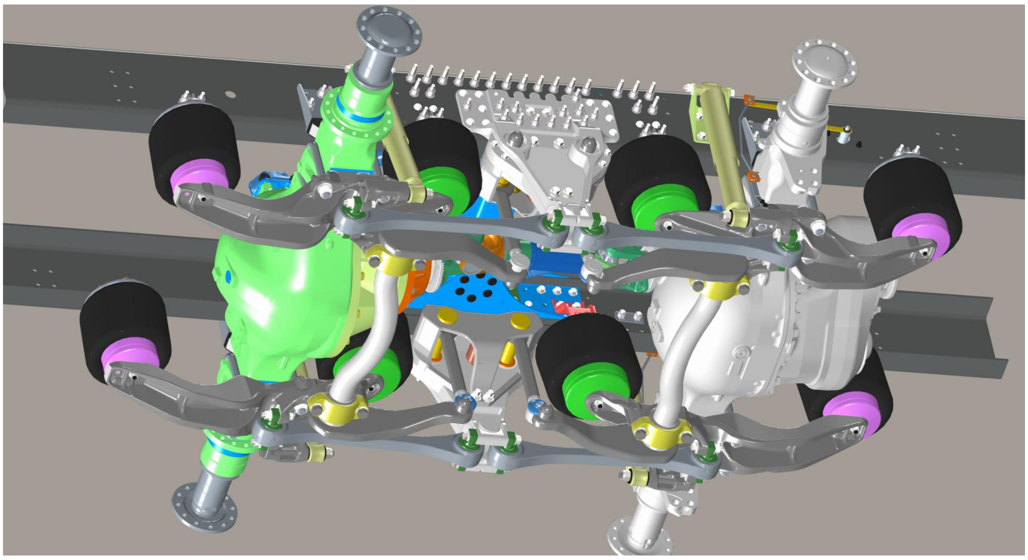


Figure 4.17: Final solution mounted on a truck

A close-up on the solution is displayed in Figure 4.18. The figure shows the two ARB's on a 6x4, one for each rear axle. On the reaction rod bracket the *Y-Change bracket* is connected and inside this bracket, the stabilizer stays are attached. Inside the bracket, there are two axles, one motor and one lead screw. When the motor turns the lead screw the stabilizer stays are pushed on the axles with a *Moving fork*. When the stabilizer stays are moved towards the centre of the truck the total roll stiffness from the ARB is lowered.

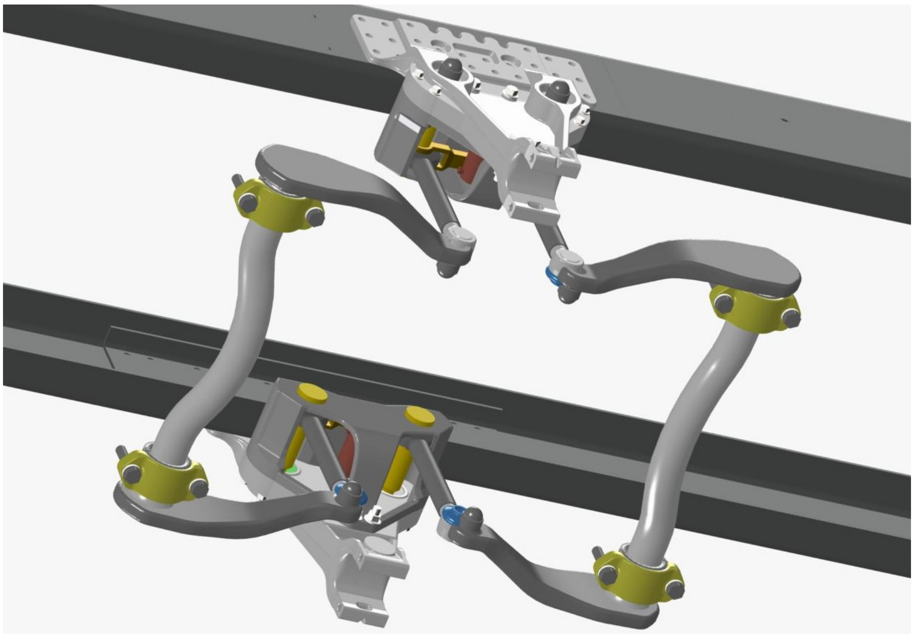


Figure 4.18: Close-up on the designed solution

4.7.1 Final Material Selection

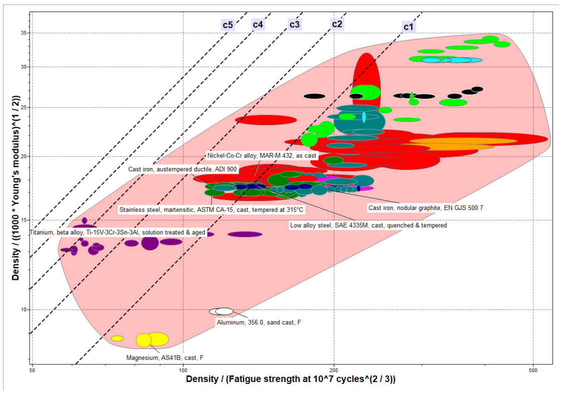
All figures presented in this section can also be seen in full size in Appendix A. At the end of this section, a table with material data for the selected material and the reference material is presented.
The material selection process in CES resulted in that the best material for the application is an austempered ductile cast iron (ADI) that is called ADI 900 (GJS-EN-900-8). This is cast iron with better mechanical properties compared to conventional nodular ductile iron (DI) GJS-500-7 (GJS-EN-500-7) [19, 22]. In recent years ADI has been used in heavy truck components and advertised by foundries that it is possible to make automotive suspension components, such as control arms, cast with ADI iron [23, 24].

ADI is mentioned from several sources as a cost efficient material replacement for components made in cast steel and even aluminium. This is because of its low cost and weight in relation to its high yield strength, fatigue strength and fracture toughness[22, 25].

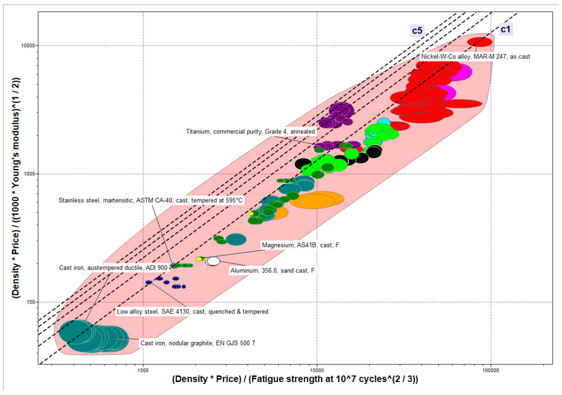
In the CES selection process there were several candidates after screening but when the cost was considered the cast iron ADI 900 was better than the other material candidates in CES. The ADI 900 ranks better than the reference material that Volvo applies in similar components, a ductile iron GJS-500-7. This can be seen in the Figures 4.19b and 4.20b.

ADI 900 is a nodular cast iron that consists of an ausferrite matrix, which consists of acicular ferrite and carbon saturated austenite. Dispersed in the matrix there are spherical graphite [22, 25]. The materials special microstructure and performance comes from that the material is austempered, quenched from around $900°C$ to a temperature of around $230°C - 400°C$ and held there for around $2 - 5$ h to let the material obtain the ausferrite structure. What decides if the matrix is coarse or fine is a complex relationship between the chemistry of the iron and the exact time and temperature for the heat treatment [22, 23]. The result is a material with a doubled increase in strength and toughness compared to the conventional DI like the GJS-500-7. Conventional DI also has spherical graphite, similar to ADI, but instead, the DI matrix microstructure consists of ferrite and pearlite that is created when the material cools to room temperature without any austempering treatment [26, 27]. Compared to the austempering in ADI the DI is often used in as-cast condition but can also be heat treated to tweak its performance and to e.g. relieve residual stresses [26] The nodular graphite gives the cast iron materials some ductility and fatigue resistance compared to grey iron with flaky graphite [26].

The plot with coupling lines shows what the dimensioning constraint is for the materials. In these plots the y-axis represent the stiffness limiting constraint and the x-axis is the fatigue limiting constraint, both when the objective is minimizing mass see 4.19a and minimizing cost 4.19b. The materials that lie above the coupling line have the dimensioning constraint on the y-axis and those below have the dimensioning constraint on the x-axis. As can be seen in Figure 4.19a, most materials are affected by both constraints when looking at the line for the coupling constant C_1 which is calculated for a maximum a deflection of $\delta = 1$ mm. The coupling constant were also calculated for deflections between $\delta = 1$ mm to $\delta = 5$ mm $C_1 - C_5$, numbers $1 - 5$ indicate the deflection. When the deflection constraint is relaxed and set to allow for a deflection of $\delta = 5$ mm, the coupling line C_5 moves upwards in the plot and all materials in the plot fall below the line in Figure 4.19a and 4.19b, thus the dimensioning constraint is fatigue when more deflection is tolerated. For this selection of materials, the coupling line C_1 was used and thus both constraints are dimensioning. Since it is desired to minimize both material indices, the best materials for minimizing mass can be found in the lower left corner in Figure 4.19a, that is magnesium. In lower left in Figure 4.19b to minimize cost, cast iron is the best selection. To find the best material for these contradicting results a trade-off plot is created.



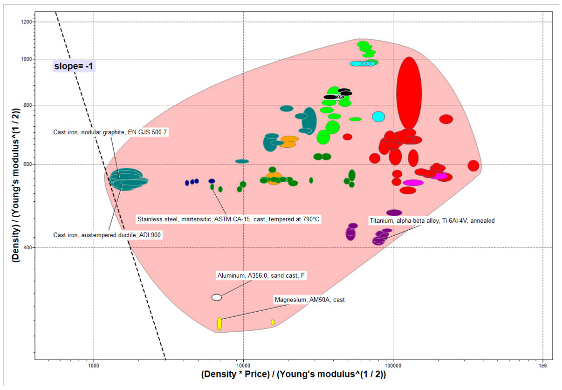
(a) Material indices for M_2 vs. M_1. Coupling lines $C_1 - C_5$ to identify the dimensioning material index to minimize mass.



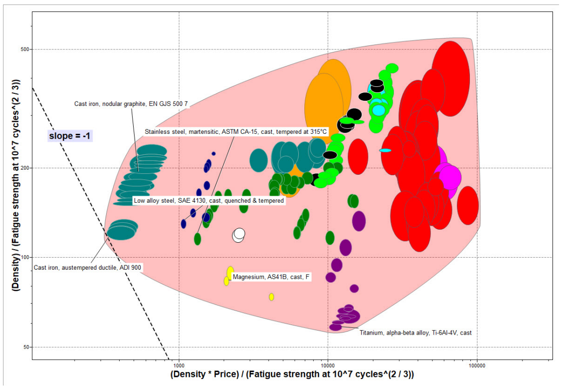
(b) Material indices for M_{2C} vs. M_{1C}. Coupling lines $C_1 - C_5$ to identify the dimensioning material index to minimize cost.

Figure 4.19: Coupling line plots to minimize mass and cost.

The trade-off plots on both stiffness vs. cost 4.20a and fatigue performance vs. cost 4.20b show that the best material is ADI types of cast iron for the given circumstances, this is represented with a line with a slope of -1, this means that both material indices are weighed as equally important.



(a) Trade-off plot on M_2 vs. M_{2C}. Line with slope -1 represent that both indices are weighed equally.



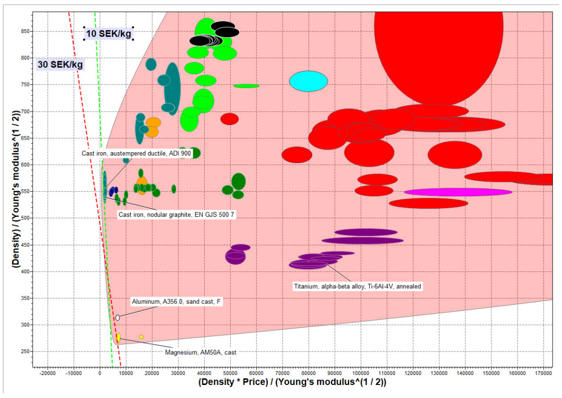
(b) Trade-off plot on M_1 vs.M_{1C}. Line with slope -1 represent that both indices are weighed equally.

Figure 4.20: Trade-off plots for stiffness and fatigue.

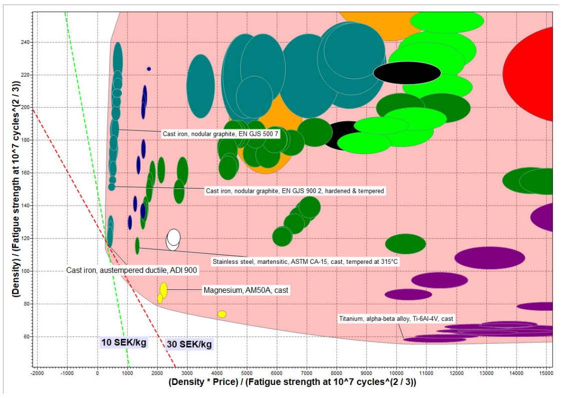
The exchange constants lines are plotted in stiffness performance vs. cost, see Figure

4.21a. In Figure 4.21a it can be seen that for an exchange constant of 10 $^{SEK}/_{kg}$ the best material selection would be a variety of cast irons, ADI type cast iron ranking equally as DI GJS-500-7. For the exchange constant of 30 $^{SEK}/_{kg}$ it can be seen that the CES plot in Figure 4.21a suggest using aluminium or magnesium before ADI iron. This is because of the increased exchange constant but also that the section area is allowed to be free when calculating the material indices. It could be a lighter solution but to keep the deflection low, an addition in section area would be needed and with the limited space in this product application, it is not a possible solution.

The exchange constants lines are also plotted in fatigue performance vs. cost, see Figure 4.21b. In this case when both the 10 $^{SEK}/_{kg}$ and 30 $^{SEK}/_{kg}$ lines are just touching the Pareto front, the ADI irons are the first selection, with ADI 900 as non-dominated candidate. The reference material GJS-500-7, that Volvo often use, is lying further away in the plot and therefore are a dominated solution, see Figure 4.21b. The Pareto front in this plot consist of ADI, Magnesium and Titanium but the Magnesium and Titanium are rated as too expensive for this application with the exchange constant between $10-30$ $^{SEK}/_{kg}$



(a) Trade-off plot with exchange constants 30 $^{SEK}/_{kg}$ and 10 $^{SEK}/_{kg}$ on Stiffness performance to minimize mass (M_2) vs. Stiffness performance to minimize cost (M_{2C}).



(b) Trade-off plot with exchange constants 30 $^{SEK}/_{kg}$ and 10 $^{SEK}/_{kg}$ on Fatigue performance to minimize mass (M_1) vs. Fatigue performance to minimize cost (M_{1C}).

Figure 4.21: Trade-off plots with exchange constant lines for stiffness and fatigue.

ADI is a valid material selection that is possible to shape by casting to these kinds of vehicle components, this is mentioned both in articles and in the product portfolios from foundry's [23, 24]. It is mentioned that ADI has around a 50 % increase in fatigue strength compared with conventional DI, this can also be further increased with surface treatments [22]. The ausferrite microstructure can heave the crack propagation and stress-induced transformation from austenite to martensite can act as a crack closing mechanism [22].

The material cost in CES is however not taking processing and post processing cost in consideration, for example the austempering for the ADI or if the DI need some kind of additional heat treatment. DI is often used in as-casted condition since it anneals in the mould by itself [26]. However heat-treatment of DI can be necessary to get an uniform matrix structure or to tweak the properties [26]. Internal sources at Volvo say that the cost for a finished part in ADI is around 25 % more expensive than a finished part in DI [28]. The added production time for ADI is adding production cost because the austempering heat-treatment process, an additional $1-5$ h of production cost is required compared to DI. This cost is not something that the software CES take in consideration. This is a problem since although there is an increase in mechanical properties for ADI 900 it might not be worth as a substitute for DI if not enough savings can be done by reducing mass of the part. If the reference material would have been cast steel or aluminium and the part could have been made with ADI instead, then there is potential savings to be made according to Volvo [28].

When it comes to environmental aspects, these two materials DI and ADI, utilize around the same amount of energy, CO_2 and water per kilogram of material in casting [19]. They can be recycled in the same way, they are both nodular cast irons and can be recycled to new products [29]. The current amount of recycled content in the supply of cast iron is around 70 % [19].

	DI, GJS-500-7	**ADI 900**	**Units**
Mechanical properties			
Young's modulus	165 - 178	163 - 170	GPa
Yield strength (elastic limit)	320 - 370	550 - 650	MPa
Tensile strength	500 - 600	900 - 995	MPa
Hardness - Vickers	175 - 225	291 - 325	HV
Elongation	7 - 15	8 - 11	%
Fatigue strength at 10^7 cycles	224 - 248	415 - 485	MPa
Fracture toughness, K_{1c}	22 - 54	99 - 121	$MPa \cdot m^{0.5}$
Physical properties			
Density	7050 - 7220	7040 - 7140	kg/m^3
Price			
Price per kilogram	2,5 - 3,8	2,7 - 4,2	SEK/kg

Table 4.4: Material data for DI GJS-500-7 and ADI 900 from CES, used in plots for ranking the materials [19].

4.7.2 Design of the Final Concept, the Y-Change Bracket

The final design of the *Y-Change bracket* can be seen in Figure 4.22. The *Y-Change bracket* is designed to be produced by casting. The design also enables the part to be easily implemented in the assembly process. This is made possible by pre-assemble on the side of the line together with the reaction rod bracket and then move the complete part to the assembly line. The bolts are placed to distribute the force around the surface where the bracket meets the reaction rod bracket. This enables a seal to be mounted in between the two parts. All the screw holes on the *Y-Change bracket* except the two on the bottom are threaded to ease the assembly process.

Figure 4.22: The new bracket, the design encapsulates the parts inside

With the new designed *Y-Change bracket* it means that the reaction rod bracket is re-designed to meet the interface of the *Y-Change bracket*. The surface that is in contact with the *Y-Change bracket* is flattened and material is added on the reaction rod bracket to enable bolts to be fastened.

4.7.3 Parts in the ARB

The components that the active ARB consist of are listed in Figure 4.23. Nuts and bolts are not listed as a component but exist in the figure as reference. The exact dimension of the motor, lead screw and gear wheels have to be tested and further evaluated before the final selection is made. Figure 4.24 shows an exploded view of the ARB system with all its parts.

