Bachelor’s thesis in

Weight Reduction of Reach Stacker

An investigation of reducing Eigen weight through design

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Summary

This Bachelor’s thesis treats a product development project at Konecranes Lifttrucks AB, Markaryd, Sweden. The thesis is made by a student within the program of Mechanical Engineering at Linnaeus University in Växjö.

The company manufactures heavy duty trucks with variable reach, called reach stackers. In order to meet the future’s need of more environmental-friendly solutions an approach is to reduce the Eigen weight of vehicles. The subject for the weight reduction is in this case the telescopic boom of the reach stacker.

The main goal of the project is to provide the company with an appropriate investigation regarding a possible new design of their product reach stacker in order to improve the machine’s efficiency. Through the methods of product development a specification of requirements is derived which leads to a final concept. To obtain the wanted results while maintaining strength structural steel of high strength has been determined to serve with good results.

The final outcome of the project is a concept where the structural steel is of designation S690QL. With support of calculations and CAD simulations the possible weight reduction is assumed to amount to 28%.
Abstract

The report is about reducing the Eigen weight of a reach stacker in order to obtain decreased fuel consumption. The detail the product development treats is the telescopic boom. By using steel of higher strength the dimension can be decreased, which in turn results in a reduced weight. Suitable steel for the application might be the high strength structural steel of designation S690QL. With support of calculations and CAD simulations the possible weight reduction is assumed to amount to 28%.

Keywords

Weight reduction; reach stacker; telescopic boom; product development; System’s Engineering; QFD; FMEA; SolidWorks.
Acknowledgement

This Bachelor’s thesis is the final and examining work of the program of Mechanical Engineering at Linnaeus University in Växjö and is made by the student Linda Ekdahl Norling during the spring semester of 2014. The thesis treats a product development project at the company Konecranes Lifttrucks AB in Markaryd, Sweden.

Throughout the process several people have been involved through consulting and tutoring. I would like to express my gratitude to all of you who participated with your knowledge and time in enabling this thesis. I want to point special thanks to the sponsor company for this thesis, Konecranes Lifttrucks AB, their technical director Anders Nilsson who has been supervising the project in a supportive way, test driver Urban Linder and Design Manager Roger Persson. Their knowledge has been fundamental for the project.

I also would like to thank supervisor Lars Ericson, examiner Samir Khoshaba and Izudin Dugic at Linnaeus University for theoretical assistance during the process of this Bachelor’s thesis.

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Linda Ekdahl Norling

Visseltofta, 11th of June 2014
# Table of Contents

## 1 Introduction
- 1.1 Background
- 1.1.1 Konecranes Lifttrucks AB
- 1.2 Objectives
- 1.3 Requirements
- 1.4 Limitations
- 1.5 Problem

## 2 Theory
- 2.1 Product Development Tools
  - 2.1.1 Product Development through System’s Engineering
- 2.1.2 Quality Function Deployment
- 2.1.3 Failure Mode and Effects Analysis
- 2.2 Material Properties
- 2.3 Design
- 2.4 Possible Materials
  - 2.4.1 Steel
  - 2.4.2 Composite
  - 2.4.3 Aluminum

## 3 Method
- 3.1 Choice of Method
- 3.2 Project Process
- 3.3 Specification of Requirements
- 3.4 Visits
  - 3.4.1 Konecranes
  - 3.4.2 AB Bröderna Jansson – Nissavarvet
- 3.5 Expertise
  - 3.5.1 Konecranes
  - 3.5.2 Sapa AB
  - 3.5.3 Marstrom Composite
  - 3.5.4 SSAB
  - 3.5.5 Ruukki
- 3.6 SolidWorks

## 4 Empirical Findings
- 4.1 Konecranes Reach Stacker

## 5 Current State Analysis
- 5.1 Defining the Project
  - 5.1.1 Selecting Project
  - 5.1.2 Pictures of the Current State
  - 5.1.3 Defining and Tailoring Concept
<table>
<thead>
<tr>
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<td>XXVI</td>
<td>XXVII</td>
</tr>
</tbody>
</table>
Figure 1. Konecranes Reach Stacker SMV 4531 TB5 ......................................................... 5
Figure 2. House of Quality Matrix .............................................................................. 9
Figure 3. Liebherr Reach Stacker ................................................................................. 10
Figure 4. The Little Giant 6430 Carrier Mounted Crane ........................................... 11
Figure 5. Mobile Crane; Optim QC Ultra-high-strength steel, Ruukki ..................... 13
Figure 6. Steel Structure External Boom .................................................................... 19
Figure 7. Chassi ........................................................................................................... 22
Figure 8. Profile of Boom ............................................................................................... 23
Figure 9. Rear End Boom ............................................................................................... 24
Figure 10. Cabin Display ................................................................................................. 25
Figure 11. Spreader Camera ......................................................................................... 25
Figure 12. Counterweight ............................................................................................... 26
Figure 13. Cabin ............................................................................................................. 27
Figure 14. Locking Device Spreader .......................................................................... 28
Figure 15. Reach Stacker in Operation 1 ..................................................................... 29
Figure 16. Reach Stacker in Operation 2 ..................................................................... 30
Figure 17. Reach Stacker ............................................................................................... 31
Figure 18. Telescopic Boom in Full Extension ............................................................. 32
Figure 19. Mobile Crane; Weldox, SSAB .................................................................. 52
Figure 20. Current State Boom; von Mises Stress ...................................................... 55
Figure 21. Current State Boom; Safety Factor .............................................................. 56
Figure 22. High Strength Concept; von Mises Stress ..................................................... 56
Figure 23. High Strength Concept ................................................................................ 57
Figure 24. Ultra-high Strength Steel 1; von Mises Stress ............................................ 57
Figure 25. Ultra-high Strength Steel 1; Safety Factor .................................................... 58
Figure 26. Ultra-high Strength Steel 2; von Mises Stress ............................................ 58
Figure 27. Ultra-high Strength Steel 2; Safety Factor .................................................... 59
Figure 28. Ultra-high Strength Steel 3; von Mises Stress ............................................ 59
Figure 29. Ultra-high Strength Steel 3; Safety Factor .................................................... 60
Figure 30. Close-up U- shape ....................................................................................... 60

Table 1. Basic design-limiting material properties, symbols and units....................... 6
Table 2. Properties S500Q, S690QL, S890QL ................................................................. 13
Table 3. Technical Data ................................................................................................. 33
Table 4. Service Interval ............................................................................................... 34
Table 5. Voice of the Customer .................................................................................... 38
Table 6. Product Objectives ......................................................................................... 39
Table 7. Use Cases ........................................................................................................ 41
Table 8. Use Cases Behaviour 1 .................................................................................. 42
Table 9. Use Cases Behaviour 2 .................................................................................. 43
Table 10. Functional Requirements ............................................................................. 44
Table 11. Final Requirements ...................................................................................... 44
Table 12. Ranked Product Objectives ......................................................................... 49
Table 13. Possible Failures ............................................................................................ 54
Table 14. Concept Comparison ..................................................................................... 61
Table 15. Accumulated Risk Priority .......................................................................... 61
Table 16. Concept 1 Figures ......................................................................................... 62
Table 17. Decreased Material Thickness ...................................................................... 62
Table 18. External Boom Concept Weight ................................................................... 63
Table 19. Heavy-Duty Vehicle Chassis Metrics and Targets ........................................ 65
Abbreviations
AB Aktiebolag (Incorporated Company)
AHSS Advanced High Strength Steel
CAD Computer-Aided Design
ECTS European Credit Transfer System
EIA Environmental Impact Assessment
EN European Standard
FMEA Failure Mode and Effects Analysis
FAT Fatigue classification
G Giga ($10^9$)
HDV Heavy Duty Vehicle
HoQ House of Quality
IIW International Institute of Welding
ISO International Organization for Standardization
J Joule
k Kilo ($10^3$)
LCA Life Cycle Assessment
LCC Life Cycle Cost
m Meter
m Milli ($10^{-3}$)
M Mega ($10^6$)
N Newton
QFD Quality Function Deployment
QL/Q/QC Quenched and tempered
Pa Pascal (N/mm²)
RPN Risk Priority Number
SF Safety Factor
SMV Silverdalens Mekaniska Verkstad
Sol-gel Solution (conversion of monomers), gel (particles or polymers, for example metal alkoxides)
SS Swedish Standard
VMT Vehicle Miles Travelled
W Watt
μ Mikro ($10^{-6}$)
1 Introduction

In this chapter the basic premises of the project is presented.