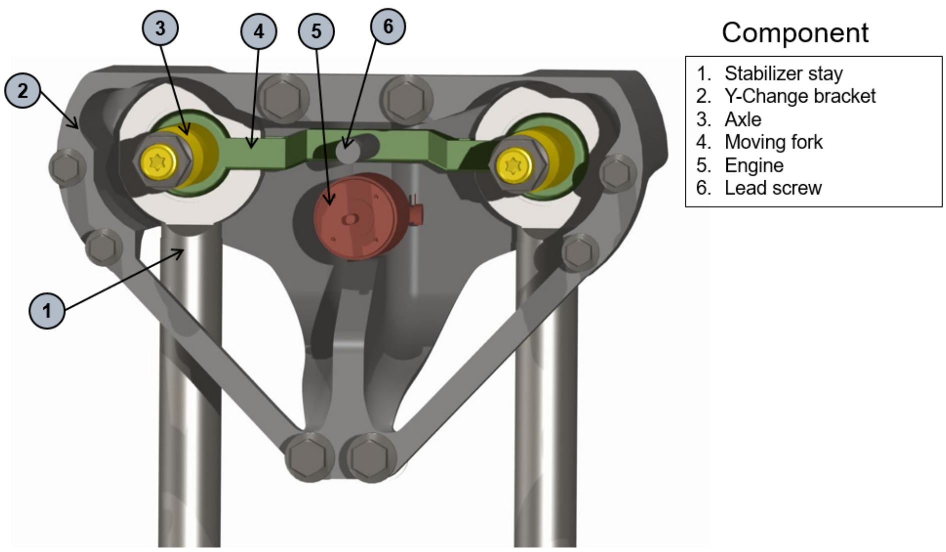


Figure 4.23: The parts included in the solution, the reaction rod bracket is hidden in this figure

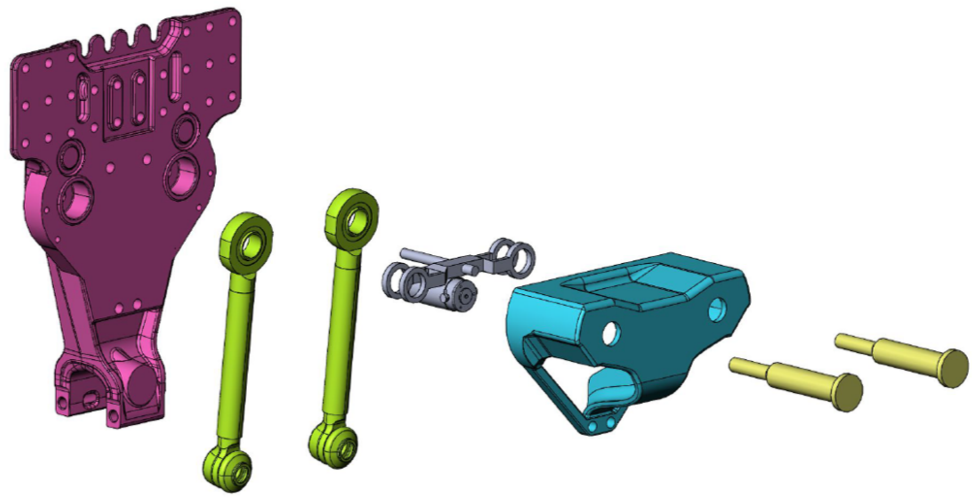


Figure 4.24: An exploded view of the solution

The idea is that the motor and lead screw will be connected with a set of gearwheels. The size and number of sprockets on the gearwheels can be decided when it has been investigated exactly how much force that is needed to move the stabilizer stays. An encapsulation to close the open space where the stays go down to the ARB is also to be included before introduced in production, to seal out water and dirt.

The Moving fork

The Moving fork is designed to allow the stay to move freely but at the same time hold it in place in the y-direction. It acts as a fork around the rod ends to move and lock them along the y-axis. The design allows the rod ends to rotate freely as the truck rolls and the ARB moves.

4.7.4 Structural FE-Analysis Results

In Figure 4.25 the results from the analysis can be seen, the figure shows a combination of the max stresses in every node independent of the load case. Red indicates that the stresses are above the yield point. The highest stress concentration is located on the old existing reaction rod bracket. This means that the weakest spot in the design is on existing parts from the truck. The stresses in the *Y-Change bracket* and the axles are below the yield point meaning that the current design will hold for the given load case. The reaction rod bracket, however, has to be redesigned to handle the load, since this is outside of the project's scope this will be a recommendation for further work. The highest stress in the new part is 300 MPa and the point where the stays are attached deflects 1.7 mm in the low stiffness setting with maximum loading.

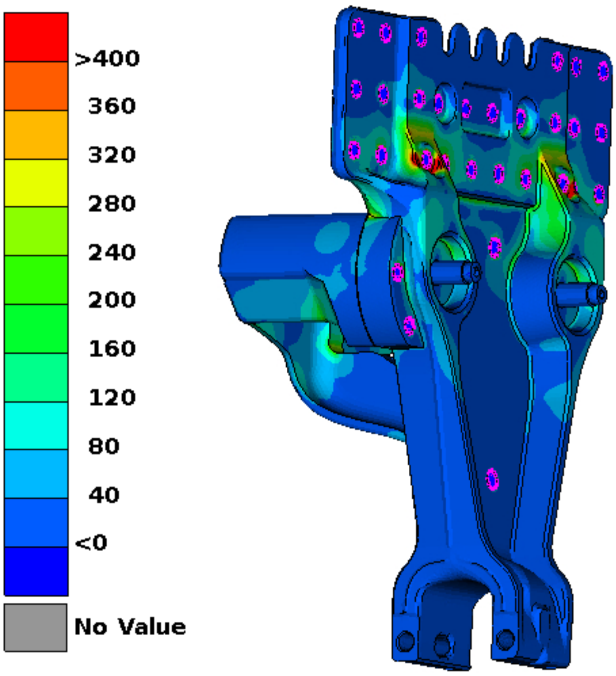


Figure 4.25: The model displays a combination of the maximum stresses in every node for the 6 different load cases.

4.7.5 Weight Estimation of the ARB-system

Table 4.5 shows weight for the complete solution compared to the old ARB. The weight of the new ARB system is 58% more than the existing ARB system.

Part	**Amount**	**Weight** (kg/part)	**Weight** (kg)
Motor	2	3	6
Lead screw	2	0,8	1,6
Axle	4	2,3	9,2
New bracket	2	15	30
stabilizer stay	4	5	20
Moving fork	2	5	10
Seal	2	1	2
The ARB (bar and arm)	2	38	76
Total			**151,8**
Existing ARB			
ARB	2	38	76
stabilizer stay	4	5	20
Total			**96**

Table 4.5: A weight approximation of the ARB system

5
Discussion

The discussion will treat the unforeseen circumstances that occurred because of Covid-19 and the effects it that may have had on the result of the project. The methodology used in the project will be reflected upon and possible skewed results. Recommendations for further work will also be suggested.

5.1 Complications of Covid-19

During this project, almost all development work was conducted at the Volvo Lundby office. This meant that close contact was kept with the department at Volvo during the construction of the concepts. It was possible to get feedback on ideas and concepts almost instantly from the department. When the Covid-19 epidemic unfolded, all of Volvo's employees were put on a short term lay-off, and with short notice, the contact with the department was broken. The project continued despite the lay-off but instead, the work had to take place remotely. There were only a few Skype meetings taking place during this period of the project with the supervisor at Volvo. This resulted in that there was a delay in feedback from the department on concepts and development work. When the lay-off was resolved and employees at the department at Volvo could come back to work, new information emerged that had an immediate effect on the progress of the project.

As pointed out earlier, the lay-off disrupted the communication with the department at Volvo and this proved to have a negative effect on the construction phase of the project. Because of the delay in information, a couple of weeks were put on developing the bracket called *the Y-Change Combination*, see Figure 4.16, that in the end was deemed not to function properly on the truck by the department at Volvo not to fulfil requirements that the project group was not aware of at the time. This is a consequence of false assumptions were being made or that the limitations were set too narrow. It is indicating how important it is to have close contact with the project customer and to include different sources of knowledge in the development work to receive feedback on the concepts. Time was spent on designing a concept that ended up to be eliminated, this is time that could have been used to further develop or test the concept to mature the final concept.

5.2 Methodology Reflection

The product development method that was used in the project was a combination of theory from "Product Design and Development" [12] and "Produktutveckling, effektiva metoder för konstruktion och design" [13]. How the combination of these two methods was used was based on experience and which of the methods the group was most comfortable to use. Since the group was small and the methods selected in some parts were based on personal preference the development and selection methods might have been biased and not the best-suited method.

Regarding the brainstorming process used for the concept generation. It was one thing to come up with solutions that could have two different stiffness levels. But a much more complex procedure to invent something that also could change between these two stiffness levels. For example, the concepts with the arm length change, the Pencil, see Figure 4.9, it was relatively easy to understand that a shorter or longer lever arm would influence the level stiffness in the ARB. But to brainstorm, and find a solution on how to the arm length change could be achieved was found to be much more complex.

The construction optimization that was used in the project helped to discover which parameters on an existing ARB that affect the stiffness the most. When the concept was selected, further development was based on experience, this means that there is an uncertainty if the best design of the selected concept has been found. Here could construction optimization have been used to investigate in which design of the final solution to use. The reason this was not implemented is a combination of lack of experience with the software and the special circumstances due to Covid-19.

5.3 Material Discussion

It would have been interesting to see how much financial and environmental savings that could have been done by changing from DI to ADI material in the final bracket. But since there was some rescheduling needed because of Covid-19 the time was not sufficient to cover this following kind of evaluation. By increasing the materials properties like yield strength and fatigue strength, in this case by changing from DI: GJS-500-7 to ADI: GJS-900-8, less material would be needed to get the equal performance in the part. This means a decrease in the dead weight of the truck and an increase in revenue by adding this weight in payload instead. There would also be savings in less CO_2 emissions from the truck when driving unloaded. Jernkontoret, [30] did this kind of calculation on the benefits of weight saving on a semitrailer for timber transport by changing to steel with higher strength. However, the weight savings would probably be around $10kg$ per *Y-Change bracket* and therefore relatively small in comparison to the weight of a whole truck. By that said, if this kind of procedure was done to all components on the truck the potential savings would be substantial.

An important factor that needs to be considered is the actual fatigue strength of the component compared to theoretical values. Although the material ADI has higher fatigue strength compared to DI in the material data provided in CES [19] and other sources. The fatigue data for materials are obtained from standardized test samples that are ground and polished to specification [9, 31]. The fatigue life is highly affected by surface roughness, and the surface roughness of an as-cast surface is much worse than a ground surface. Tests of DI with as-cast and ground surface show that the as-cast surface has much less fatigue strength compared to a ground surface [32]. The as-casted surface microstructure also contains a different microstructure with more defects compared to the ground surface beneath, this can have a negative effect on fatigue [32]. Volvo's test has shown that there is not so much difference in fatigue strength between an as-cast surface roughness ADI and DI.

5.4 Reflection on the Results

There are no known heavy-duty trucks on the market today that have the ability to change the ARB stiffness to get more traction on rough roads. We came to understand the difficulty of making alterations on the ARB itself because of the high loads and limited packaging space in the area between the reaction rods. The final concept can hereby be seen as an addition to the development of heavy-duty trucks suspension systems by that the stiffness of the ARB can be changed by changing where the stabilizer stays upper mounting points are located.

The dimension of the bar on the ARB was not included in the construction optimization when the different parameters were examined. Because the bar was excluded the optimization in modeFRONTIER might not have found the best concept on how to change the stiffness with as little variable change as possible. The decision to exclude the bar in the optimization was because the group wanted to move away from the high stresses in the bar and to keep the characteristics of the existing ARB. The exact desired stiffness levels on the ARB are not explored and is something that should be done before an optimization on the dimension of the ARB can be performed. The exact distance of *the Y-Change* and dimensions of the ARB is something else that can and should be optimized to find the exact stiffness levels desired.

The stiffness changing design is not optimized in this solution. Time was allocated to create a concept that was able to be realized and function on a truck. After the feedback from Volvo on that the concept *The Y-Change Combination* would not work, the remaining time of the project was limited and it was not possible to both re-design the concept and optimize it. It was regarded as more valuable, by the project group, to go back to the drawing table and re-design the concept to the new *Y-change bracket* that is regarded to function on a heavy-duty truck instead of continuing optimizing the rejected concept *the Y-Change Combination* on exactly how much the stabilizer stays would have to move in order to produce a change in stiffness by a factor 2.

One of the major problems with the development was how to make sure that the design would work no matter what the conditions are. Volvo pushed on the importance that the variable function had to be protected from dirt and water. The new bracket is designed to be able to encapsulate the moving function. By mount all parts inside the bracket and then seal the bracket together with the reaction rod bracket it will create a tight seal. The idea is that the selected lead screw will be self-locking, meaning that there is no need to apply brakes or an external locking device to keep the stabilizer stays in place in. The lead screw will hold them in place in inner- or outer position and only allow the stays to change position when the screw is turned by the electric motor.

At the beginning of the project, it was anticipated to develop a new type of ARB that was interchangeable with the current GRAS 2.7 - ARB. During the project, it became clear that the high loads on the ARB in combination with the little packaging space. Taking to account the surrounding parts as reaction rods, axles and also ground clearance. Made it difficult to develop a bolt-on variable ARB that could be swapped out on any truck with a GRAS 2.7. The GRAS 2.7 was also a complex component to alter, as described in the Theory chapter 2, the GRAS 2.7 is a well-made component regarding the high loads it has to endure. Therefore the scope of the project widened from looking only at the ARB as a system to also include the stabilizer stays that mount the ARB to the frame of the truck.

5.5 Recommendation for Future Work

The concept called *the Engine* was the most active of all concepts. It has similarities with the variable Schaeffler ARB for cars, mentioned in the theory under existing solutions 2.3. It has an electric motor in the centre that applies torque to counteract the roll of the vehicle. In this project, the packaging and the anticipated cost for this concept to be implemented on the truck was the reason for it to be eliminated. It would be interesting for future projects to investigate if this concept would work with the high loads that the ARB on a truck is subjected to.

The new y-change bracket that was developed in this project could be further developed and optimized before it is tested and implemented in production. Some recommendations for future development work of the new y-change bracket are:

- Optimize and investigate exactly how much change in the y-direction is needed to change the stiffness by a factor 2.
- Optimize the casting process and casting shape for the new y-change bracket.
- Investigate the optimum number of bolts and the placement of them.
- Develop the transmission and calculate the gearing ratio between the electric motor and the lead screw.
- How much force that is needed from the electric motor to move the stays on the axles in real-life load cases on the truck need to be tested and evaluated.
- Redesign the reaction rod bracket to match the design of *the Y-Change bracket* and decrease maximum stress on the outside.

6
Conclusion

The project work has found a way to vary the roll stiffness of the anti-roll bar on a heavy-duty truck. This solution moves the upper attachment points where the stabilizer stay is connected to the frame towards the centre of the truck. The stabilizer stays are pushed on an axle that is attached to a bracket on the frame. When the stabilizer stays moves towards the centre of the truck the contributing roll stiffness from the ARB decrease. This means that the contributing roll stiffness from the ARB is lowered without changing the characteristics of the existing ARB. The foundation of this solution was generated with construction optimization calculations on a general ARB. The calculations found the concept with the max roll angle that differed the least in layout from the existing ARB.

The solution that is proposed by this project is a result from the given limitations and Volvo's requirements and desires. The main requirements and desires that resulted in the final concept was: The solution should be robust, have as few moving parts possible and alter the existing layout of surrounding components as little as possible.

The best-suited material for the selected concept is ADI cast iron for performance, low weight and cost. The base of this selection comes from an analysis of data from CES, where the best group of materials concerning fatigue limited design and deflection limited design were found. With the first screening, the cost and weight of the materials could be analyzed. With additional information searching and consultation from Volvo Materials Technology, the selection and recommendation of which material to be used could be done.

The specification of the issue under investigation for this project was:

- How should the anti-roll bar be designed to allow the stiffness to be actively selected between one low and one high stiffness level?
- What material should be used for the best performance, lowest environmental impact and cost?
- What parameters on the anti-roll bar is impacting the stiffness the most?
- What is the optimal solution with the selected design?

As previously mentioned the project has found a way to allow the stiffness to be actively selected between to different levels of stiffness. The material that has the best combination of performance, low environmental impact and cost is ADI. The parameter that affects the ARB the most without changing the characteristics of the existing ARB is the attachment of the stabilizer stays to the frame. How the optimal solution of the selected concept would be designed has not been investigated thoroughly due to special circumstances and is something that is a recommendation for future work.

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A

Appendix - Material Selection

A.1 Derivation of the Material Index 1, M_1

A beam fixed in one end and free in the other where the free end is subjected to a bending force.

Objective: Minimize mass, m
Constraint: Stress, $\sigma \leq \sigma_{u,fatigue}$
Free variables: Section thickness, t
Parameters:
$F = 70\ kN$
$L = 200\ mm$
$A = t^2$
$y_m = {}^{t}/_{2}$
$I = \frac{t^4}{12}$
$M = F \cdot L$
$m_1 = \rho \cdot L \cdot A$

$$\begin{aligned}
\sigma_u &= \frac{M \cdot y_m}{I} = \frac{F \cdot L}{{}^{t^4}/_{12}} \cdot \frac{t}{2} = \frac{F \cdot L \cdot 6}{t^3} \Rightarrow \\
t &= \left(\frac{F \cdot L \cdot 6}{\sigma_u}\right)^{1/3} \Rightarrow \\
m_1 &= \rho \cdot L \cdot \left(\frac{F \cdot L \cdot 6}{\sigma_u}\right)^{2/3} = \left(\frac{\rho}{\sigma^{2/3}}\right) \cdot L \cdot \left(F \cdot L \cdot 6\right)^{2/3}
\end{aligned} \tag{A.1}$$

This means that:

$$\text{Minimize mass } \Rightarrow M_1 = \frac{\rho}{\sigma_u^{2/3}} \tag{A.2}$$

$$\text{Minimize cost } \Rightarrow M_{1C} = \frac{C_m \cdot \rho}{\sigma_u^{2/3}} \tag{A.3}$$

A.2 Derivation of the Material Index 2, M_2

A beam fixed in one end and free in the other where the free end is subjected to a bending force.

Objective: Minimize mass, m
Constraint: Deflection, $\delta \leq 1\ mm$
Free variables: Section thickness, t
Parameters:
$F = 70\ kN$
$L = 200\ mm$
$A = t^2$
$C_1 = 3$
$I = \frac{t^4}{12}$
$M = F \cdot L$
$m_1 = \rho \cdot L \cdot A$

$$
\begin{aligned}
S = \frac{F}{\delta} &= \frac{C_1 \cdot E \cdot I}{L^3} \Rightarrow \\
I = \left(\frac{F \cdot L^3}{3 \cdot E \cdot \delta}\right) &= \frac{A^2}{12} \Rightarrow \\
A^2 &= \frac{4 \cdot F \cdot L^3}{E \cdot \delta} \Rightarrow \\
m_2 = \rho \cdot L \cdot \left(\frac{4 \cdot F \cdot L^3}{\delta \cdot E}\right)^{1/2} &= \frac{\rho}{\left(E\right)^{1/2}} \cdot L \cdot \left(\frac{6 \cdot F \cdot L^3}{\delta}\right)^{1/2}
\end{aligned}
\tag{A.4}
$$

This means that:

$$\text{Minimize mass } \Rightarrow M_2 = \frac{\rho}{E^{1/2}} \tag{A.5}$$

$$\text{Minimize cost } \Rightarrow M_{2C} = \frac{C_m \cdot \rho}{E^{1/2}} \tag{A.6}$$

A.3 Derivation of the Coupling Line

$$C = \frac{M_1}{M_2} \tag{A.7}$$

Equation A.1 and A.4 gives the following:

$$C = \frac{L \cdot \left(F \cdot L \cdot 6\right)^{2/3}}{L \cdot \left(\frac{6 \cdot F \cdot L^3}{\delta}\right)^{1/2}} \tag{A.8}$$

Different displacement, δ, implies different values on C.

$$
\begin{aligned}
\delta = 1mm &\implies C_1 = 0.13 \\
\delta = 2mm &\implies C_2 = 0.18 \\
\delta = 3mm &\implies C_3 = 0.22 \\
\delta = 4mm &\implies C_4 = 0.26 \\
\delta = 5mm &\implies C_5 = 0.29
\end{aligned}
$$

A.4 Figures

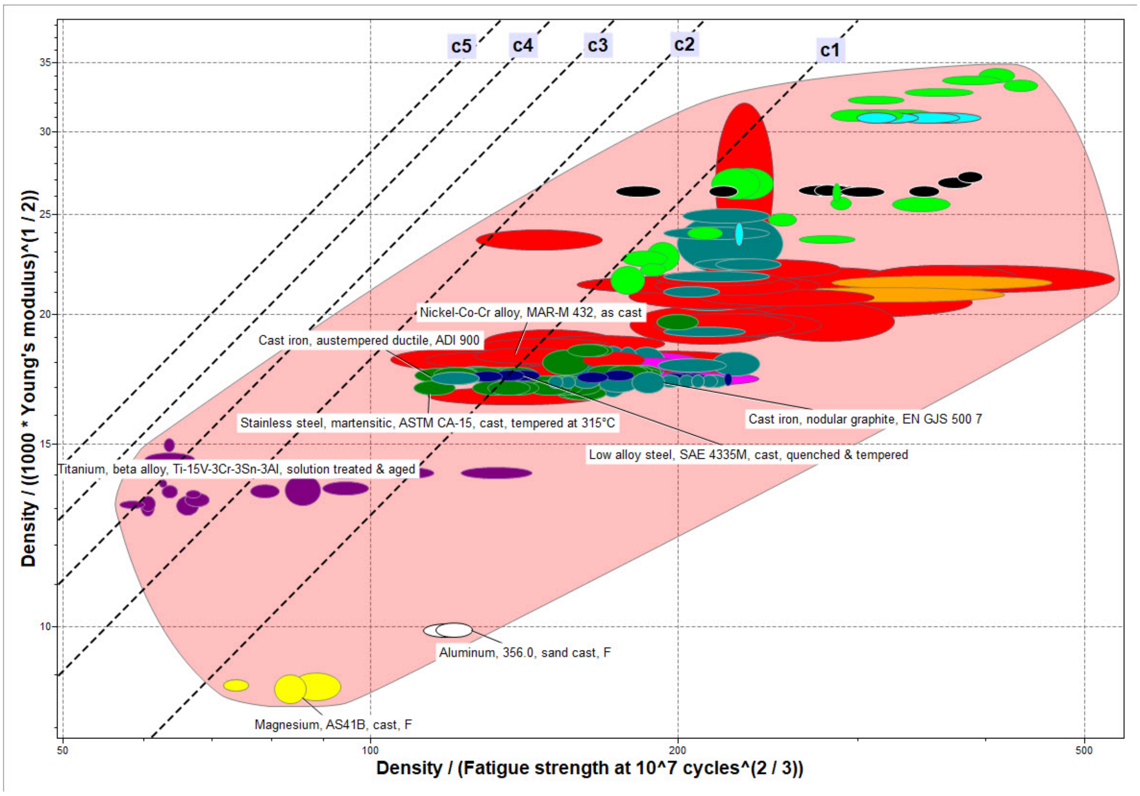

Figure A.1: Coupling plot Stiffness performance to minimize mass (M_2) vs. Fatigue performance to minimize mass (M_1). Coupling lines $C_1 - C_5$ to identify the dimensioning material index to minimize mass.

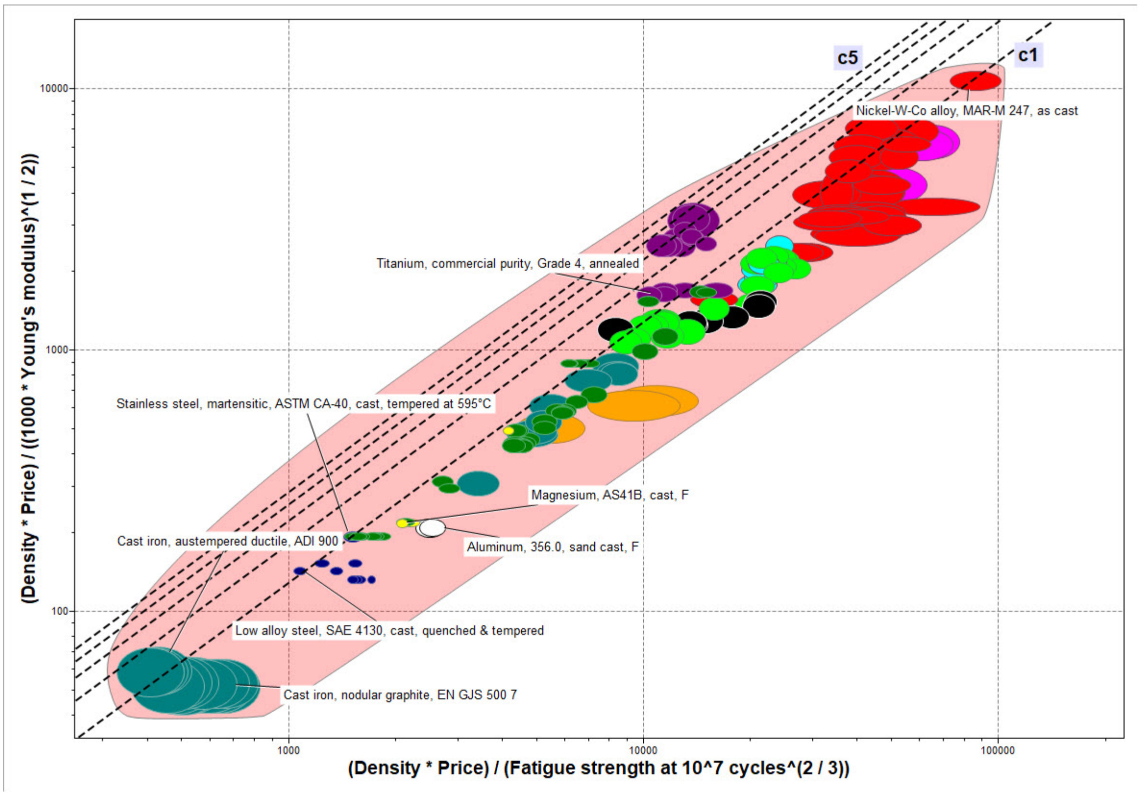

Figure A.2: Coupling plot Stiffness performance to minimize cost (M_{2C}) vs. Fatigue performance to minimize cost (M_{1C}). Coupling lines $C_1 - C_5$ to identify the dimensioning material index to minimize cost.

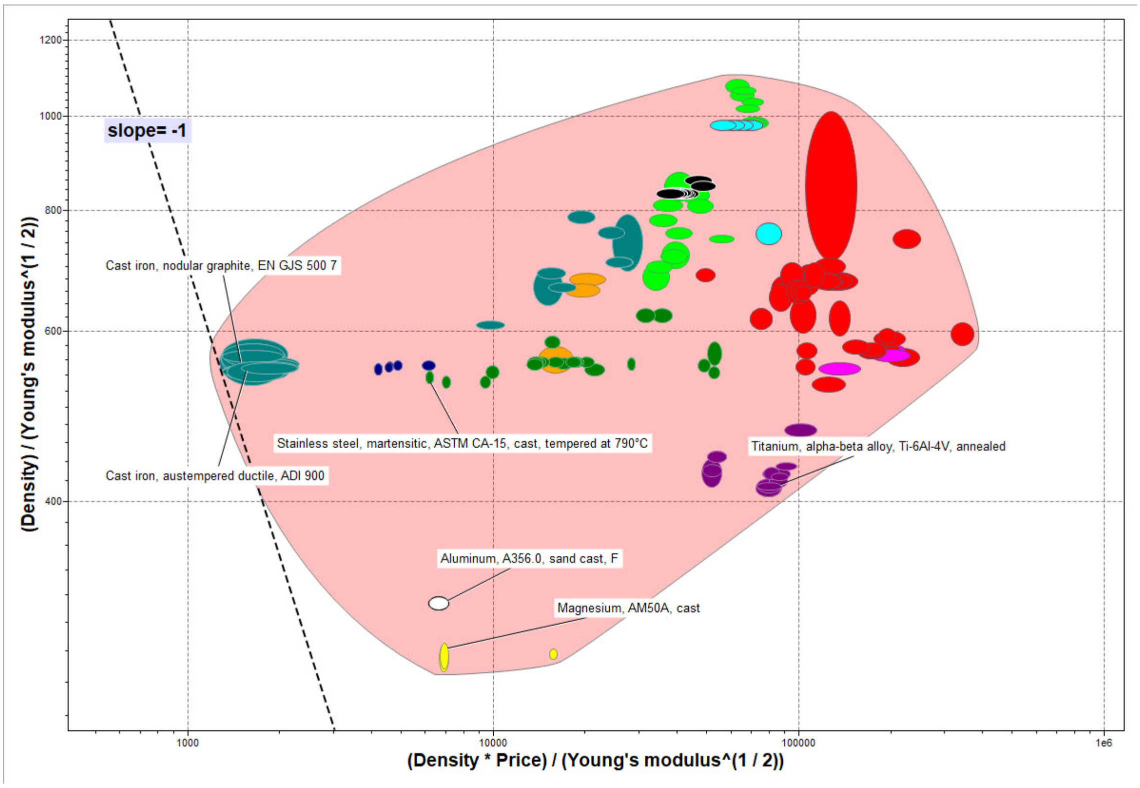

Figure A.3: Trade-off plot on Stiffness performance to minimize mass (M_2) vs. Stiffness performance to minimize cost (M_{2C}). Line with slope -1 represent that both indices are weighed equally.

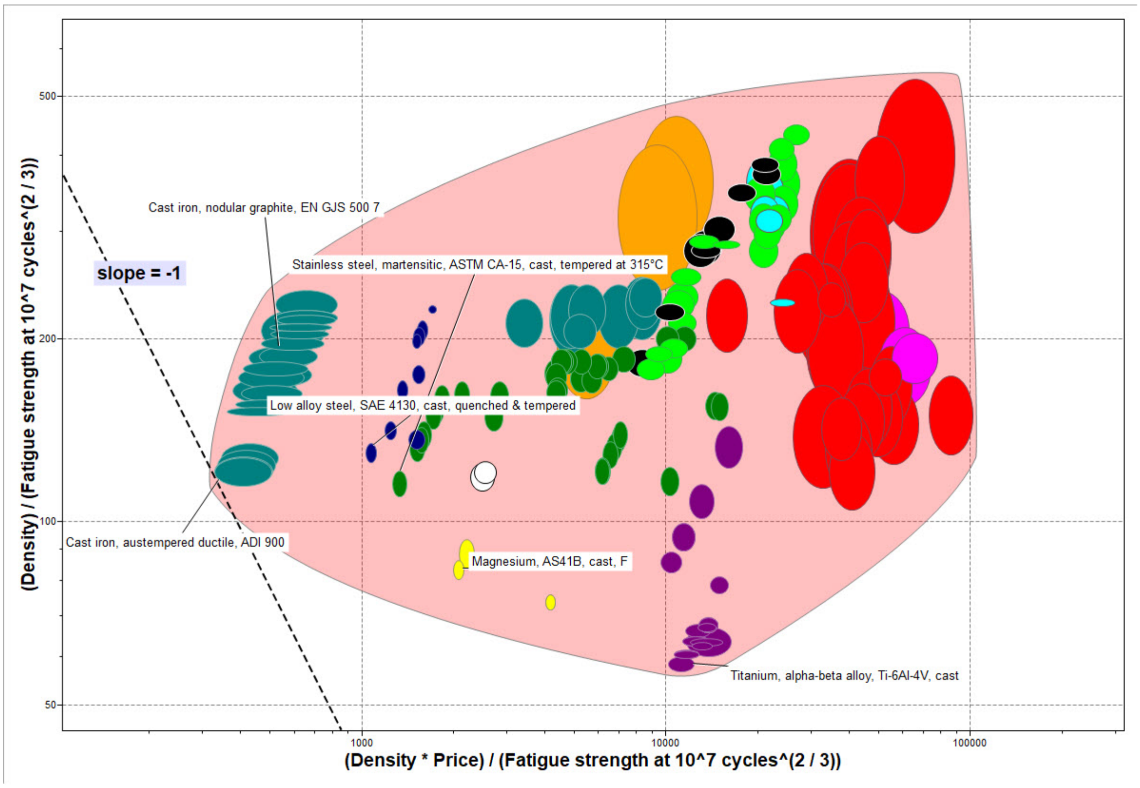

Figure A.4: Trade-off plot on Fatigue performance to minimize mass (M_1) vs. Fatigue performance to minimize cost (M_{1C}). Line with slope -1 represent that both indices are weighed equally.

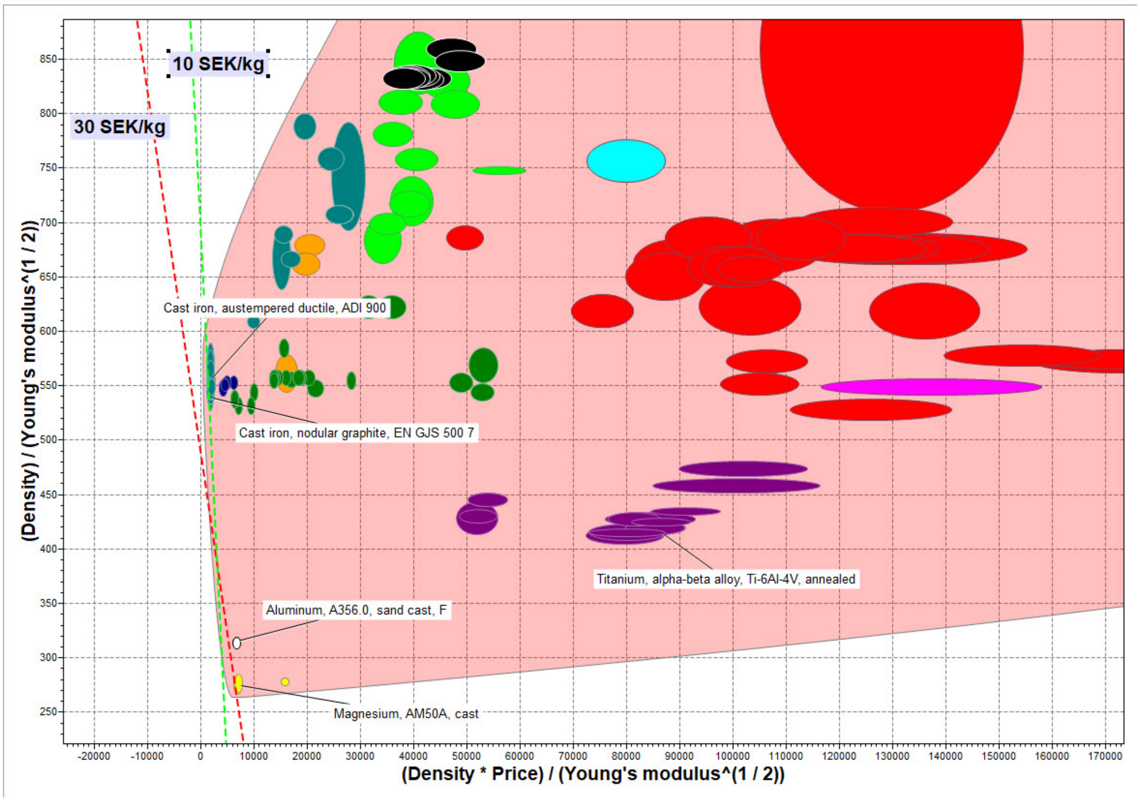

Figure A.5: Trade-off plot with exchange constants 30 $^{SEK}/_{kg}$ and 10 $^{SEK}/_{kg}$ on Stiffness performance to minimize mass (M_2) vs. Stiffness performance to minimize cost (M_{2C}).

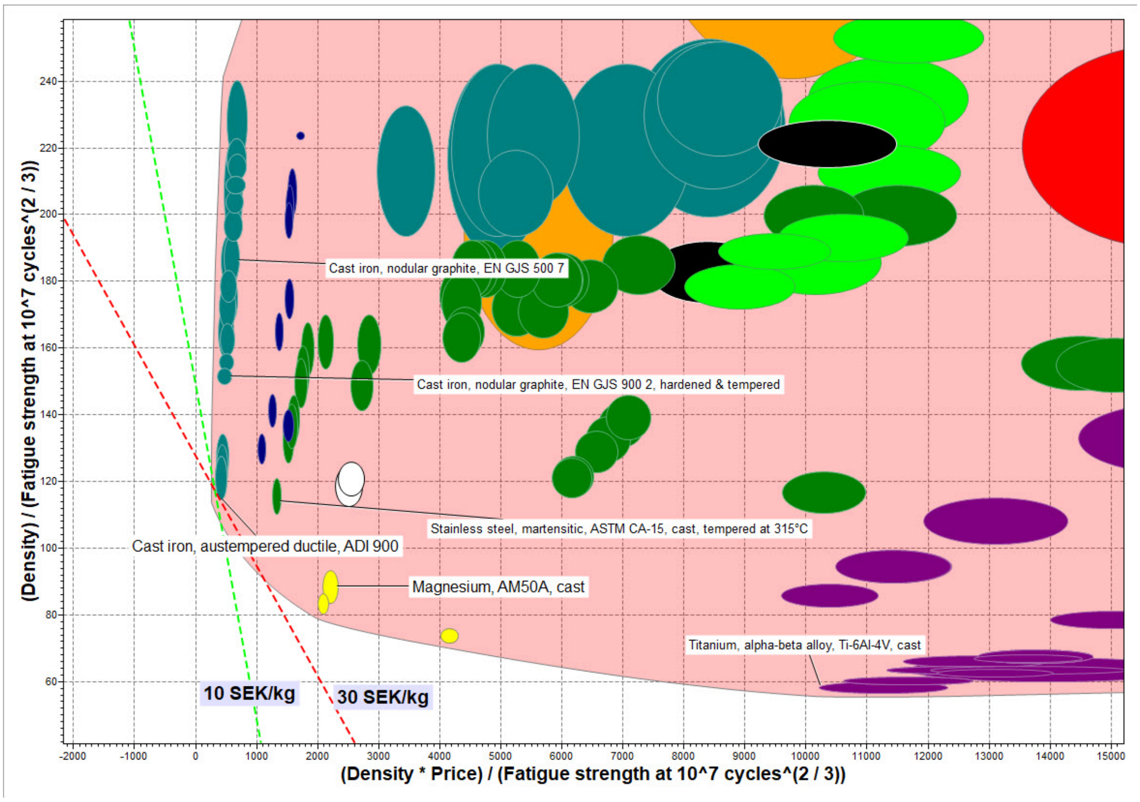

Figure A.6: Trade-off plot with exchange constants 30 $^{SEK}/_{kg}$ and 10 $^{SEK}/_{kg}$ on Fatigue performance to minimize mass (M_1) vs. Fatigue performance to minimize cost (M_{1C}).