1.1 Background

The Bachelor’s thesis is the final work within the program of Mechanical Engineering at Linnaeus University in Växjö. The project is made by a student or a group of students in cooperation with a sponsor company. The sponsor company of this project is Konecranes Lifttrucks AB in Markaryd, Sweden, which henceforth is referred to as Konecranes. The company is presented in paragraph 1.1.1.

Konecranes manufactures heavy duty trucks and trucks with variable reach called reach stackers. In order to meet the future’s need of more environmental-friendly products Konecranes is searching for more efficient solutions regarding life cycle cost as well as life cycle assessment. More specifically such improvements should result in reduced operating costs concerning fuel consumption and through the entire life cycle a reduced impact of the environment.

1.1.1 Konecranes Lifttrucks AB

Konecranes is a group of Lifting Businesses™, which consists of both lifting equipment and services for a broad range of customers. Konecranes Lifttrucks belongs to Business Area Equipment. The customers comprise of manufacturing and process industries, shipyards, ports and terminals. Through its global service network, Konecranes’ Business Area Service offers a full range of service solutions, specialized maintenance and modernization services for all types of industrial cranes, port equipment, and machine tools. Konecranes’ Business Area Equipment offers components, cranes and material handling solutions for wide range of industries, including process industries, the nuclear sector, industries handling heavy loads, ports, intermodal terminals, shipyards and bulk material terminals.

The concern has totally 12100 employees at 626 locations in 48 countries all over the world. In 2012, the Konecranes Group sales totaled EUR 2170,2 million. Konecranes is listed on the NASDAQ OMX Helsinki. In 2004 Konecranes acquired SMV (Silverdalens Mekaniska Verkstad) in Markaryd. In appendix G a history of Konecranes Lift Trucks, former SMV, can be found.
1.2 Objectives

The purpose of the Bachelor’s thesis is for the student to use his or her obtained knowledge from the education and through that be able to assist a sponsor company with a relevant study. The study should be presented in a written report where verified facts and evidence are the basis for the conclusion.

The main goal of the project is to provide Konecranes with the appropriate investigation regarding a possible new design of their product reach stacker in order to improve the machine’s efficiency. The project will be presented as a concept proposal through an oral presentation and a written report.

1.3 Requirements

A Bachelor’s thesis requires an accurately reported investigation presented in a written project report within the confines of the Mechanical Engineering program at the Bachelor’s degree. The conclusion must be based on a well-researched study where the thesis must be substantiated by scientific articles, calculations of strength and sector knowledge.

The company Konecranes requires that a disclosure agreement is established between Konecranes and the student where the student agrees not to reveal any confidential information.

1.4 Limitations

The thesis is made within the scope of the Mechanical Engineering program. The limit in time for the Bachelor’s thesis is ten weeks of full time studies which is equal to 15 ECTS (credit points). This thesis is the work of one student.

In the study many factors determine how the final conceptual design turns out. The constraints of time lead to the fact that an approach that yields the widest possible dividends of the work should be designated as the starting point of the project.
When considering the total life cycle of a machine many factors must be weighted. The selection of construction material plays a major role where the entire life cycle of the material must be investigated from manufacturing to destruction and recycling. Transportation of material and details to the final assembling at Konecranes in Markaryd affects both the life cycle cost and the impact of the environment. Different consumable items such as hydraulic oil, seals and gaskets must also be considered. All the factors and aspects must be weighted against the economic requirements for investments, both regarding Konecranes own production and what the end costumer is willing to pay for the product. An adequate life cycle analysis requires a holistic perspective that cannot be investigated to a relevant extent within the framework of a Bachelor’s thesis.

Hence the design problem should be viewed from a perspective where the factors under investigation have an unquestionable impact on the machine’s life cycle, but which run a low risk to cause unexpected negative consequences.

The primary effects of a weight reduction of the reach stacker can be derived to a decreased operational cost due to reduced fuel consumption. Other possible primary effects are decreased material consumption and cheaper purchase price. A weight reduction also leaves room for a range of secondary effects which also have a positive effect on the total life cycle. A lighter reach stacker and in particular lighter booms rise for downsizing the driving components in terms of dimensions. These components comprise the hydraulic system that operates the booms, the driveline, pumps and all belonging gears. This so called spin-off effect is not further treated and charted in this thesis. Still, it is notable to be aware of it and motivates a weight reduction as the starting point of this project.

1.5 Problem

The current design of the reach stacker restricts the various possibilities to achieve the desired results in terms of weight reduction. This is due to the design of the reach stacker where the principle of counterweight is implemented to lift objects. The system is based on the momentum the payload is creating over the front axle which is counteracted by a counterweight. Because of this, the weight behind the front axle is counterweight which makes little sense to consider until the weight in front of the front
axle is decreased. This leaves the boom and the lifting device called spreader, as most interesting alternatives to consider. The spreader is designed and manufactured by another company, Elme Spreader AB, which limits possible design choices to only deal with the boom.

Konecranes’ reach stacker structural concept is based on a platform from 1994. The company is now considering different ways to develop the machine in order to achieve an increased efficiency in terms of fuel consumption, total life cycle cost and total impact of the environment by reducing its structural weight.

The total weight is proportional to the fuel consumption of the machine which is an important factor in the pursuit of a more energy efficient machine, both regarding the environment and the economy.

In order to concretize the problem the starting point of the project will be to evaluate whether a possible change of material can reduce the weight of the boom and in this way obtain the desired resulting effect. The weight of the boom is central in the project because of the principle of weight and counterweight which the reach stackers are based upon.

When a conclusion regarding the material selection can be made the project will continue with investigating the consequences that are supposedly positive. Furthermore a concept design will be calculated and drawn in SolidWorks. The problem formulation of this thesis can be derived to following issues:

- Is a weight reduction motivated?
- If a motivation for weight reduction is ensured, can the evidence form basis for a relevant concept?
Figure 1. Konecranes Reach Stacker SMV 4531 TB5
2 Theory

_The theory presented in this chapter constitutes the basis of the thesis._

2.1 Product Development Tools

This thesis is categorized as a product development project and will be treated with those in the field of engineering recognized tools of product development.

2.1.1 Product Development through System’s Engineering

System’s Engineering is an approach used when developing a product. The technique is based upon a systematic investigation of every aspect regarding the product in question. This project will to a certain extent be carried out through the systematic approach explained in the book Getting Design Right; a Systems Approach by Peter Jackson¹.

In the product development process materials whose property profile is consistent with the wishes and demands that are put on the finished product are scanned. Basic design-limiting material properties and their symbols are shown in the table below².

<table>
<thead>
<tr>
<th>Class</th>
<th>Property</th>
<th>Symbol</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>General</td>
<td>Density</td>
<td>( \rho )</td>
<td>(kg/m³ or Mg/m³)</td>
</tr>
<tr>
<td></td>
<td>Price</td>
<td>( C_m )</td>
<td>(€/kg)</td>
</tr>
<tr>
<td>Elastic Moduli (Young’s, shear, bulk)</td>
<td>( E, G, K )</td>
<td>(GPa)</td>
<td></td>
</tr>
<tr>
<td>Yield strength</td>
<td>( \sigma_y ) or ( S_y )</td>
<td>(MPa)</td>
<td></td>
</tr>
<tr>
<td>Ultimate strength</td>
<td>( \sigma_u ) or ( S_u )</td>
<td>(MPa)</td>
<td></td>
</tr>
<tr>
<td>Compressive strength</td>
<td>( \sigma_c ) or ( S_c )</td>
<td>(MPa)</td>
<td></td>
</tr>
<tr>
<td>Failure strength</td>
<td>( \sigma_f ) or ( S_f )</td>
<td>(MPa)</td>
<td></td>
</tr>
<tr>
<td>Hardness</td>
<td>( H )</td>
<td>(Vickers)</td>
<td></td>
</tr>
<tr>
<td>Elongation</td>
<td>( \varepsilon )</td>
<td>(-)</td>
<td></td>
</tr>
<tr>
<td>Fatigue endurance limit</td>
<td>( \sigma_e ) or ( S_e )</td>
<td>(MPa)</td>
<td></td>
</tr>
<tr>
<td>Fracture toughness</td>
<td>( K_{IC} )</td>
<td>(MPa*( m^{1/2} ))</td>
<td></td>
</tr>
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</table>

¹ (Jackson, 2010)
² (Ashby, 2005)
<table>
<thead>
<tr>
<th>Property</th>
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<tbody>
<tr>
<td>Toughness</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Loss coefficient (damping capacity)</td>
<td>$\eta$</td>
<td>(-)</td>
</tr>
<tr>
<td><strong>Thermal</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Melting point</td>
<td>$T_m$</td>
<td>(C or K)</td>
</tr>
<tr>
<td>Glass temperature</td>
<td>$T_g$</td>
<td>(C or K)</td>
</tr>
<tr>
<td>Maximum service temperature</td>
<td>$T_{max}$</td>
<td>(C or K)</td>
</tr>
<tr>
<td>Minimum service temperature</td>
<td>$T_{min}$</td>
<td>(C or K)</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$\lambda$</td>
<td>(W/m*K)</td>
</tr>
<tr>
<td>Specific heat</td>
<td>$C_p$</td>
<td>(J/kg*K)</td>
</tr>
<tr>
<td>Thermal expansion coefficient</td>
<td>$\alpha$</td>
<td>(K$^{-1}$)</td>
</tr>
<tr>
<td>Thermal shock resistance</td>
<td>$\Delta T_s$</td>
<td>(C or K)</td>
</tr>
<tr>
<td><strong>Electrical</strong></td>
<td></td>
<td></td>
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<tr>
<td>Electrical resistivity</td>
<td>$\rho_e$</td>
<td>($\Omega<em>m$ or $\mu\Omega</em>cm$)</td>
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<td>Dielectric constant</td>
<td>$\varepsilon_d$</td>
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<td>Breakdown potential</td>
<td>$V_b$</td>
<td>($10^6$ V/m)</td>
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<tr>
<td>Power factor</td>
<td>$P$</td>
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<tr>
<td><strong>Optical</strong></td>
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<tr>
<td>Optical, transparent, translucent, opaque</td>
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<td></td>
</tr>
<tr>
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<td>$n$</td>
<td>(-)</td>
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<tr>
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<td>Energy/kg to extract material</td>
<td>$E_r$</td>
<td>(MJ/kg)</td>
</tr>
<tr>
<td>CO$_2$/kg to extract material</td>
<td>CO$_2$</td>
<td>(kg/kg)</td>
</tr>
<tr>
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<tr>
<td>Oxidation rates</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Corrosion rates</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wear rate constant</td>
<td>$K_A$</td>
<td>(MPa$^{-1}$)</td>
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2.1.2 Quality Function Deployment

The purpose of Quality Function Deployment, QFD, is to plan the product development customer-centred with a systematic and structured approach\(^3\). A definition\(^4\) of the QFD reads as follows.

“A system to translate the customer’s requests to relevant specifications in every step of the product development process, from market to development, production and sale and service.”

In concrete terms the QFD approach in this project will include a summary of customer interviews, Voice of Customer\(^5\), and a House of Quality matrix. The main focus of this project is the development and production.

A central part of product development is to ensure that the customer’s needs and requests are met. To make sure this is achieved a summary of the customer interviews, called Voice of Customer, is done.

The House of Quality\(^6\) is a matrix chart where customer’s requests and product properties are described. The matrix enables grading of relations between the different groups. The fields of the matrix are illustrated in figure 2.

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\(^3\) (Bergman & Klefsjö, 2012)  
\(^4\) (Slabey, 1990)  
\(^5\) (Jackson, 2010)  
\(^6\) (Bergman & Klefsjö, 2012)
2.1.3 Failure Mode and Effects Analysis

Failure Mode and Effects Analysis\(^7\), FMEA, is implemented as a qualitative analysis of the different concepts putative failure modes and their correlating failure consequences. The FMEA is a helpful tool in the strive for the final proposed concept.

2.2 Material Properties

The amount of possible material choices are limited by a number of factors. Cost represents a significant such concerning total life cycle cost; extraction, manufacturing, production, operational, maintenance and destruction. A factor specially considered in this project is sustainability referring to the materials ability to be more environmental friendly than the present choice of material. Further aspects that determine for the project suitable construction materials are availability, machinability, maintainability and repair ability. These aspects are closely linked to the first mentioned factor; cost.

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\(^7\) (Bergman & Klefsjö, 2012)
2.3 Design

The choice of material cannot be treated separately from the choice of design. Similar applications and solutions of the problem can give valuable input and guidelines for the later steps of the product development where concepts are generated.

Among reach stacker boom designs many of them are similar to Konecranes boom design. The competing reach stacker manufacturer Liebherr stands out a bit with their curved telescopic booms. The advantage is that the reach capacity is increased due to the possibility of reaching a third row of containers as figure 3 shows. The telescopic boom is operated by one hydraulic cylinder instead of two on each side. Liebherr states\(^8\), however, not the benefits with the mentioned design that are sought in this project.

![Figure 3. Liebherr Reach Stacker](image)

In similar sectors different solutions can be found. Mobile cranes for the construction industry are constructed to endure exceptionally heavy loads and to be able to handle

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\(^8\) (Liebherr, 2014)
the loads at heights that demands telescopic booms with several sections. To sustain this booms often have a U-shaped design and rounded corners.

**Figure 1. Mobile Crane**

Another solution is booms made of a structure of latticework, see figure 4. The principle is interesting but difficult to apply directly on a telescopic boom. It raises, however, notions about the stress distribution in a boom. The stress concentrations in a boom are distributed along force arrows. If the stresses are mapped it will show areas where the stress becomes insignificant. In these areas, material may be removed without the boom’s strength is negatively affected. The holes appearing where the material is removed should be round or oval to avoid sharp edges where fatigue cracks can occur.

**Figure 4. The Little Giant 6430 Carrier Mounted Crane**
2.4 Possible Materials

In order to roughly limit the number of applicable material choices the industrial sector and similar sectors has been investigated. Materials appropriate for consideration are hereby described.