B

Appendix - Requirement Chart

Requirements/Desires	R/D	Justification	Verification/Evaluation
Appearance			
Fit inside available packaging area (dimesions)	R	Volvo	CAD
Mounted to existing mounting points, (torsion bar e.g.)	D	Volvo	CAD
Not alter the existing layout	D	Volvo	CAD, packaging
Be able to change between two different stiffnesses	R	Volvo	Prototype test
Be able to change between three or more different stiffnesses	D	Team	Prototype test
Be able to change stepless between different stiffnesses	D	Team	Prototype test
Take advantage of existing air system instead of implementing new sub sys.	D	Team	CAD
Electrical steering			
Allow for existing ground clearence	R	Volvo	CAD
Weight of the part less than 50 kg	R	Volvo	CAD
Lowest possible weight of the part	D	Volvo	CAD
Compatible with several chassi variants	D	Volvo	
Be able to scale down to a softer setting, for front suspension	D	Volvo	Simulation/CAD
Safety			
No fracture during whole lifecycle,	D	Team	Simulation
Should always work, be failsafe if not activated	D	Team	Prototype test
Changing stiffness should not be able to be done by accident	R	Team	User test
Not impact the overall safety of the truck,	R	Team	Test
Changing stiffness should not harm the user, while driving	D	Team	User test
Handling			
Possible to change stiffness between two steps with one action from the user	R	Team	User test
The stiffness should be able to change for one person from inside the cab	R	Team	User test
The stiffness changing operation on the product should done in as few step as possible	D	Team	Prototype test
Easy to operate, by the driver	D	Team	User test
Possibility to change stiffness automaticly, computor controlled	D	Team	User test
Be able to change stiffness at any time	D	Team	Test
Lifecycle			
95 % of the ARB's should last longer than lifetime of a truck	R	Volvo	Simulation
99 % of the ARB's should last longer than lifetime of a truck	D	Team	Simulation
The product should be able to recycle	R		
The product should be made from recycled material if it is possible	D	Team	
Materials should if possible be made from sustainable resources	D	Team	
Minimal ecological footprint	D	Team	
Weldability of the material if the solution require it	D	Team	Test or research docum
formability, e.g. cast or wrought depending of the solution	D	Team	Test or research docum
Robustness			
Support the weight of the truck	R	Volvo	Simulation
Should be able to handle single peak loads without deformation	R	Volvo	Simulation
Stiffnes equals to 1250 N/mm +-10%, in high setting	R	Volvo	Simulation
Stiffnes equals to 625 N/mm +-10%, in low setting	R	Volvo	Simulation
Possible to change the stiffness with a factor 2 between the higher and lower setting	D	Volvo	Simulation
Fatigue life longer than the expected lifetime of the truck, volvo requirement?	R	Volvo	Simulation
Fracture toughness condition	R	Volvo	Question
Horisontal deflection less than 100 mm from horisontal peak loads	R		
No buckling when exposed to single peak loads	R	Volvo	Simulation
Be able to handle the vibrations from the road	R	Volvo	Simulation
Not impact the possibility to rise and lower the vehicle			
Regular dampening +/- 50 mm	R	Volvo	Simulation
Environmental Conditions			
Low risk for corrosion volvo specification on "no rust??"	R	Volvo	Test?
Withstand the environment under a truck	R	Volvo	User test
Be able to be used between tempratures of -40 and +50 C	R		
Not have areas where water/liquid can get trapped and cause corrosion	R	Volvo	CAD
Be able do handle snow that gets stuck underneath the truck	R	Volvo	CAD
Market			
The anti-roll bar should not cost more than 10 times the existing one	R		Cost analys
The anti-roll bar should not cost more than 5 times the existing one	D		Cost analys
Packaging, Storaging and Shipping			
Use parts available at Volvo	D	Team	CAD
Prefferably be able to be mounted as an aftermarket part on existing trucks	D		
Production and Manufacturing			
It should be possible to manufacture with existing tools	D		
The production should not require additional facilities	D		