2.4.1 Steel

Steel is by far the most time-tested material in the heavy duty automotive industry. With that come a number of advantages such as a well-developed sub-sector, well-known by customers, good basic knowledge, a well proven and a refined maintenance sector. Another major advantage of steel is that it is completely recyclable which is well compatible with the objective of eco-efficiency. With this as background steel by means of construction material can mean a cheaper product development process due to a minimal investigation about material selection. However, this approach is at high risk of bringing disadvantages due to that the material choice is not optimal for its purpose. The disadvantages can be such as unnecessarily large dimensions causing needlessly heavy parts and a greater material consumption. It can also mean that properties like corrosion resistance, is overlooked. The consequences are not only a perceived higher cost but also an increased impact of the environment. Suitable structural steels are further presented in coming paragraphs. All steel designations are given in European Standard, EN.

According to European Standard tempered plates up to 70 millimeter thick are included in the structural steels. This plate is designed for welded constructions and offers higher strength than non-treated or normalized plates.

Tempered steel has a decent toughness and good weld ability although it should not be welded with high energy methods like electro slag welding. That might cause an unnecessarily wide soft zone along the welded joint.

The current material for the boom is structural steel with the designation S500Q which is tempered steel categorized within the group of high strength steel. SSAB is one of

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9 (Metallnormcentralen, 1988)  
10 (Swedish Standards Institute, 2014)
Konecranes’ suppliers of steel sheets. In their Weldox series\textsuperscript{11} they provide steel of designation S690QL (Weldox 700) and S890QL (Weldox 900). These materials can be proper for further studies in this project\textsuperscript{12}.

Table 2. Properties S500Q, S690QL, S890QL

<table>
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<th>Properties; S500Q</th>
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</tr>
<tr>
<td>Min</td>
</tr>
<tr>
<td>3</td>
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</table>

\textbf{S690QL}

<table>
<thead>
<tr>
<th>Dimensions (mm)</th>
<th>Yield Strength (MPa)</th>
<th>Ultimate Strength (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Min</td>
<td>Max</td>
<td>Min</td>
</tr>
<tr>
<td>4,0</td>
<td>53,0</td>
<td>700</td>
</tr>
</tbody>
</table>

\textbf{S890QL}

<table>
<thead>
<tr>
<th>Dimensions (mm)</th>
<th>Yield Strength (MPa)</th>
<th>Ultimate Strength (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Min</td>
<td>Max</td>
<td>Min</td>
</tr>
<tr>
<td>4,0</td>
<td>53,0</td>
<td>900</td>
</tr>
</tbody>
</table>

Another of Konecranes steel suppliers is Ruukki who offers the ultra-high strength steel series Optim QC.

Figure 5. Mobile Crane; Optim QC Ultra-high-strength steel, Ruukki

\textsuperscript{11} (SSAB Informationsavdelning; Lena Westerlund, 2011)
\textsuperscript{12} (SSAB)
2.4.2 Composite
Composite as construction material comes with great advantages in terms of both weight and strength. Generally it is stated that the weight is greatly reduced if composite is used instead of steel. Other advantages are good fatigue properties and good thermal stability.

According to relevant expertise\textsuperscript{13} the design of this application demands an increased height and a greater cross section in order to achieve a stiffness that sustains the loads. A boom made of composite will otherwise bend more than a steel boom which must be considered in the material selection process. In order to achieve the high strength composite that is needed for this application the manufacturing technique is crucial.

Although the several advantages structural composite brings it is too comprehensive to be treated within the scope for a Bachelor’s thesis and is therefore excluded from this point. With this as background this project report will not further investigate composite as material selection.

2.4.3 Aluminum
Aluminum is a material with several appealing properties such as light weight and uncomplicated machining. A great advantage with aluminum is the possibilities to press the material through matrices and in this way obtain long profiles without weakening weld joints. Die pressing means a minimized machining of the details and it also allows the profiles to be designed in structurally advantageous manner. Aluminum does not run the same risk for corrosion as steel due to the material’s ability to form a protecting surface of oxides.

Fundamental is that density and Young’s modulus\textsuperscript{14} of aluminum is one third of steel’s. Strong aluminum alloys have yield strengths around 250 MPa\textsuperscript{15}.

\textsuperscript{13} (Wärn, 2014)
\textsuperscript{14} Modulus of Elasticity
\textsuperscript{15} (Schagerlind, 2014)
The properties of aluminum mean that compared to steel more material is needed to make the application endure the high strains it is subjected to. It is important to take into consideration that either the deflection or the stresses are dimensioning the design. The deflection requires that there is a possibility to rearrange the load and increase the outer dimensions. If not the thickness of the goods must be increased up to three times of corresponding steel application and therefore not mean a reduced weight compared to a steel construction. According to design engineers at SAPA the material thickness this application requires will demand special methods for pressing which needs further investigations together with manufacturer¹⁶.

With the thus far gathered knowledge it is possible to assume that a sufficiently strong and stable telescopic boom made out of aluminum will not result in particular gain regarding weight reduction. Conceivably, such goal can be achieved with an advanced design but this entails an investigation to an extent this thesis cannot embrace. With this as background the conclusion is that aluminum is not relevant for further exploration in this project.

¹⁶ (Schagerlind, 2014)
3 Method

Method describes the approach of the project.

3.1 Choice of Method

The method used for data acquisition concerning the project in this thesis is predominantly qualitative. This means the information and data have been collected through interviews with key personnel and through technical documentation. The latter consists of handbooks, drawings, CAD drawings and product presentations. To solve the present problem and subsequently come to a conclusion where an appropriate concept can be proposed, a wide range of factual sources has been used together with from the education obtained knowledge. The library of Linnaeus University, databases such as Science Direct\textsuperscript{17} and Academic Search Elite (EBSCO)\textsuperscript{18} have been serving for this purpose.

3.2 Project Process

Initially a presentation of the problem was held by Konecranes’ Technical Director Anders Nilsson who also has been the company mentor for the project. The project began with comprehensive research about the design problem. After the first meeting the process was planned in a so called Gantt\textsuperscript{19} chart where the time required for each step were visualized. In order to achieve a full perception of the task several meetings were held where the emphasis was to prepare the project and to engage a common approach. This is in respect of the sponsoring company, the student and the university’s expectations on the thesis.

Student visits in the assembling hall as well as where the test drives were set up. A visit at the sub-contractor who manufactures the telescopic boom was made where a thorough review of the manufacturing process also was given.

With the gathered information the theoretical parts of the report were written and the writing continued while describing and documenting the empirical studies. The empirical studies comprised of the current state analysis and definition of the problem

\textsuperscript{17}(Science Direct, 2014)
\textsuperscript{18}(Academic Search Elite, 2014)
\textsuperscript{19}(Gantt.com, 2012)
which lead to a number of requirements that became the directive in the investigation of a suitable concept.

Through the method of Quality Function Deployment\textsuperscript{20}, QFD, the requirements were refined and transformed into figures that were specified as target values in order to use as guidelines. The target values were also used to create a sufficient House of Quality matrix. From the results of the QFD a compilation of conclusions regarding the concept was made. With the compilation taken as a basis, concepts were calculated with respect to required strength. The concepts that were assessed to endure the strains was then drawn in the CAD program “SolidWorks”. In the program the concepts’ suitability are estimated both in terms of stress concentrations. These drawn concept sketches were evaluated in a Failure Modes and Effects Analysis\textsuperscript{21}, FMEA, which along with the QFD is a well-recognized tool in product development. The outcome of the FMEA was a weighting of concept sketches which is presented in the chapter Analysis. The best weighted was to be considered as the most suitable concept and from there on called the final concept. The final concept was specified and described in the chapter Conclusions.

The conclusion should be considered as an initial guidance in a continued product development work, because of the extent of a product development project of this magnitude.

The student was given the beneficial opportunity to work with the project in the office at Konecranes in Markaryd.

3.3 Specification of Requirements

The Specification of Requirement\textsuperscript{16} is a fundamental basis for product development and will be further investigated in later chapters. An old Specification of Requirements does not exist to take into consideration when establishing a new version for this application.