Figure B.1: The requirement chart

C

Appendix - Concept Sketches Iteration 1

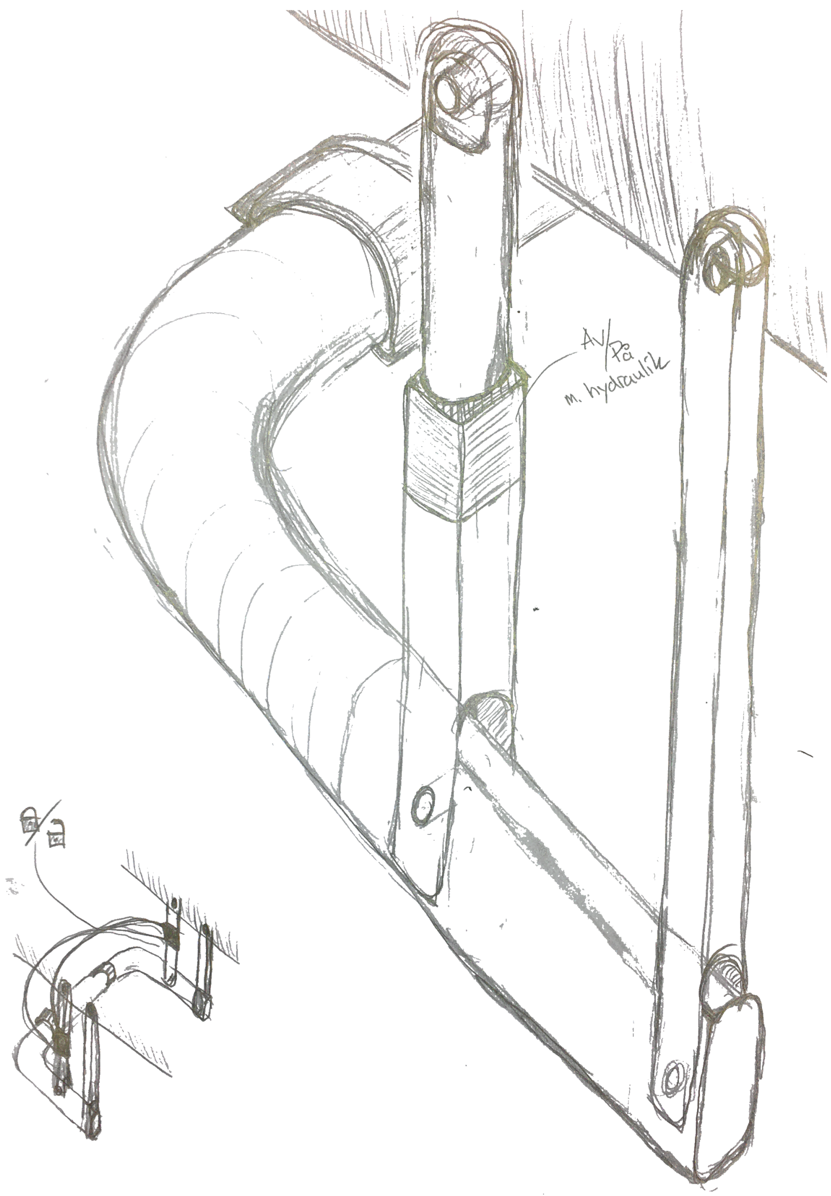

Figure C.1: The Extra Strut

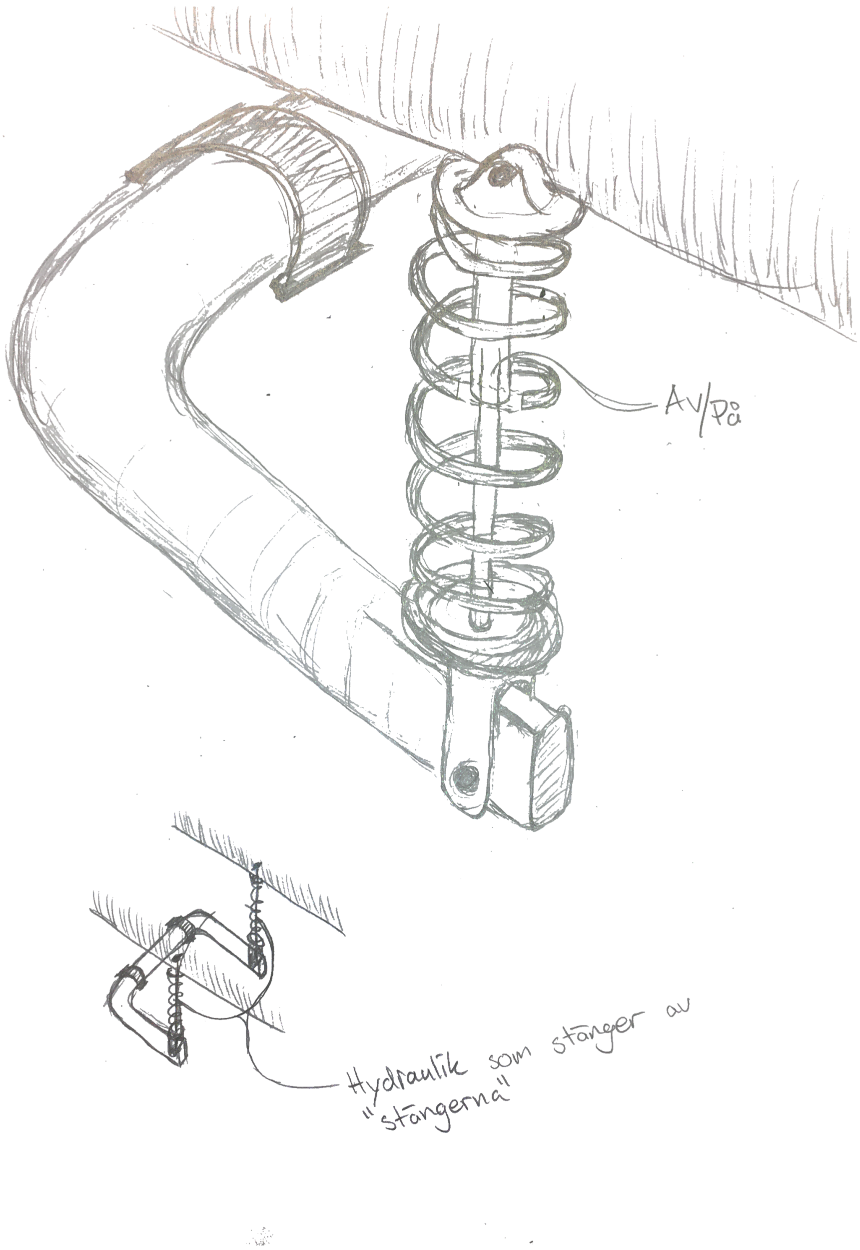

Figure C.2: The Spring strut

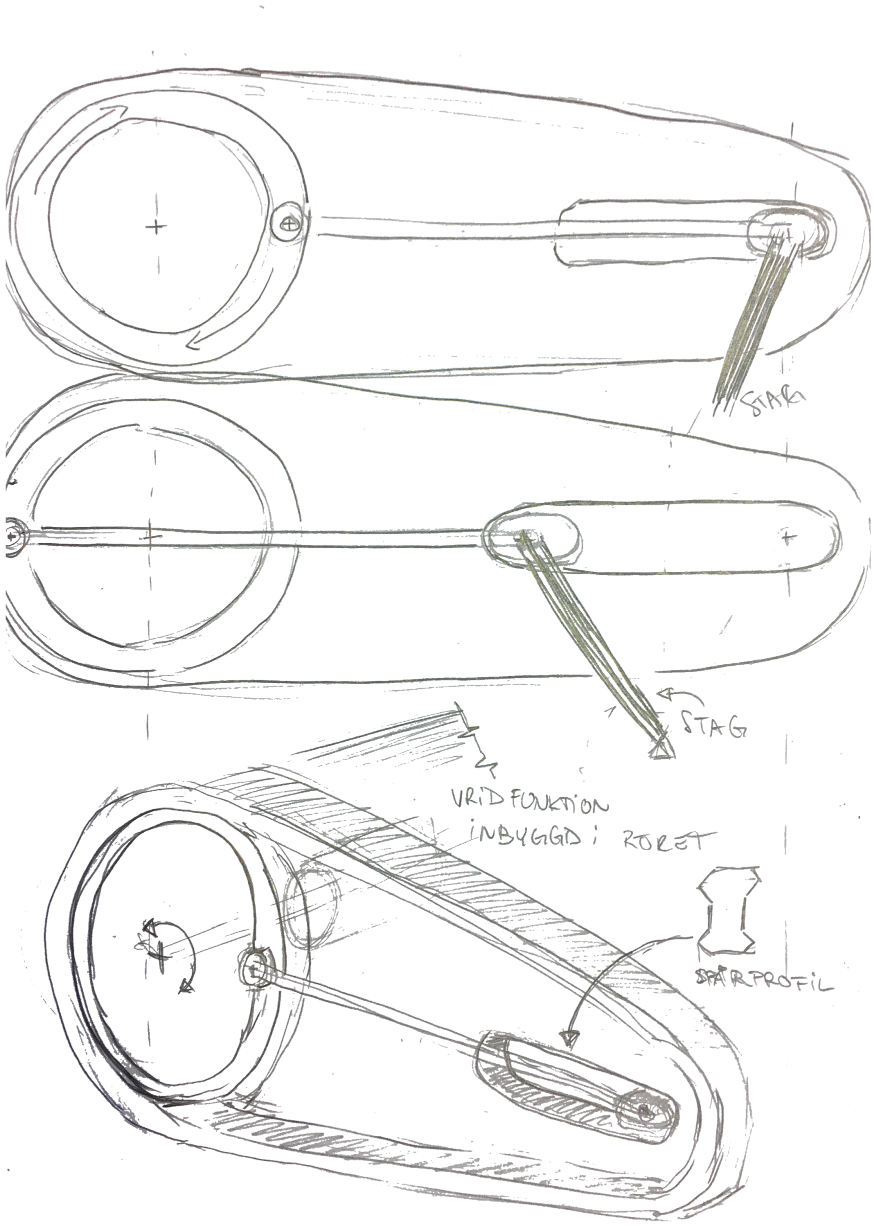

Figure C.3: The Spinning Disc

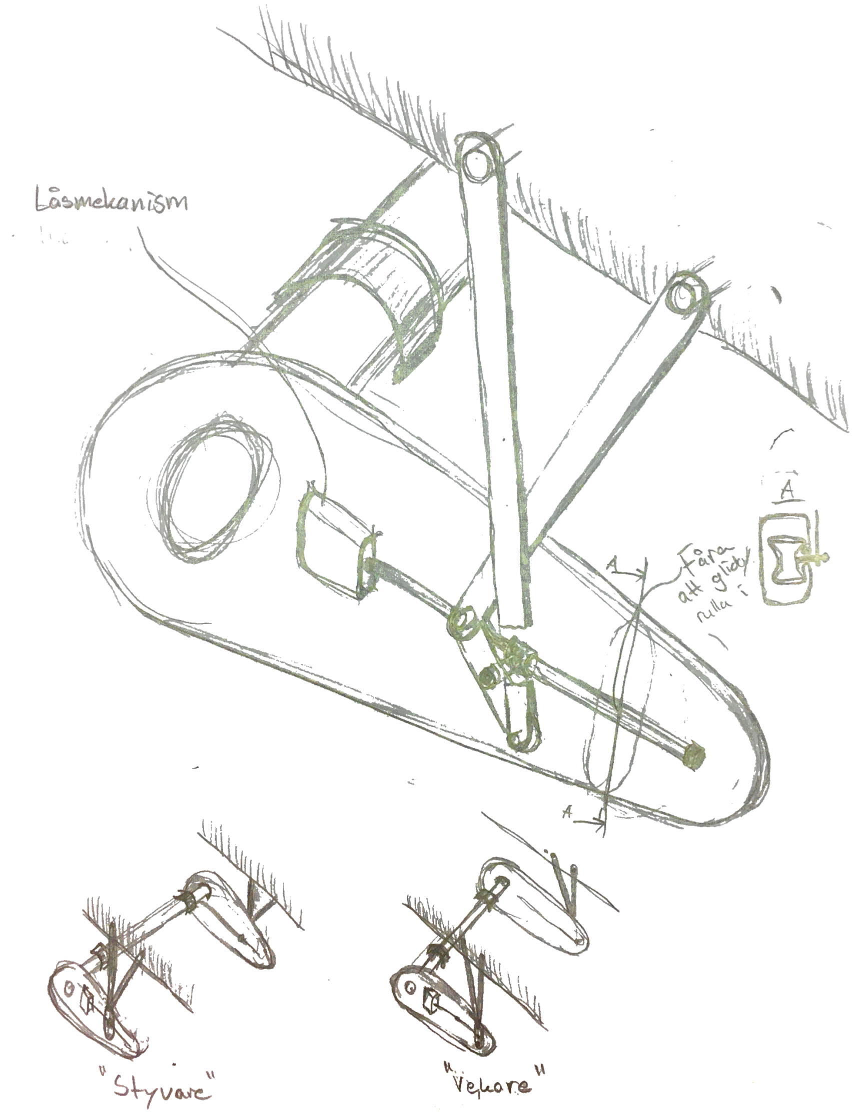

Figure C.4: The Pencil 1 - Chebyshev

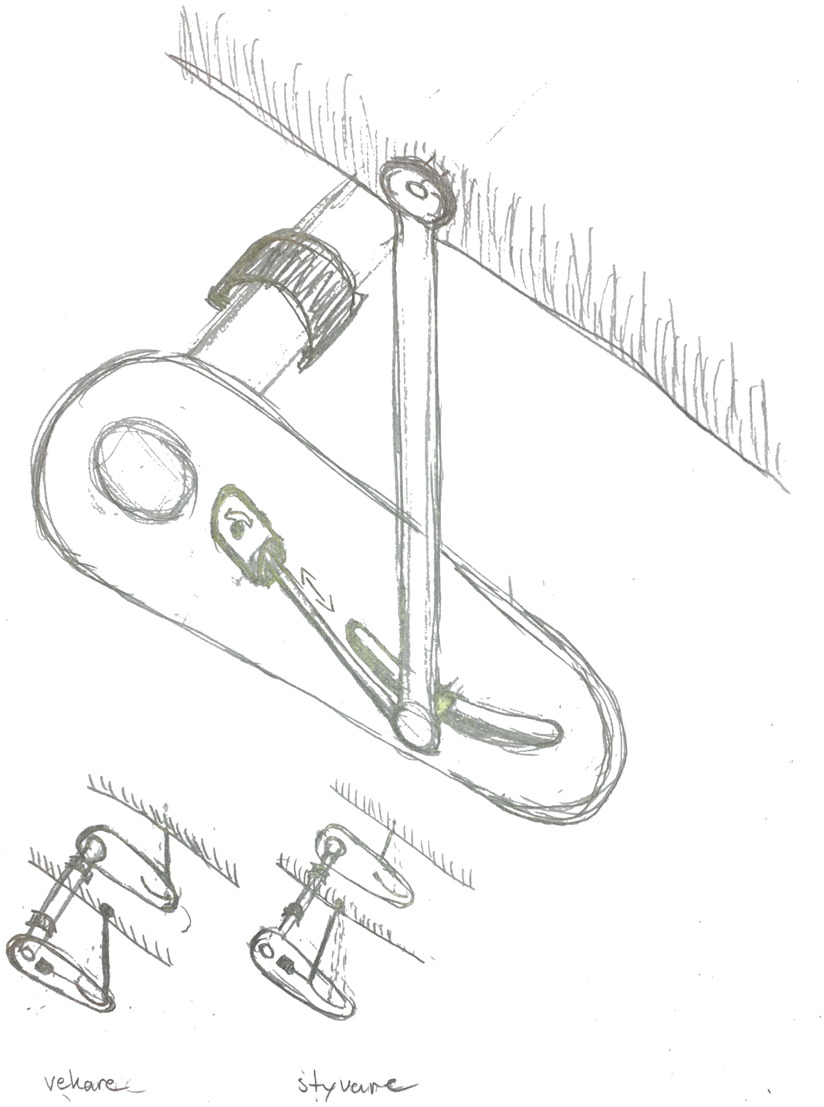

Figure C.5: The Pencil 2

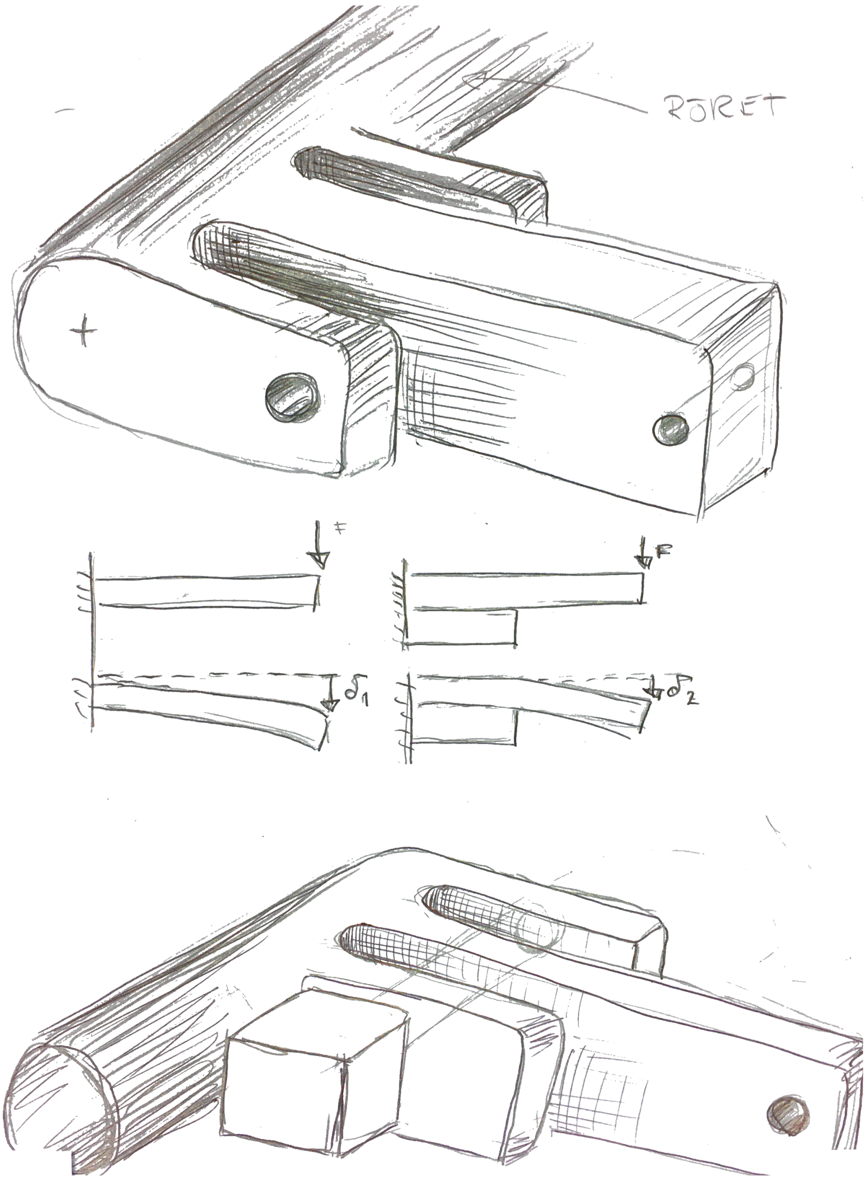

Figure C.6: The Extra arm

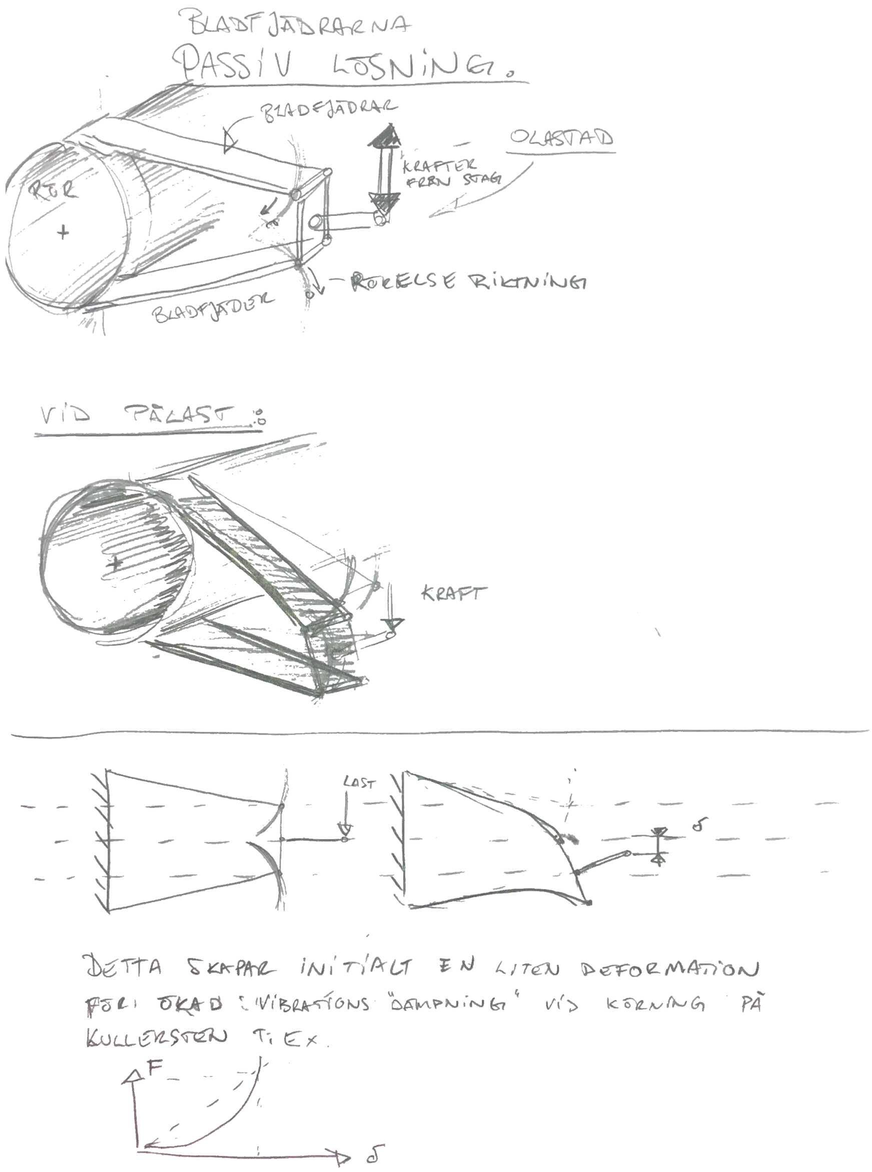

Figure C.7: The Leaf Spring

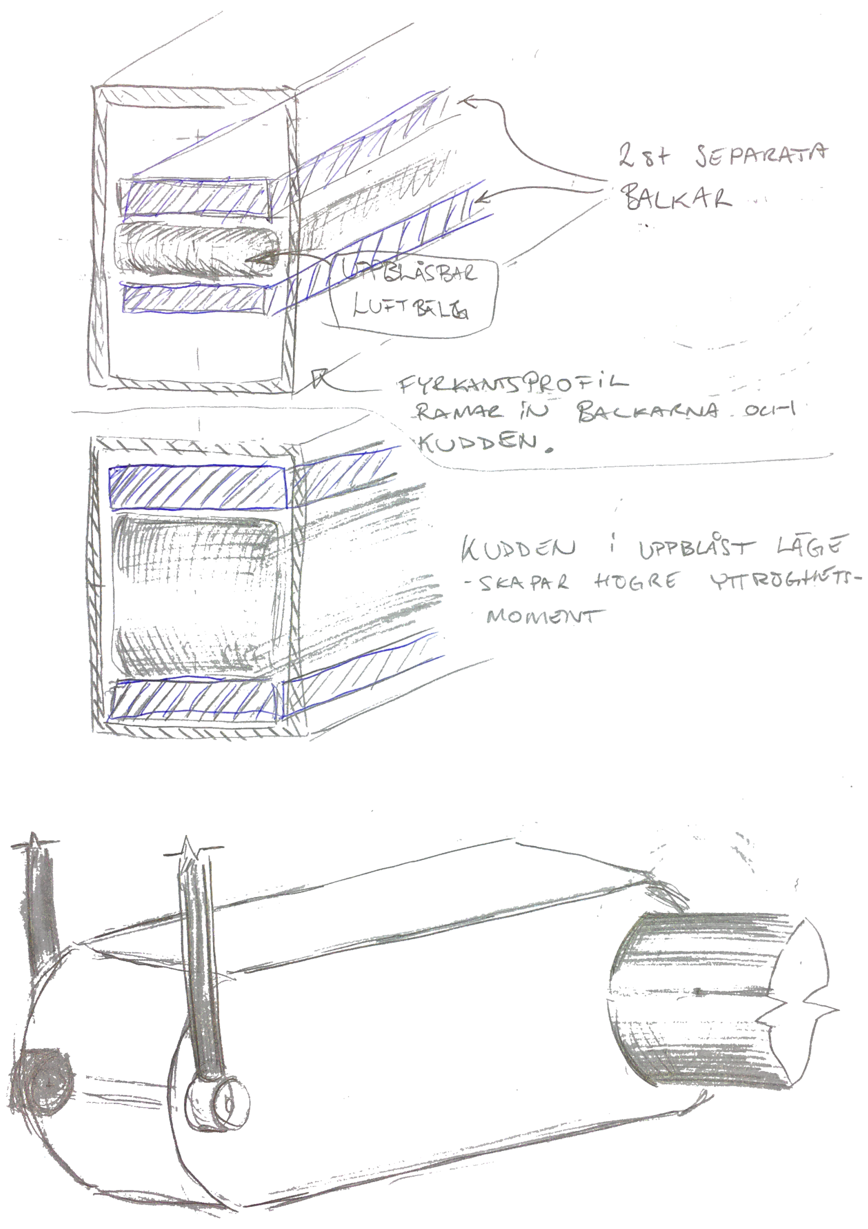

Figure C.8: The I-Beam

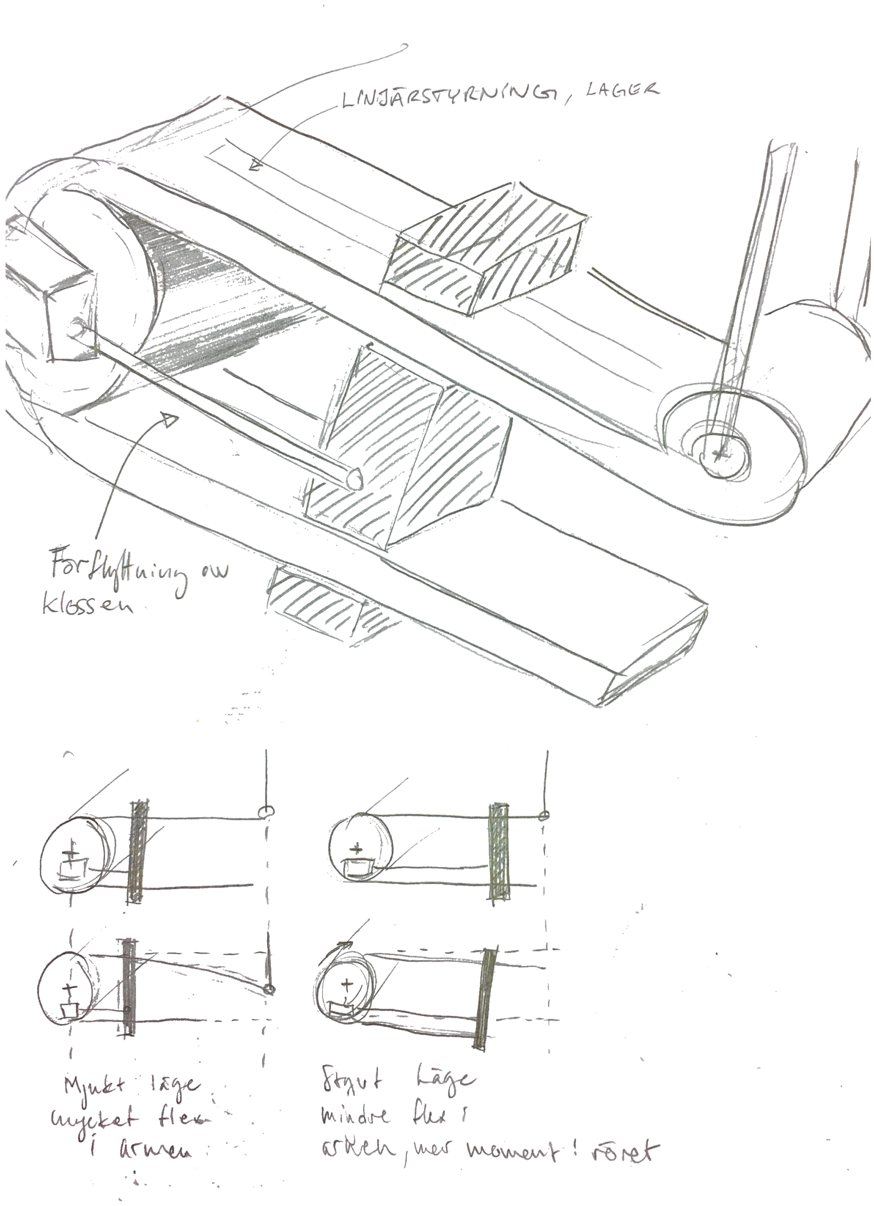

Figure C.9: The Waist

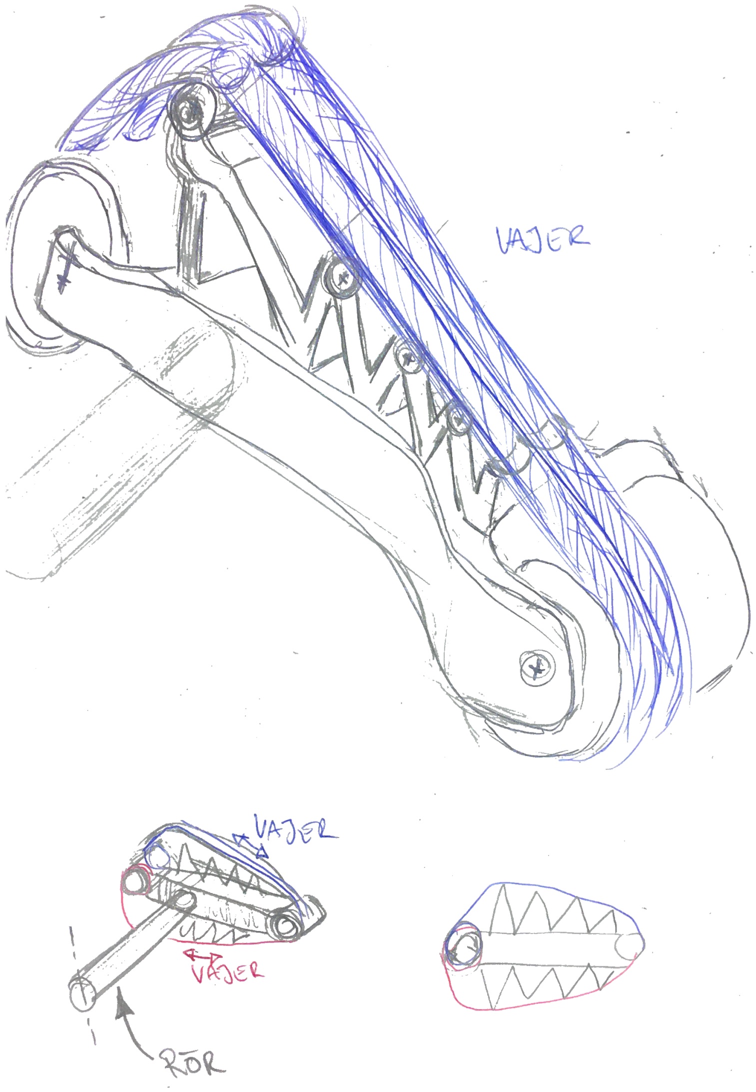

Figure C.10: The Wire

D

Appendix - Concept Sketches Iteration 2

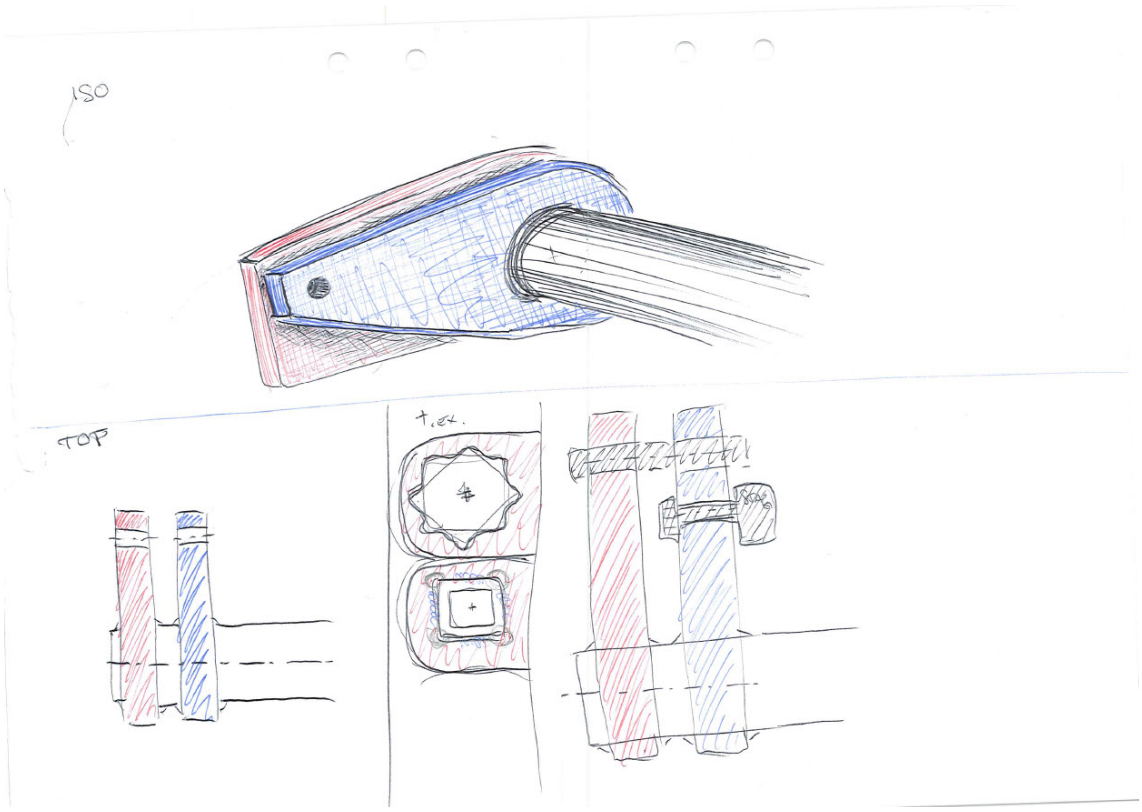

Figure D.1: The Extra arm, arms connect with a pin

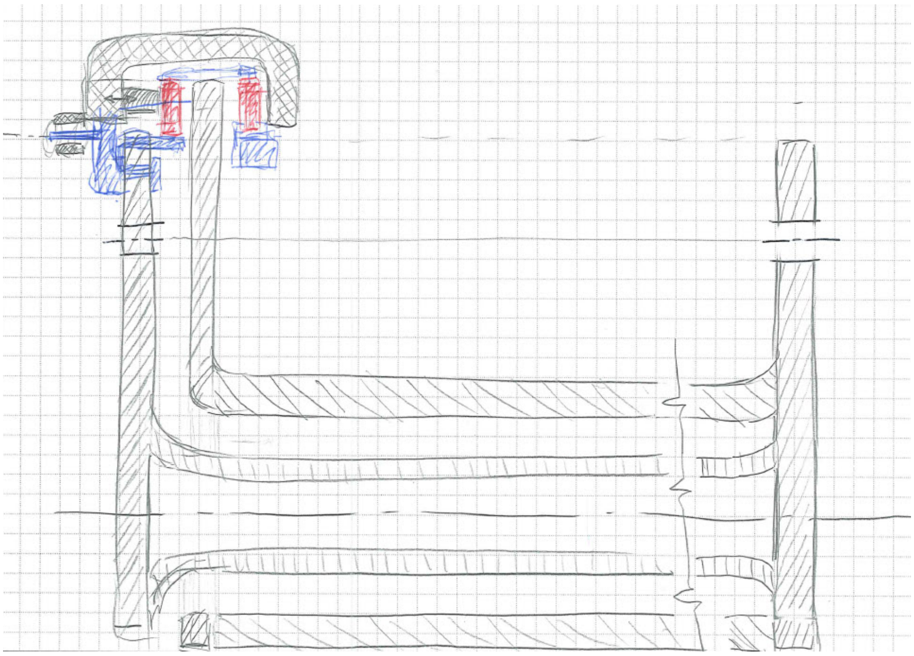

Figure D.2: The Extra arm, in this case also a second torsion bar that increase the stiffness. Connection and disconnection is achieved with a brake caliper

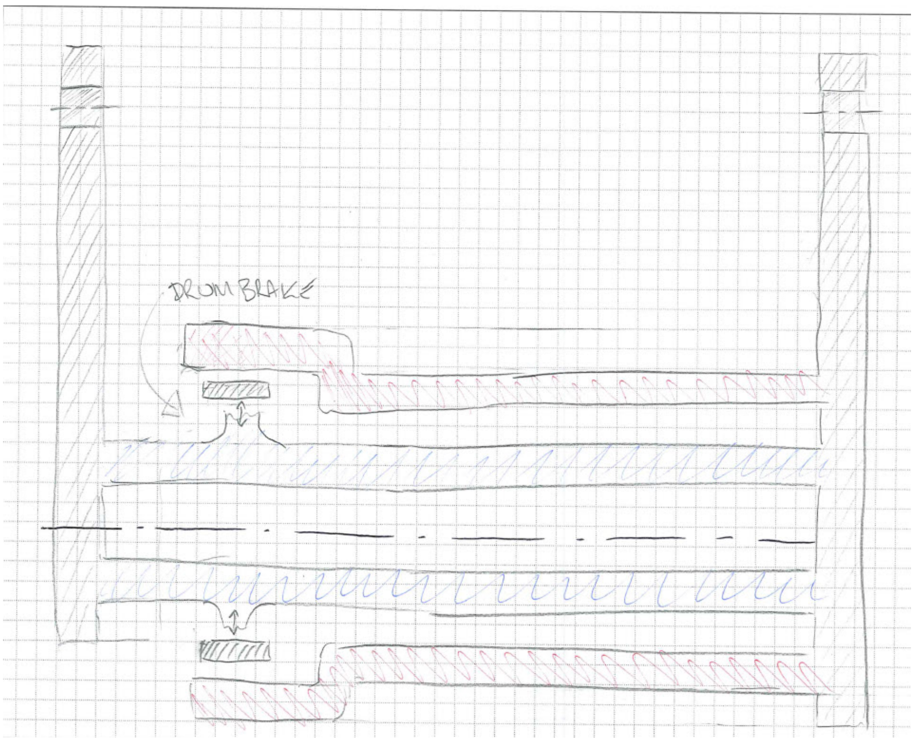

Figure D.3: The Extra arm, in this case it is an extra bar that increase the stiffness. The activation is done by a drum brake.

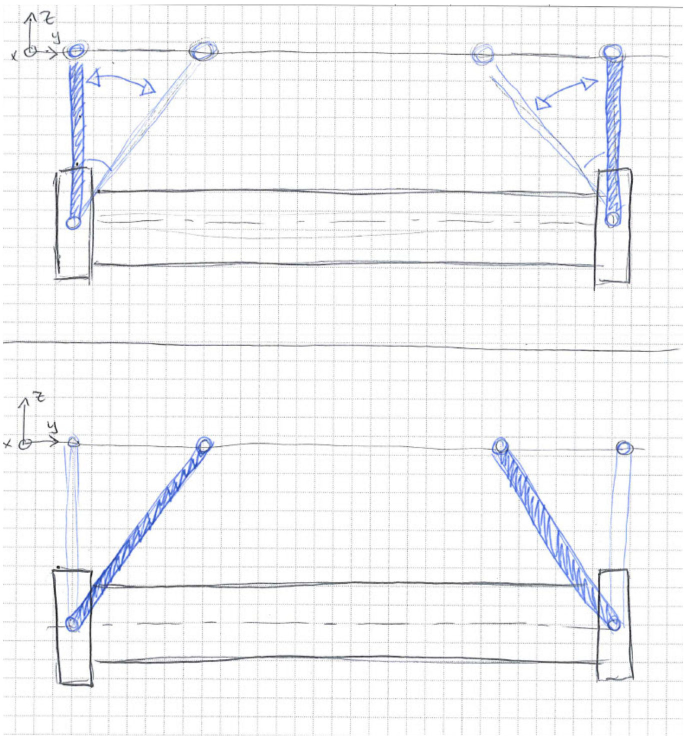

Figure D.4: The Y-Change

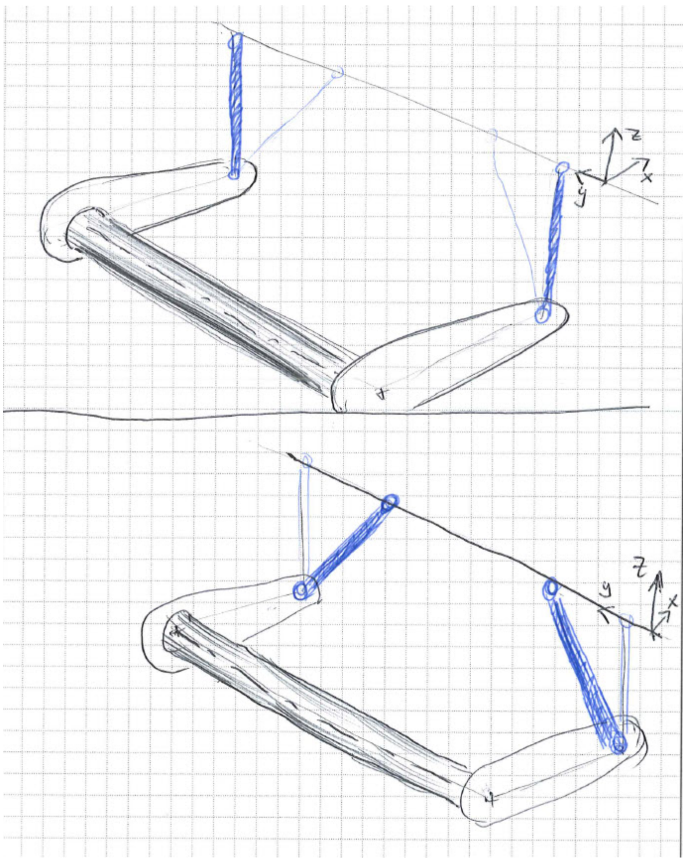

Figure D.5: The Y-Change

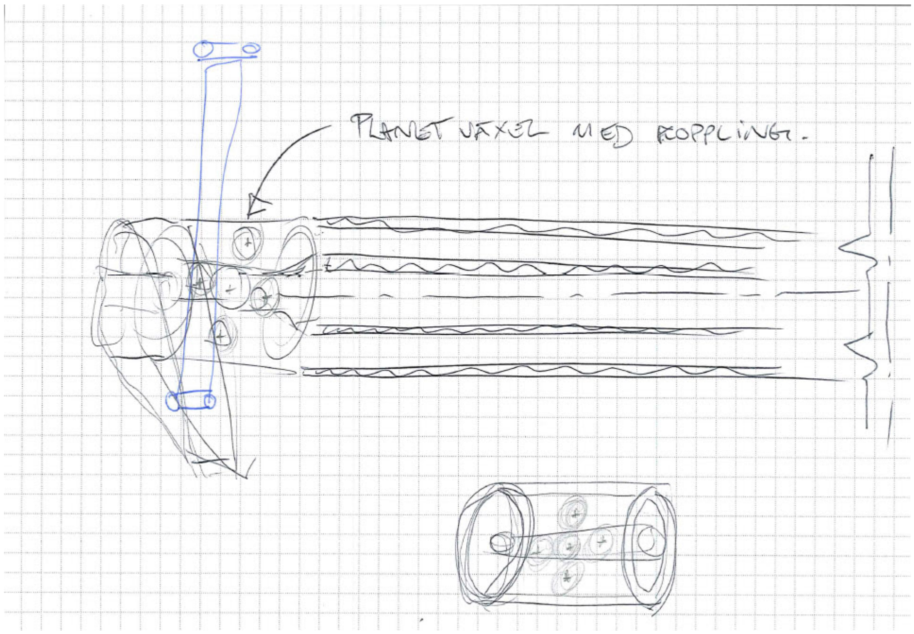

Figure D.6: The planetary gear, the idea is that the planetary gears on each side shift the amount of torque that is transferred from the arms to the torsion bar.

E

Appendix - Concept Selection and Screening Matrices

Criteria	Reference (ARB with splines)	The Extra strut	The Spring Strut	The Waist	The Extra Arm	The Pencil	The I-Beam	The Y-Change	The Engine	The Planetary gear
		Concepts								
Fit inside available packaging area		-	0	-	-	0	0	-	-	-
Be able to scale down for the front roll bar. Scaleable		0	+	-	0	0	0	+	0	0
Compatible with several chassi variants	2	0	0	0	-	0	0	-	-	-
Not impact the overall safety of the truck. Failsafe	0	0	0	0	0	0	0	0	-	0
95 % of the ARB's should last longer than lifetime of a truck	2	+	+	0	+	-	0	0	-	-
Should be able to handle single loads of 110 kN without plastic deformation or buckling	0	0	0	0	-	0	-	0	0	0