\textsuperscript{20}(Bergman & Klefsjö, 2012)
\textsuperscript{21}(Jackson, 2010)
3.4 Visits

Following visits were made in order for the student to get a holistic view of the project. The information and observations made during the visits will be developed in Chapter 4, Empirical Findings.

3.4.1 Konecranes

The project began with a visit in the workshop in Markaryd to which parts and components arrive and are assembled to reach stackers and heavy duty trucks. Under supervision of test driver Urban Linder the student was given the opportunity to drive reach stacker model 4533. The test run consisted of moving containers and try different manoeuvres. Urban Linder gave an informative tuition consisting of technical data, possible failure modes and operational use case behaviors which have become a valuable input to the project. As the reach stacker is used all over the world and in different climates the failure modes and behaviors differs, thus must these aspects be considered.

3.4.2 AB Bröderna Jansson – Nissavarvet

The boom is manufactured by the company AB Bröderna Jansson – Nissavarvet in Halmstad, among others. The collaboration between Konecranes (former SMV) and AB Bröderna Jansson – Nissavarvet dates back to the beginning of the 1990’s.

AB Bröderna Jansson – Nissavarvet is a mechanical workshop placed in Halmstad. The production comprises of part steel structures part contract production. The latter consists mainly of heavier, welded components to the heavy automotive industry. Examples of components are chassis, lifting booms, counterweight details and masts. The products are delivered surface treated and ready for assembling to the customers’ production. The methods are for instance drilling, plasma cutting, pressing in press brakes. Currently the company has capacity to press up to 7300 millimeter long and 15 millimeter thick plates in their own workshop. The welding operation is to a certain extent automatized where only more complicated weld joints are manually welded by licensed welders.
3.5 Expertise

In order to collect sufficient information about various possible material selections expertise has been consulted. The information about the consulted companies is derived from respective websites.

3.5.1 Konecranes
Supervisor of this project is Technical Director Anders Nilsson. Other individuals who assisted with support during the project are Design Manager Roger Persson, Designer Miroslav Antolovic and Test Driver Urban Linder.

3.5.2 Sapa AB
Sapa is a world leading actor in aluminum solutions. It is a new company which has joined the aluminum extrusion businesses of Sapa and Hydro. Through a global reach and local presence they are dealing with extrusions, building systems and precision tubing. The headquarters are located in Oslo, Norway. Contact person at Sapa is Johan Schagerlind, Design Engineer.

3.5.3 Marstrom Composite
Marstrom Composite is a manufacturer of composite structures located in Västervik, Sweden. Contact person is Per Wärn acting within technical sales and production at Marstrom Composite.
3.5.4 SSAB
SSAB, former Svensk Stål AB, is a Swedish steel concern. SSAB manufactures and supplies the global industry with steel. Contact person at SSAB is Mika Stensson.

3.5.5 Ruukki
Rautaruukki Abp, or Ruukki, is a Finnish steel company. Contact person at Ruukki is Bogoljub Hrnjez.

3.6 SolidWorks
SolidWorks is a CAD-program that is used in this project to analyze drawings and to perform simpler simulations through the function SimulationXpress. The function allows an insight in the behavior of strains in the detail. In order to do so the correct material and properties are selected and applied to the drawn application. Thereon fixtures and loads are designated and the program can then automatically calculate the expected behavior.
4 Empirical Findings

The chapter of Empirical Findings describes the current situation of the reach stacker in question. At this stage it is difficult to determine what is of importance and relevance in the further product development process. Thus, a comprehensive and mapping description of the machine follows.

4.1 Konecranes Reach Stacker

- Background
The machine was developed by engineers with previous experience of reach stackers. These developers possessed a great knowledge about reach stackers and with that as background the reach stacker was designed.

- Function
The reach stacker’s function is to handle intermodal cargo containers in ports, dry ports and terminals before further transportation to the final destination. The vehicle can transport containers fast and are able to stack them in three rows up to six containers high, depending on its access and the weight of the containers. Due to their great flexibility the reach stackers are very important handling solutions for containers with high demands on capacity and reliability.

- Work Environment
As mentioned the reach stacker is used in ports and terminals. The environment in ports can be highly corrosive which put demands on material and surface treatments. The machine is therefore painted to prevent that details corrode. Konecranes knows, however, that the corrosion that despite the treatment inevitable occurs on the machines is not a structural problem.

On the whole, the reach stackers are exposed for an aggressive surface wear, not only in corrosive environments like ports but also because of the tough mechanical wear the machines are subjected to. Examples of mechanical wear are collisions, impacts and shocks from cargo or other machines/objects in the work environment. Unevenness like potholes and railroad trails (which are common in terminals and ports) cause higher load stresses in the supporting structures when they are being overrun. This is owing to the reach stackers’ lack of shock absorption; only the tires work as dampers. Another
type of mechanical wear might take place in environments with a lot of sand. Here problems may occur when sand get stuck in lubricating oil or grease of vital parts in the machine, for instance in bearings and pistons.

Konecranes’ reach stackers are used in every continent of the world and hence subjected to very different climates. This is a factor that has to be taken in consideration when selecting material. For example because of that material can behave differently in different temperatures.

- **Design**
The supporting structure of the reach stacker is manufactured by medium and high strength steel. This requires certain dimensions in order to withstand the loads which totals up to 60-75% of a reach stacker’s weight.

- **Chassis**
The longitudinal beams of the reach stacker’s chassis are made of 4 welded steel plates, as a closed box section. The tower section has a cross member on its top. This gives the structure a high lifting capacity and a good torsional stiffness.
- Telescopic Boom

The telescopic boom has a slight rectangular profile and consists of an internal and an external boom with a hydraulic cylinder inside. The booms are manufactured from tempered steel with the designation S500Q. On top of the internal boom the bracket for the spreader is placed. The bracket is welded to the boom. In the other end the telescopic boom is attached to the cross member of the tower section. The boom is supported by hydraulic cylinders on both sides. The operating (lifting/sinking) of the boom demands a great part of the machine’s power, hence the weight of the boom is significantly affecting the total fuel consumption. From the logged data of the operation can be derived that the most common lift load is 25 ton and 7-12 ton (empty containers).

The boom’s safety factor is not calculated with the by standard statuary value as basis. Instead the boom is amply dimensioned and is therefore exceeding the valid safety factors.

Figure 8. Profile of Boom

The rectangular shape of the booms have uncomplicated seams and are therefore both easy and cost efficient to weld. A characteristic of the design is that the shape causes big stress points near the weld seams when the boom is in its full extension. To avoid failure the steel goods have to be dimensioned to endure the stress. The current choice of steel, S500Q, is of medium-high strength which consequently demands thicker goods than steel with higher strength. The boom is reinforced with welded pieces to give sufficient buckling resistance.

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22 See appendix I: Truconnect Logged Data
23 See appendix B: Safety Factor Calculations
The picture shows the boom from the short end perspective. The internal boom is tightly fitted into the external boom. The hydraulics that manoeuvres the boom is mounted in the middle of the cavity. The device placed inside the external boom in the left bottom corner holds a wire that measures the boom reach. Just underneath but on the outside of the outer boom an inclinometer is visible. Together with a third measuring instrument that measures the hydraulic pressure a computer calculates how much load the reach stacker can handle in every lifting situation. The operator can see this along with other data on a screen in the cabin.
View from inside of the cabin. In the right upper corner in figure 10 a screen is showing live views from two cameras placed in the front of the spreader.
- **Counterweight Principle**

The reach stacker acts by the principle of counterweight, which means that the momentum the payload creates over the front axle is counteracted by a rear counterweight. This means that all weight behind the front axle act as counterweight. Between the wheels in the rear end of the machine is a recess for the added counterweight. The recess contributes to a good front center visibility.