Stiffnes equals to 1250 N/mm +-10%, in high setting	-	?	?	?	?	?	?	?	?	?
Stiffnes equals to 800 N/mm +-10%, in low setting	0	?	?	?	?	?	?	?	?	?
Be able to change stepless between different stiffnesses	3	0	0	+	+	0	0	+	+	0
Be able to change stiffness at any time, not just at 0 deg roll	-	+	+	+	+	+	+	+	+	+
Withstand the environment under a truck	0	+	+	-	0	-	+	0	0	0
Lowest cost	4	0	0	-	-	0	0	0	-	-
Lowest weight		0	0	0	-	+	+	0	-	-
Alter existing layout as little as possible		0	-	-	-	0	0	-	-	-
	Total +	3	4	2	3	2	3	3	2	1
	Total -	1	1	5	6	2	1	3	7	6
	Total 0	8	7	5	3	8	8	6	3	5
	Net	2	3	-3	-3	0	2	0	-5	-5
	Rank	2	1	6	6	4	2	4	8	8
	Further Deveopment	Yes	Yes	Yes	Yes	Yes	Yes	Yes	No	No

Figure E.1: Pugh Matrix - Iteration 1

Criteria	Reference (The spring strut)	Concepts					
		The Extra strut	The Waist	The Extra Arm	The Pencil	The I-Beam	The Y-Change
Fit inside available packaging area	2020-03-05	0	-	-	+	0	-
Be able to scale down for the front roll bar. Scalable		?	?	?	?	?	?
Compatible with several chassi variants: G2 / GR 2.7		0	0	-	-	0	-
Not impact the overall safety of the truck. Failsafe		0	0	0	0	0	0
95 % of the ARB's should last longer than lifetime of a truck		+	-	+	-	0	0
Should be able to handle single loads of 110 kN without plastic deformation or buckling		0	0	0	0	-	0
Stiffnes equals to 1250 N/mm +-10%, in high setting		?	?	?	?	?	?
Stiffnes equals to 800 N/mm +-10%, in low setting		?	?	?	?	?	?
Be able to change stepless between different stiffnesses		0	+	0	+	+	+
Be able to change stiffness at any time, not just at 0 deg roll		0	0	0	+	0	0
Withstand the environment under a truck		0	-	-	-	0	0
Lowest cost		0	0	0	0	0	0
Lowest weight		0	0	0	0	+	0
Alter existing layout as little as possible		-	0	0	0	0	-
	Total +	1	1	1	3	2	1
	Total -	1	3	3	3	1	3
	Total 0	9	7	7	5	8	7
	Net	0	-2	-2	0	1	-2
	Rank	2	4	4	2	1	4
	Further Deveopment	Yes	Yes	Yes	Yes	Yes	Yes

Figure E.2: Pugh Matrix - Iteration 2

F

Appendix - Concept FE-Analysis

In the following chapter the results from the FE-analysis of each concept are presented. The concept analysed in ANSA with a beam model displace one figure with displacements and one with stresses. In each figure there is at least one model in the stiff setting and one in the soft setting.