![Counterweight Accessability](image)

*Figure 12. Counterweight*

- **Accessibility**

In order to have easily accessible features the machine has flat surfaces and slip free steps so that maintenance and repair are facilitated. The reach stacker is maneuvered from the centered cabin by a steering wheel, a joystick and various controls. The customer can have several different optional extras according to their own requirements. Examples are sliding and elevating cabins, remote control units to steer the machine from distance and different functions in the software.
For safety reasons the reach stacker is equipped with fire and heat protection gears like fire extinguisher, fire suppression system, fire protection hoses and an oil cooling unit.

- **Spreader**

There are different options to handle the goods. Depending on the type of cargo either spreaders or a tool carrier system with various tools can be attached to the bracket on the inner boom. The spreader is developed and manufactured by the company Elme Spreader AB in Älmhult. Below the pictures show the locking device, namely pin, for lifting containers. The four pins in each corner of the spreader are centered and lowered into the lifting holes of the container. Inside the holes the pins automatically lock and can only be unlocked manually for emergency reasons.
- Standard and Safety

The safety regulations are described in “Safety of industrial trucks – Self-propelled variable reach trucks” according to the European standard EN 1459:1998. The reach stacker is governed by the International standard for industrial trucks; ISO 22915-3:2008 where the stability requirements are described. The requirements and guidelines apply to machinery operating within the temperature range of -20°C – 50°C. Safety requirements relevant for this project are following below.

- If corrosion of a part will interfere with its proper functioning it shall be provided with a corrosion resistant protective coating\(^\text{24}\).
- Pressure reducing valves shall be readily accessible for inspection and maintenance.
- Structural tests are governed by standard paragraph EN 1459 6.2.
- Dynamic tests are governed by standard paragraph EN 1459 6.3.2.
- Trucks shall be designed in such a way that they can be equipped with load retention devices such as load backrest extensions and top clamp stabilisers\(^\text{25}\).

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\(^\text{24}\) 5.4.5.3 Equipment (Swedish Standards Institute, SIS, 1999)
\(^\text{25}\) 5.5.8 Load stability (Swedish Standards Institute, SIS, 1999)
- Manufacturing and Assembling

Konecranes has two factories that produce reach stackers: one in Markaryd, Sweden, and one in Shanghai, China. Subcontractors deliver parts to the assembling hall in Markaryd where they then are assembled. No parts are manufactured in Markaryd. As this product development is aimed at, among other things, reducing environmental impacts, it is important to consider the transports of the material and components. In terms of total cost the shipping cost might probably also play a major role.

The reach stackers’ design may vary due to clients’ option. After assembling, every reach stacker is thoroughly test driven for approximately 8 hours by test drivers at the factory’s test track. During the test run the test drivers calibrate the reach stacker to optimize stability, pumps and hydraulics. The quality is hereby assured.

![Figure 15. Reach Stacker in Operation 1](image)
5 Current State Analysis

The current state analysis is performed in order to get a holistic perspective of the reach stacker. The objective is to cover every aspect that can be of importance for the project.

5.1 Defining the Project

A definition of the project is made to further clarify the purpose of the project and can be seen as an extension of the problem formulation in the chapter of Introduction.

5.1.1 Selecting Project

The project aims to improve the product reach stacker model SMV 4531 by making the heavy duty machine more efficient with respect to reduced fuel consumption, a decreased impact on environment and a lower total life cycle cost than the current design. In order to achieve an increased efficiency the weight of the machine should be reduced without impairing the current properties of the machine. One way to manage to fulfill these requirements is to change the material in certain details of the machine. Because of that the reach stacker is a heavy duty truck that acts by the principle of counterweight, the details possible to consider for weight reduction are limited to the part of the machine that operates in front of the front axle. As figure 13 shows these details are the telescopic boom and the spreader. The spreader is not manufactured by Konecranes and therefore this project will only treat the internal and external booms. The project will from now on be named Weight Reduction of Reach Stacker.

Figure 16. Reach Stacker in Operation 2
5.1.2 Pictures of the Current State

Figure 14 and 15 show Konecranes’ reach stacker SMV 4531 TB5. In figure 15 the telescopic boom can be seen in its full extension.

![Figure 17. Reach Stacker](image-url)
5.1.3 **Defining and Tailoring Concept**

The project runs within the confines of the Bachelor’s thesis. That means that the time limit is set to the extent of the course which is 15 credit points (ECTS) or 10 weeks of full-time work. Because the project is carried out by a student as a part of the education the budget for the project itself can be neglected.

5.1.4 **Identifying the Persons**

The owner of the project is the student performing this Bachelor’s thesis. The user of the project is both the assemblers of the crane and the end consumer. The client is the one who orders the project and also informs the student, in this case Anders Nilsson, technical Director at Konecranes. The customer is in this project defined as Konecranes.

5.1.5 **Mission Statement**

The goal is to propose a suitable and relevant concept design for a telescopic boom with a lower weight than the current design. The project will result in a report and an oral presentation that thoroughly outlines the conclusions.
5.2 Defining the Context

This paragraph contains the context study of the reach stacker. The context affects the product development to a great extent and it is therefore of importance to describe it. All the data is taken from Konecranes’ technical data sheets.

5.2.1 Reach Stacker SMV 4531

Konecranes’ model SMV 4531 has a service weight of 71 800 kg and is able to handle a maximum payload that amounts up to 45 000 kg. The capacity is given as the maximum payload per container row. The total weight of the telescopic boom is 13800 kg whereas the weight of the hydraulic extension cylinder is 763,4 kg. The technical data is listed below.

Table 3. Technical Data

<table>
<thead>
<tr>
<th>Reach Stacker SMV 4531</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Service Weight (kg)</strong></td>
</tr>
<tr>
<td><strong>Payload capacity 1st – 2nd – 3rd row (kg)</strong></td>
</tr>
<tr>
<td><strong>Telescopic Boom (kg)</strong></td>
</tr>
<tr>
<td><strong>Boom Weight/Service Weight (%)</strong></td>
</tr>
</tbody>
</table>
Technical Data:
1. **Wheelbase**: 6400 mm
2. **Tires**: 18.00x25”/PR40
3. **Engine (standard)**: Volvo TAD-1340-VE 256 / 1770 / 12,8. Stage 3B/4 within EU/USA/CAN/JP. Stage 2 elsewhere than EU/USA/CAN/JP due to higher required emission levels.
4. **Automatic Transmission (standard)**: DANA TE-27418 4+4 speed
5. **Drive Axle (Wet Disc Brakes) (standard)**: Kessler D102 (110 T / W=4,15 m) 18,00 x 25” / PR40 4127 – 4531 TB
6. **Pumps (Parker)**: Parker Load sensing pumps (105+75+60 cc) Electronic servo joystick
7. **Service Intervals**:

<table>
<thead>
<tr>
<th>Item</th>
<th>Filter (hours)</th>
<th>Oil (hours)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>500</td>
<td>500</td>
</tr>
<tr>
<td>Transmission</td>
<td>1000</td>
<td>1000</td>
</tr>
<tr>
<td>Drive Axle</td>
<td>2000</td>
<td>4000</td>
</tr>
<tr>
<td>Work Hydraulics (STD)</td>
<td>2000</td>
<td>4000</td>
</tr>
<tr>
<td>Work Hydraulics (HLL)</td>
<td>1000</td>
<td>12000</td>
</tr>
</tbody>
</table>

Operating Data:
1. **Life Length**: 20 000 hours or 500 000 load cycles
2. **Fuel Consumption**: 18-22 liter/hour depending on usage
3. **Average Operation Intensity**: 20 load cycles/hour
4. **Average Load**: 25 000 kg
5. **Average Power**: 60 kW
6. **Peak Power**: 250 kW
5.2.2 Current Manufacturing of Boom

Out of the total amount of reach stackers that were produced in Markaryd the year of 2013 a number of the booms were manufactured by AB Bröderna Jansson – Nissavarvet in Halmstad, Sweden. Some booms manufactured in Halmstad are more complicated in their design and therefore called “special booms”. Professedly, the material consumption for a standard boom is 9 tons while the special booms require 9.5-10 tons per piece. A review of the manufacturing process of the booms manufactured in Halmstad will hereby follow.