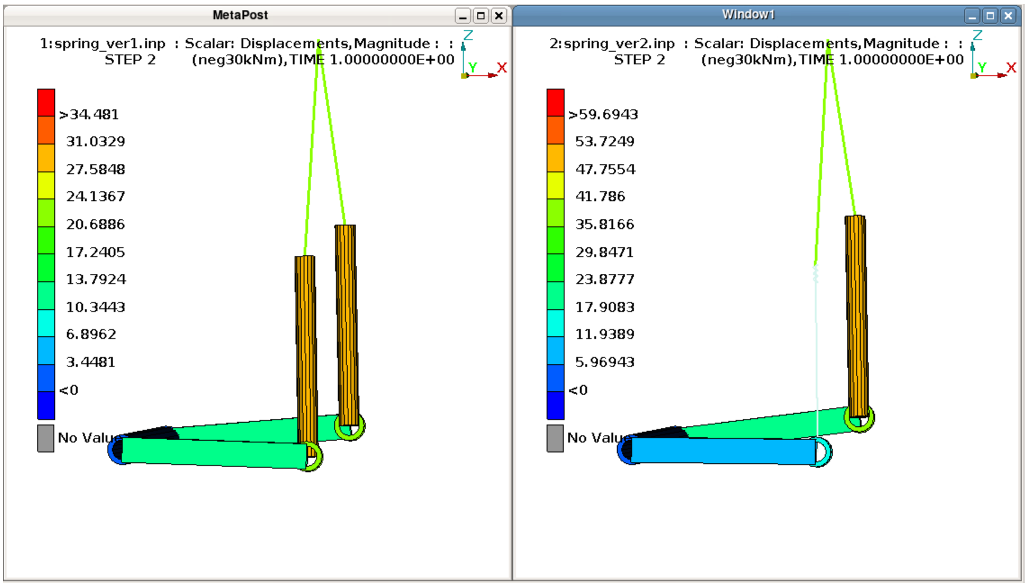

Figure F.1: Spring Strut, displacement

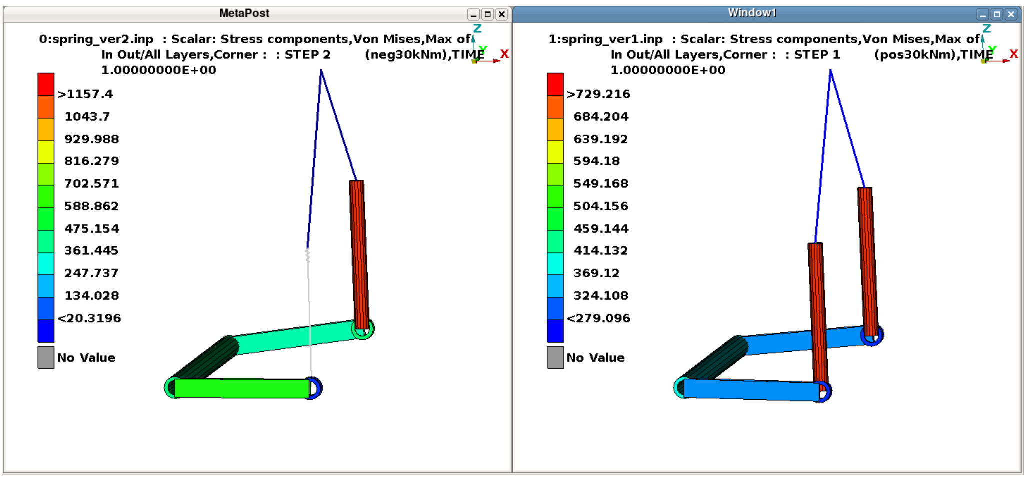

Figure F.2: Spring Strut, stresses

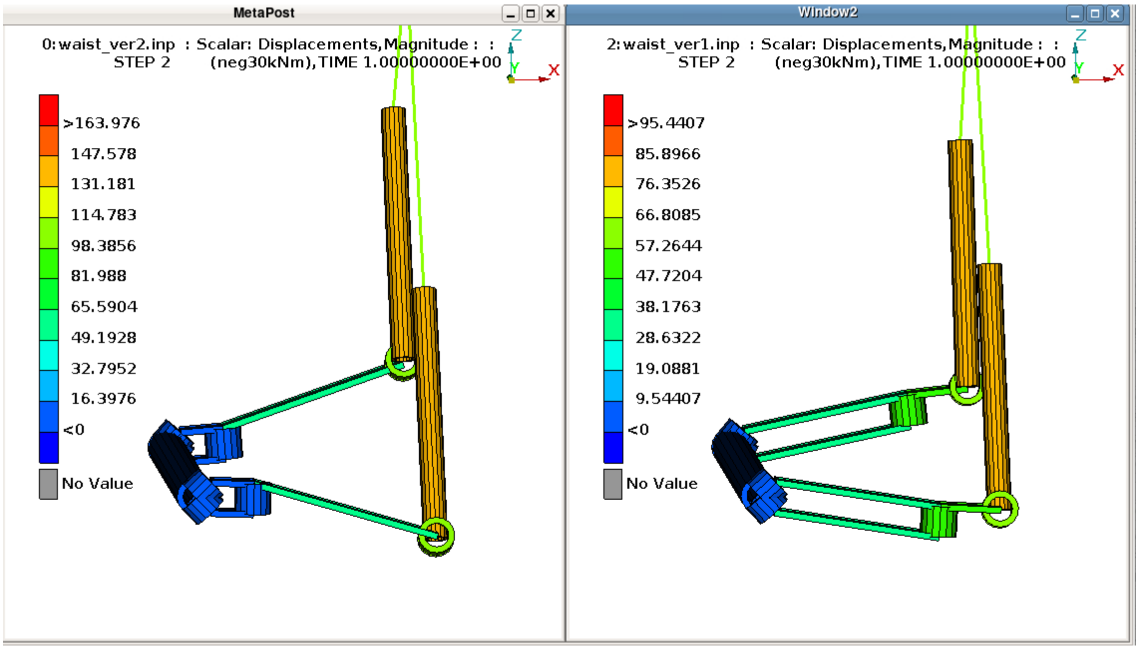

Figure F.3: The Waist, displacement

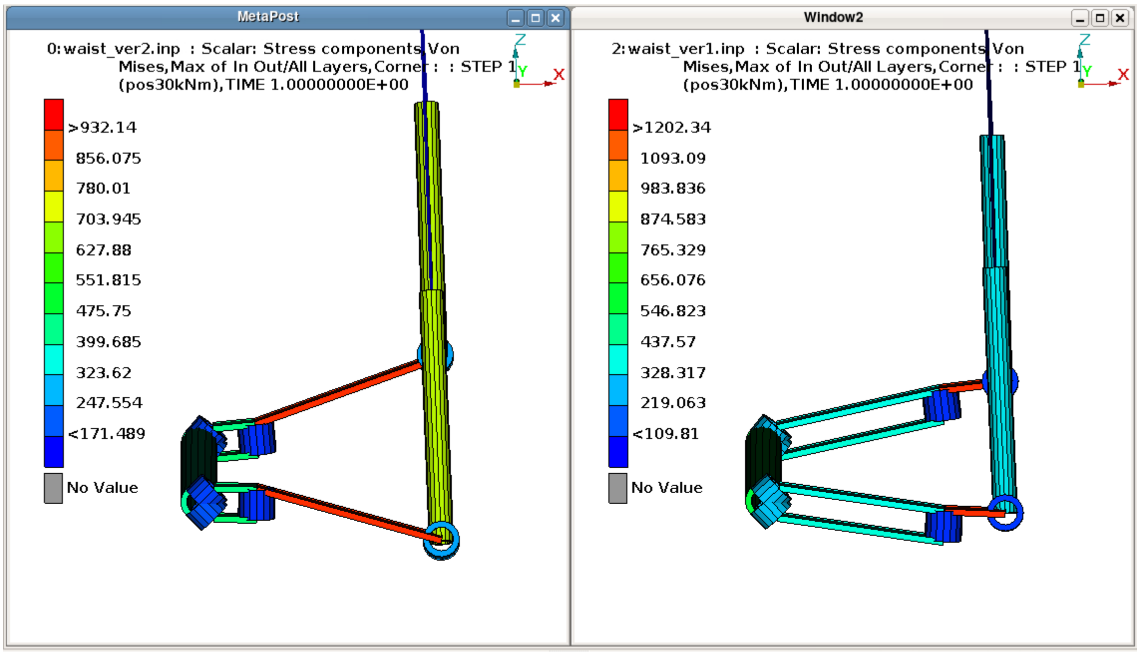

Figure F.4: The Waist, stresses

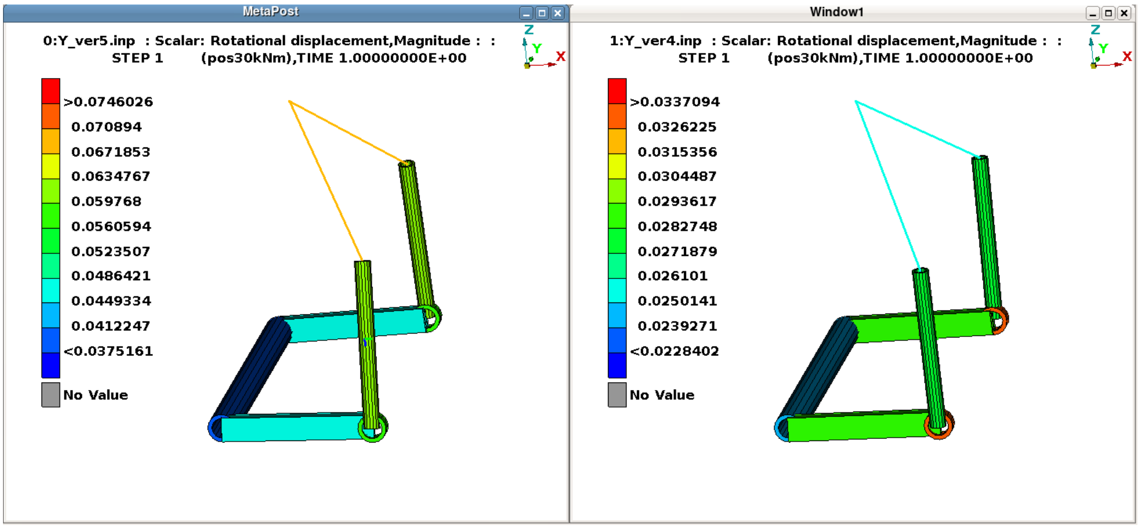

Figure F.5: Y-Change, rotational displacement

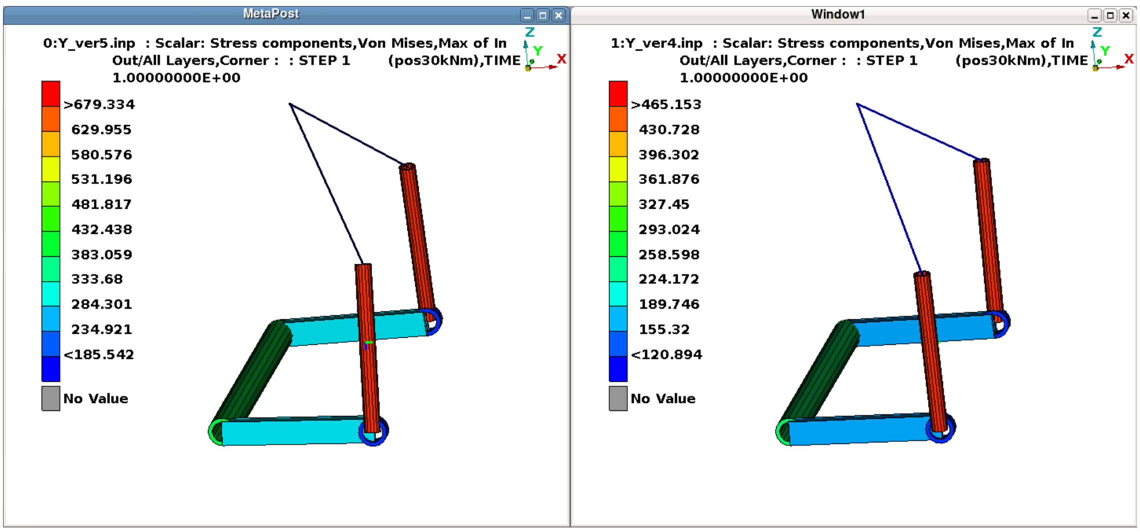

Figure F.6: Y-Change, stresses

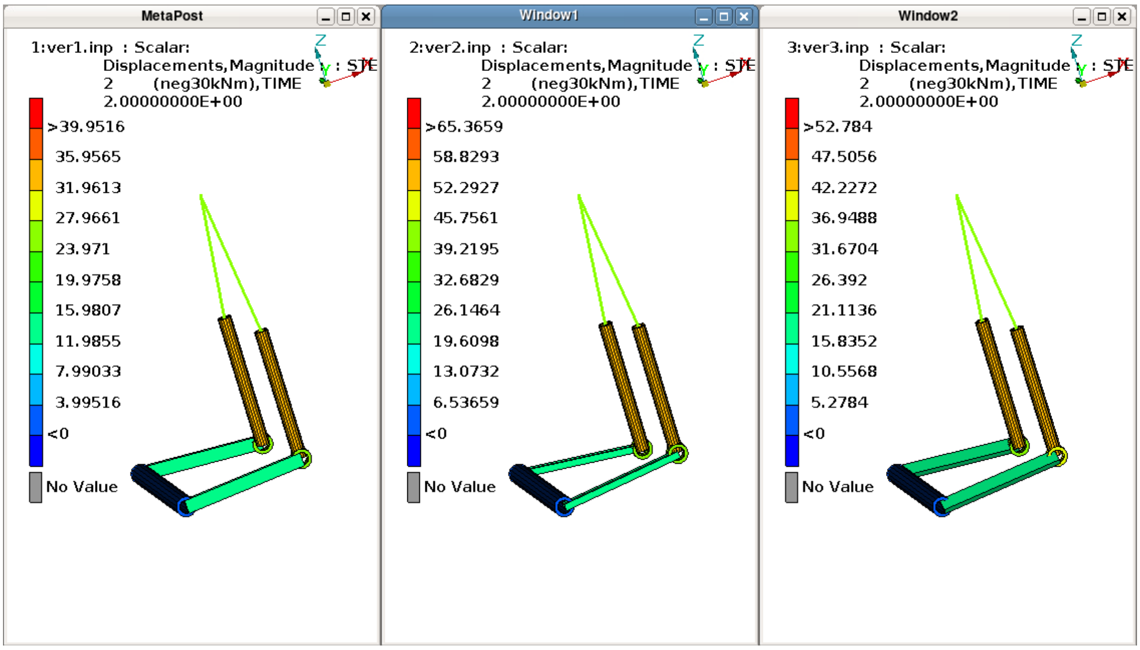

Figure F.7: I-Beam, displacement

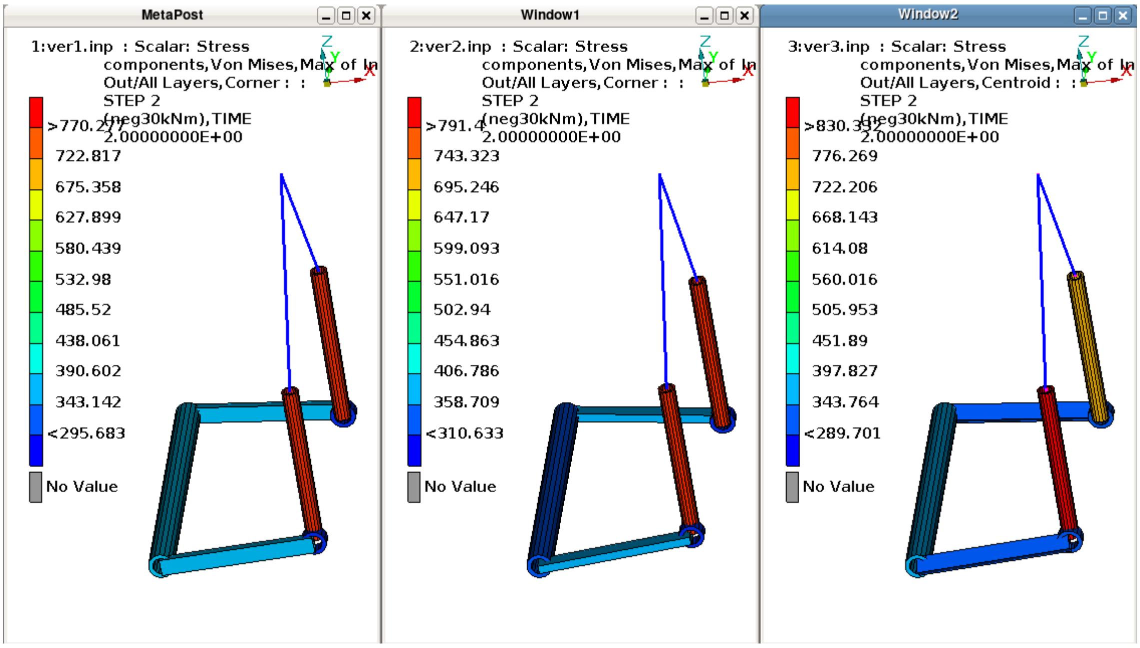

Figure F.8: I-Beam, stresses

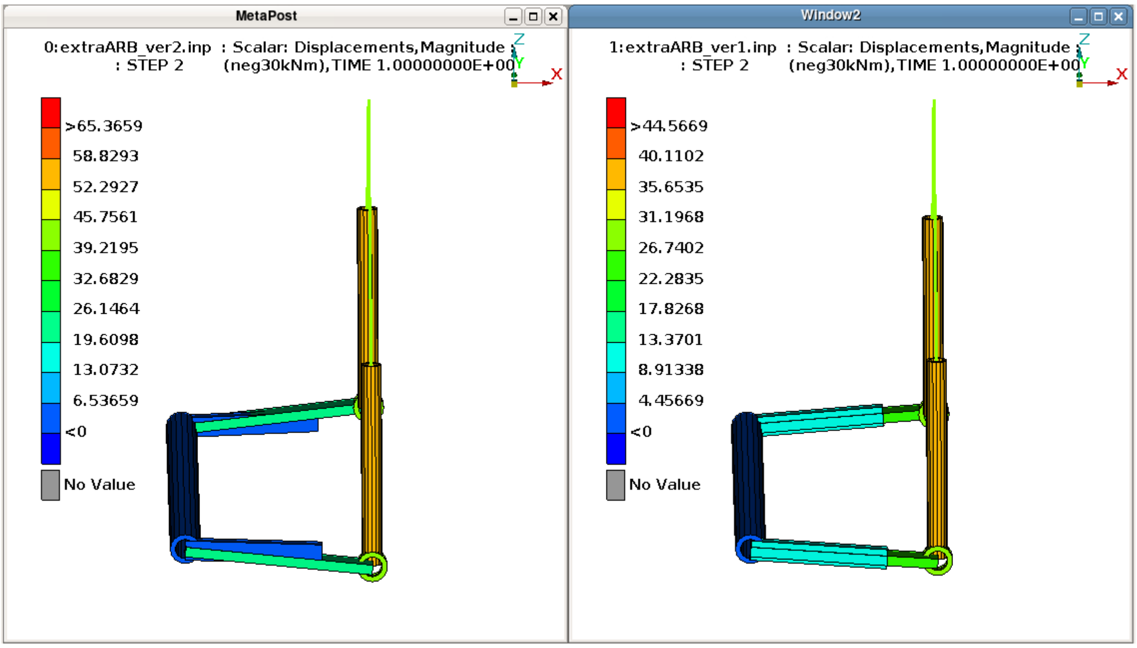

Figure F.9: The Extra arm, displacement

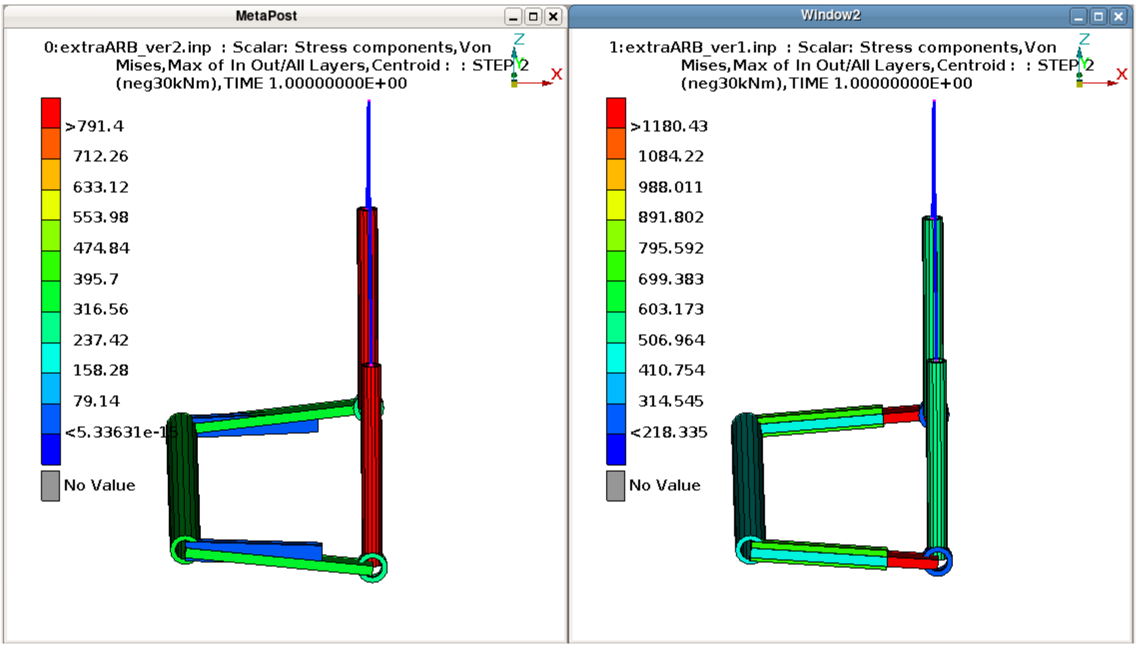

Figure F.10: The Extra arm, stresses

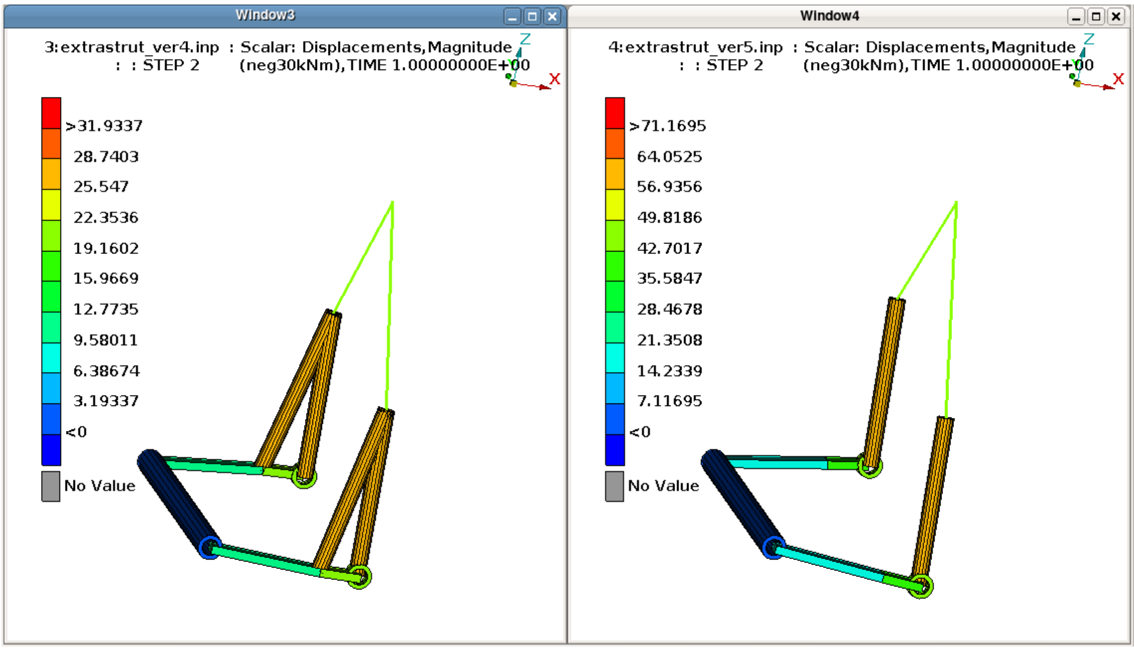

Figure F.11: Extra strut, displacements

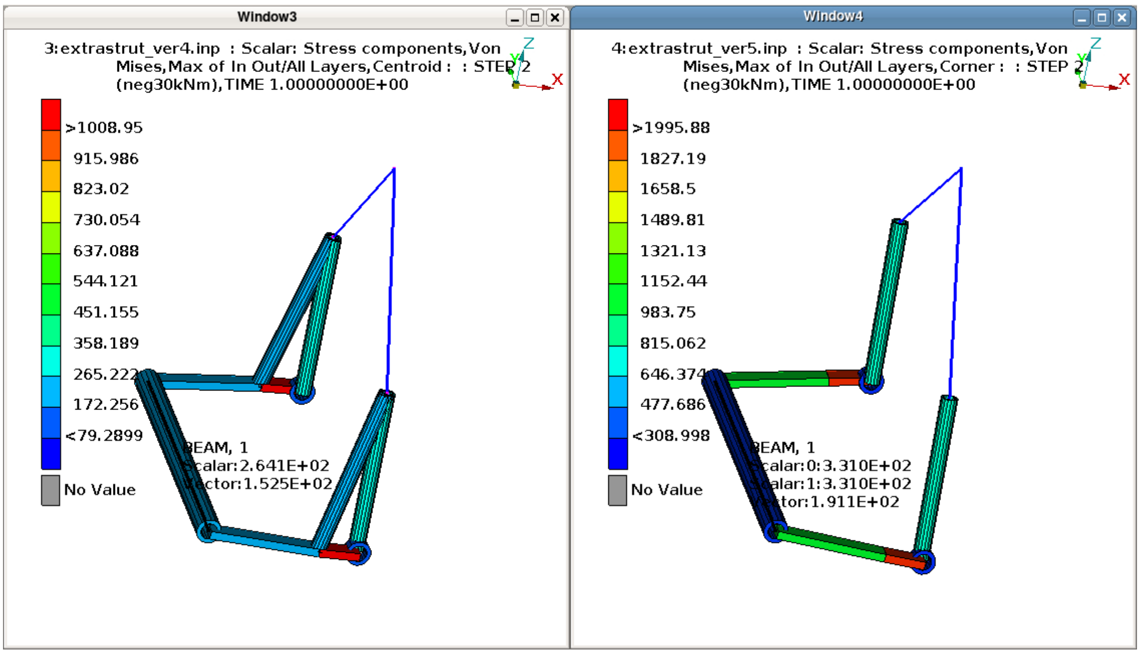

Figure F.12: Extra strut, stresses

Bellow follows the analysis of *the Pencil* concept. Each figure shows both max stresses and max displacements.

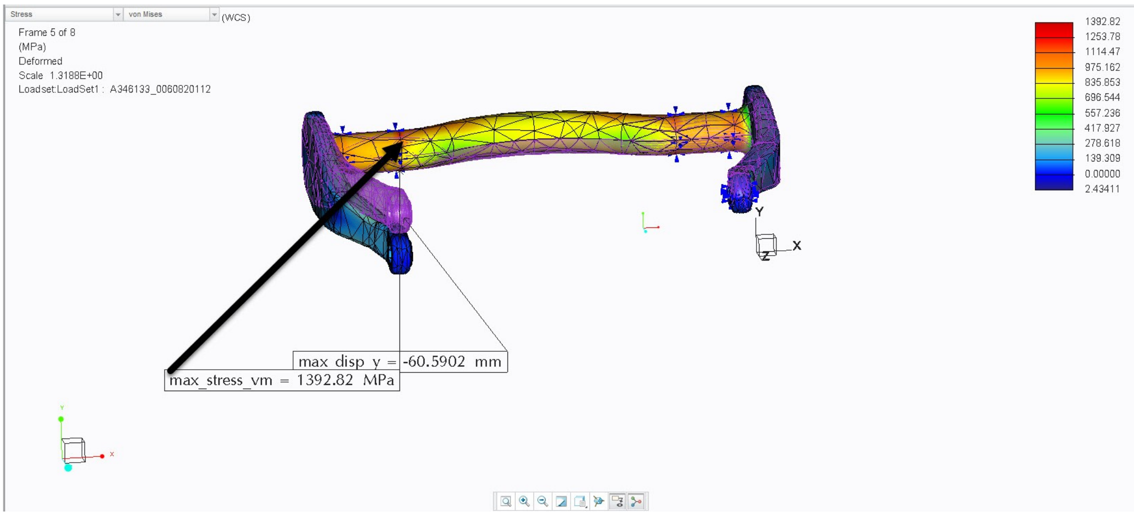

Figure F.13: Pencil concept with 70 mm bar and long arms. Peak stress and max displacement plotted when subjected to 70 kN load.

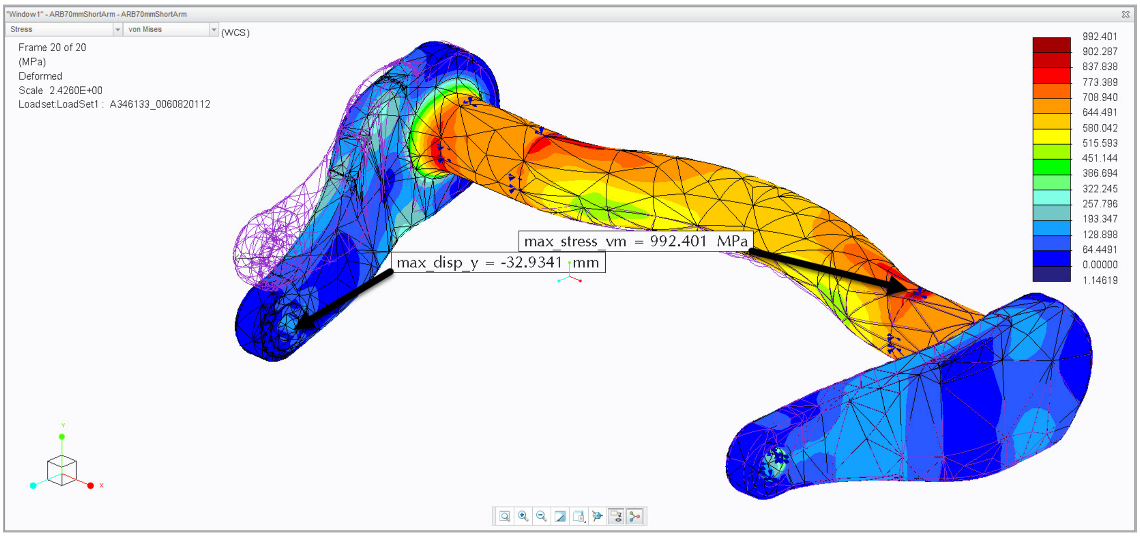

Figure F.14: Pencil concept with 70 mm bar and short arms. Peak stress and max displacement plotted when subjected to 70 kN load.

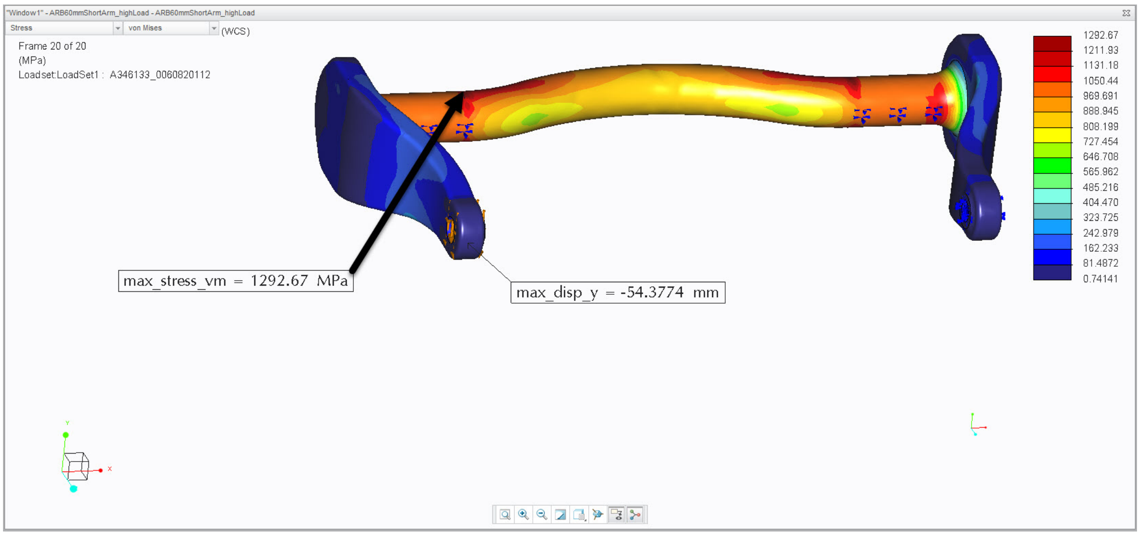

Figure F.15: Pencil concept with 60 mm bar and short arms. Peak stress and max displacement plotted when subjected to 70 kN load.

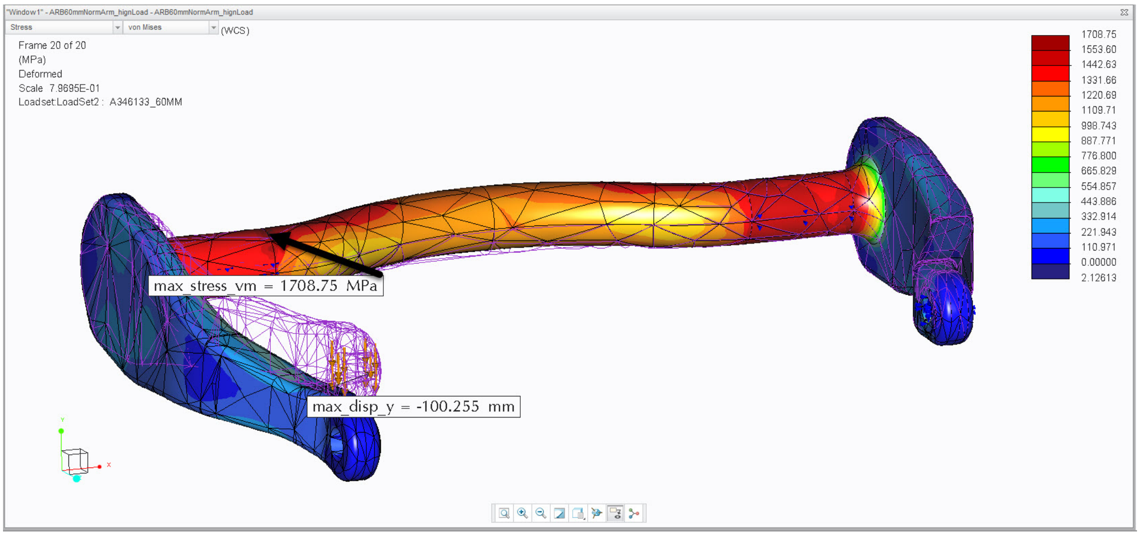

Figure F.16: Pencil concept with 60 mm bar and long arms. Peak stress and max displacement plotted when subjected to 70 kN load.

G

Appendix - Pros and Cons List

Decision	Concept	Pros	Cons	Further Developent	Comments	Packaging
	The Extra strut	No movement ine the ARB arms when changing stiffness.	High stresses in bar when the short arm setting is on. Has no real backup since the two struts cant be active at the same time	Size and dim of the strut?	After analyse - seams possible to dim. Another version of the pencil, therfore its removed	Can be problematic