The boom arrives to the company as steel plates in the right dimensions in the material S500Q which is a steel belonging to the category Hot Rolled Products of Structural Steels. The plates are delivered in correct thickness to the workshop in Halmstad where the final shapes are flame cut. The plates are then assembled in fixtures and joined together manually by weld staples. After this the structure is fixed by hydraulic force to get ready for the longitudinal welds. The longitudinal welds are performed automatically by submerged arc welding. The dimension of the welding cord is 2 mm and to get the correct throat thickness of 10 mm there are double welding nozzles. With other words the joint is filled up by a weld pool of 2 * 2 mm welding cords of the brand Oerlikon. Some of the details are further machined before assembling and welding. The welding of the details is performed manually by licensed welders. All of the welds are visually inspected of both the welders and supervisors. When needed in aspect of crucial welds an accredited inspector is also controlling the weld joints.

When the boom design is complete the boom is blasted by steel granules or steel balls. This is made to prepare the surfaces for spray painting. The boom is finally painted with a solvent-based paint that is sprayed on the surface, both with and without added compressed air depending on wanted paint thickness. The paint is not eco-labeled and hard to recycle. At the present time the company considers it to be too inefficient to use water based paint that is more environmentally friendly due to the extended drying time. Instead the company use emission allowances for the solvent-based paint. After the boom is painted it is delivered to Konecranes in Markaryd by truck.
5.2.3 Collecting Customer Comments

Reportedly, the heavy duty automotive industry is a conservative line of business, which means that steel, the most conventional material in this area, is often preferred both by manufacturers and customers. Not only because of the advantages of using a well-established material that already dominate the market but also because of convention.

The goal is to develop a concept that is more efficient than the current design. A more efficient design will have positive effects in terms of environment and economy where one cannot be at the expense of another. A requirement that binds the economy and environment together is to develop a concept that contributes to reduced fuel consumption. This requirement is central and can be considered of high priority.

The weight of the reach stacker is like for all vehicles linked to its fuel consumption. It takes fuel both for transportation and for operating the load. The heavier the load is the more fuel is combusted which means that a weight reduction of the moving part, the telescopic boom, will gain lower fuel consumption because of the lower total lifting weight. A putative secondary benefit is that a lighter boom may give rise for downsizing of other components; for instance pumps and hydraulic cylinders.

The design life of the telescopic boom is usually limited by fatigue cracks. Therefore a requirement of high priority is that the telescopic boom must endure $10^6$ load cycles without risk fatigue cracks to occur. The reliability of a new concept shall comply with the current design and regulations which requires a system that endures the above mentioned number of load cycles. The telescopic boom must not have an impaired capacity than the present design.

As this application requires a high stiffness, the telescopic boom is dimensioned accordingly. This means that deflection, stability and buckling need to be considered. From a production technical perspective it is advantageous to use flat plates as they allow relatively high buckling loads without necessarily having particularly high yield strength.
To slide easily inside the external boom the internal boom rests on smooth plastic sections. Because of that the area that the inner boom rests upon becomes very small in full extension the stress increases with biggest concentrations in the side plates. A rectangular shape needs welded reinforcement ribs to increase buckling resistance of its side plates.

Regarding the economy for a product development Konecranes assesses that a new concept should repay itself within 3 years or 6000 hour in operation if a potential customer shall be willing to pay. Konecranes requires that a new concept must repay itself within a period of 2-3 years with respect to manufacturing and production. However, if the savings are great in long term longer repayment periods can be motivated in both scenarios.

Corrosion is not considered to be of great importance as long as the function of the boom is not impaired. The esthetics of the reach stackers is of less concern while in operation.

Other characteristics to touch in the project are maintenance, life cycle cost and total life cycle where the impact of the environment should be reduced compared to current design.

Konecranes’ reach stacker can be customized through a number of optional extras after every customer’s specifications. Despite this the material and machinery are the same for every reach stacker regardless of where in the world it is put into operation.

The market can be called “Business to Business” which means the market for the reach stacker by natural cause comprises more or less only of companies. At purchase Konecranes offers different types of financing programs. The machines can be leased or bought with service agreements that extend for a pre-determined period of time. During this time Konecranes or their dealers will handle all service and reparation of the machines. Konecranes has thus separate company sections for service and finances besides production.

Table 6 illustrates the gathered information from the customer.
<table>
<thead>
<tr>
<th><strong>Voice of the Customer</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Economy</strong></td>
</tr>
<tr>
<td>Limited cost</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td><strong>Design</strong></td>
</tr>
<tr>
<td>Capacity</td>
</tr>
<tr>
<td>Material Properties</td>
</tr>
<tr>
<td><strong>Maintenance</strong></td>
</tr>
<tr>
<td>Limited Cost</td>
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<tr>
<td></td>
</tr>
<tr>
<td>Accessibility</td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>

\(^{26}\) Tare weight.
### Summary of Product Objectives

Table 7 compiles wanted product objectives which will lay as ground for the specification of requirements.

Table 6. Product Objectives

<table>
<thead>
<tr>
<th>Product Objectives</th>
</tr>
</thead>
<tbody>
<tr>
<td>High strength</td>
</tr>
<tr>
<td>Investment cost that repays itself within a period of 2-3 years.</td>
</tr>
<tr>
<td>Any price increase of the detail repays itself within 3 years or 6000 hours in operation.</td>
</tr>
<tr>
<td>Cost efficient maintenance</td>
</tr>
<tr>
<td>Easy performable maintenance</td>
</tr>
<tr>
<td>Contribute to an eco-efficient operation</td>
</tr>
<tr>
<td>Low weight</td>
</tr>
<tr>
<td>Safe</td>
</tr>
</tbody>
</table>

### 5.3 Defining Functional Requirements

Defining functional requirements means that use cases and preferences are refined in order to get a specified conclusion of the functional requirements.
5.3.1 Collecting Use Cases
To collect the use cases it is needed to further define the persons that will come in contact with the process. The person that will handle the machine on daily basis is the operator. The operator can be assumed to be employed by the customer where the customer is a company that purchases a Konecranes reach stacker. Service personnel will maintain and repair the process. The client of the project is as mentioned in previous chapter the Technical Director of Konecranes Lifttrucks AB, Anders Nilsson. The company is Konecranes Lifttrucks AB. The company is presented by a Management Board who is the executive body regarding projects and investments. The system is the object for this product development which is the reach stacker’s telescopic boom. The input the boom is handling is cargo; containers and Bulk.

5.3.2 Use Cases and Prioritization of Use Cases
The use cases found for the system are prioritized after importance in handling of the system. H stands for high priority, M is medium and L is low importance. This is visible in table 4.4. The table the focus on both the primary use cases and also on the secondary use cases; when the systems fails or is used improperly.