	The Waist	Stepless stiffness variation	Exposed sliding parts, problematic for the ARB to be stiff enough., No "balkliv" between the beams.,	Protect from the surounding environment?, make a "balkliv"	After analyse - hard to achive the stiffness variation	Good possibilities
	The Extra Arm	No sliding part, just locking. Good for packaging	Activation mechanism, can malfunction or break.	dimensions of extra arms / extra back pipes? Mounting points and bushings with extra pipe? Packaging with p-strut + extra arm coupling points?	Possible to bolt the weak arm to the rigid arm instead of welding?	Good possibilities
	The Pencil	Change of stiffness far from area with high torque. Good packaging	Sliding parts. High stresses in bar	possible to make the arms weaker further out in the ends to reduce the stress in the pipe?		Good possibilities
	The I-Beam	Simple, Existing solution for cars,	Fatigue, bearing loads,	Strong enough? Fatigue in arms soft setting? Bearing radial force? How to mount the bar and the arm?	Evaluate the bearing size, 50 inner and 85 outer diameter can hold up forces - SKF 33110	Problematic
	The Y-Change	Stepless stiffness variation. Little stress variation between soft and stiff setting	Need to integrate the y-translation mechanism in the crossmember - packaging - structural mech.,	Packaging?The strut will move in an arc when changing stiffness?		Can be problematic
	The spring strut	Low number of moving parts, Keep the stiffness variation away from the ARB.	Locking mechanism. Design of leaf spring	Packagimg? Locking mechanism? Design of leaf springs? Fatigue?	Use a solenoid for on/off settings. After analyse - spring need approx 1000 N/mm in stiffness	Good possibilities

Figure G.1: Pros and Cons List

H

Appendix - Analysis of Section Forces

In figure H.1 the high stiffness setting is to the right and the low to the left. As seen in the figure the forces in the strut in the low setting is 82.7 kN and in the high setting it is 137.3. This means that the section force in the strut is reduced to 60 % when the upper part of the strut moves 100 mm towards the center of the truck.

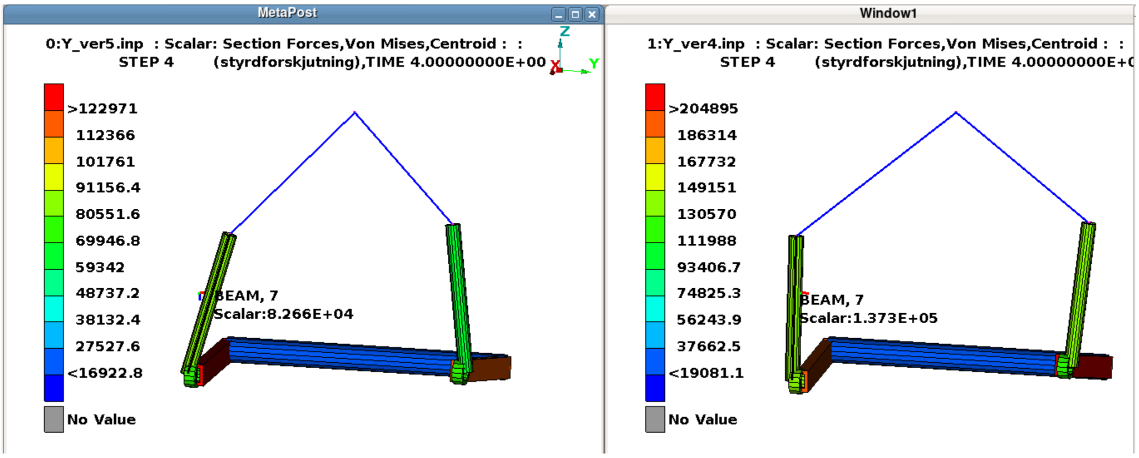

Figure H.1: Section forces in the Y-Change concept