The priority of use cases is made from a design point of view. The function is therefore prioritized higher than other issues. For example; the boom’s construction will not affect the operator which is the reason that has a low priority.
<table>
<thead>
<tr>
<th>Use Case</th>
<th>Priority</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Primary</strong></td>
<td></td>
</tr>
<tr>
<td>Operator operates system.</td>
<td>L</td>
</tr>
<tr>
<td>Operator contacts Konecranes’ service personnel if problem occurs.</td>
<td>L</td>
</tr>
<tr>
<td>Service personnel maintain and repair the system.</td>
<td>M</td>
</tr>
<tr>
<td>Customer employs/informs operator.</td>
<td>L</td>
</tr>
<tr>
<td>Client informs owner of project.</td>
<td>L</td>
</tr>
<tr>
<td>Customer purchases/hires/leases reach stacker.</td>
<td>M</td>
</tr>
<tr>
<td>Company employs personnel.</td>
<td>L</td>
</tr>
<tr>
<td>Company invests in project and possible changes in system’s production.</td>
<td>H</td>
</tr>
<tr>
<td>System is used by operator/personnel.</td>
<td>L</td>
</tr>
<tr>
<td>System is handling input.</td>
<td>H</td>
</tr>
<tr>
<td>System performs (approximately) 15 load cycles per hour.</td>
<td>M</td>
</tr>
<tr>
<td>System has a life length of 200 000 hours in operation.</td>
<td>M</td>
</tr>
<tr>
<td><strong>Secondary</strong></td>
<td></td>
</tr>
<tr>
<td>The system is used constantly.</td>
<td>L</td>
</tr>
<tr>
<td>The system fails due to fatigue failure.</td>
<td>M</td>
</tr>
<tr>
<td>The system fails due to vibrations.</td>
<td>M</td>
</tr>
<tr>
<td>The system fails due to incorrect use.</td>
<td>L</td>
</tr>
<tr>
<td>The system fails due to unexpected events.</td>
<td>L</td>
</tr>
</tbody>
</table>

5.3.3 **Description of Use Cases Behavior**

The described use cases are “Company invests in project and possible changes in system’s production” and “System is handling input”. This use cases are prioritized H, high, and therefore further described.
### Table 8. Use Cases Behaviour 1

**Company invests in project and possible changes in system’s production**

<table>
<thead>
<tr>
<th>Company</th>
<th>System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Company holds production.</td>
<td></td>
</tr>
<tr>
<td>Company prepares order from customer.</td>
<td></td>
</tr>
<tr>
<td>Company purchases material.</td>
<td></td>
</tr>
<tr>
<td>Company purchases system from subcontractors.</td>
<td></td>
</tr>
<tr>
<td>System is manufactured by subcontractor.</td>
<td></td>
</tr>
<tr>
<td>System is delivered to Konecranes production.</td>
<td></td>
</tr>
<tr>
<td>The system is assembled in Konecranes’ production.</td>
<td></td>
</tr>
<tr>
<td>The product (including system) is tested and calibrated.</td>
<td></td>
</tr>
<tr>
<td>The system is ready for delivery to customer.</td>
<td></td>
</tr>
<tr>
<td>Company sells product (including system).</td>
<td></td>
</tr>
<tr>
<td>The system is delivered to customer.</td>
<td></td>
</tr>
</tbody>
</table>

The second use case behaviour of high importance is the input handling which is seen in table 10.
<table>
<thead>
<tr>
<th><strong>System is handling input</strong></th>
<th>Operator</th>
<th>System</th>
<th>Input (cargo, containers, Bulk)</th>
</tr>
</thead>
<tbody>
<tr>
<td>The operator starts (system).</td>
<td></td>
<td></td>
<td>Input is ready for handling.</td>
</tr>
<tr>
<td>The operator steer the reach stacker to desired location.</td>
<td></td>
<td></td>
<td>The system is activated.</td>
</tr>
<tr>
<td>Operator navigates boom and spreader.</td>
<td></td>
<td></td>
<td>Boom is lowered/elevated.</td>
</tr>
<tr>
<td>(The engine spins in order to provide hydraulic pumps with power to maneuver boom and spreader).</td>
<td></td>
<td></td>
<td>Boom and spreader are put in right position to handle (lift and move) input.</td>
</tr>
<tr>
<td>Input is automatically locked to spreader.</td>
<td></td>
<td></td>
<td>Input is handled.</td>
</tr>
<tr>
<td>Operator manually unlocks input from spreader.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operator navigates boom and spreader.</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
5.3.4 **Functional Requirements**

After analyzing customer comments and use cases a summary of functional requirements of the system can be consolidated in following table.

*Table 10. Functional Requirements*

<table>
<thead>
<tr>
<th>Functional Requirements</th>
</tr>
</thead>
<tbody>
<tr>
<td>The system shall at least be able to handle the same payload and have the same scope as current design.</td>
</tr>
<tr>
<td>The system shall be resistant to corrosion</td>
</tr>
<tr>
<td>The system shall have a safety factor of 2.</td>
</tr>
<tr>
<td>The system shall have a reduced weight compared to current design.</td>
</tr>
<tr>
<td>The system shall be easy to maintain and facilitate maintenance of other components (for instance valves).</td>
</tr>
</tbody>
</table>

5.3.5 **Final Requirements**

The Final Requirements aims to specify the requirements that shall be prioritized in the investigation of a possible change of material for the Reach Stacker’s telescopic boom. The final requirements of the system are visible below.

*Table 11. Final Requirements*

<table>
<thead>
<tr>
<th>Index</th>
<th>Originating Requirements (OR)</th>
<th>Abstract Function Name</th>
</tr>
</thead>
<tbody>
<tr>
<td>OR 1</td>
<td>The system shall repay itself within a period of 2-3 years with respect to own production.</td>
<td>Investment</td>
</tr>
<tr>
<td>OR 2</td>
<td>The system shall in operation repay itself within 3 years or 6000 hours.</td>
<td>Price</td>
</tr>
<tr>
<td>OR 3</td>
<td>The system shall have a total life cycle cost that is more cost efficient than current design.</td>
<td>Life Cycle Cost (LCC)</td>
</tr>
<tr>
<td>OR 4</td>
<td>The system shall have a reduced weight compared to current design.</td>
<td>Weight Reduction</td>
</tr>
<tr>
<td>-------</td>
<td>------------------------------------------------------------------</td>
<td>-------------------</td>
</tr>
<tr>
<td>OR 5</td>
<td>The system shall at least be able to handle the same payload and have the same scope as current design.</td>
<td>Payload; Scope</td>
</tr>
<tr>
<td>OR 6</td>
<td>The system shall endure $10^6$ load cycles.</td>
<td>Endurance</td>
</tr>
<tr>
<td>OR 7</td>
<td>The system shall be easy to repair/maintain.</td>
<td>Maintenance Easy</td>
</tr>
<tr>
<td>OR 8</td>
<td>The system shall be cost efficient to repair/maintain.</td>
<td>Maintenance Cost</td>
</tr>
<tr>
<td>OR 9</td>
<td>The system shall be resistant to corrosion.</td>
<td>Corrosion</td>
</tr>
<tr>
<td>OR 10</td>
<td>The system shall contribute to reduced energy consumption while operating.</td>
<td>Energy Consumption in Operation</td>
</tr>
<tr>
<td>OR 11</td>
<td>The system should be produced in an eco-efficient way.</td>
<td>Eco-Production</td>
</tr>
<tr>
<td>OR 12</td>
<td>The system shall have a life cycle assessment where the impact of the environment is reduced compared to current design.</td>
<td>LCA</td>
</tr>
<tr>
<td>OR 13</td>
<td>The system shall have a safety factor of 2.</td>
<td>SF</td>
</tr>
<tr>
<td>OR 14</td>
<td>System has a life length of 200 000 hours in operation.</td>
<td>Life Length</td>
</tr>
<tr>
<td>OR 15</td>
<td>The system shall be reliable and safe for the operator.</td>
<td>Safety</td>
</tr>
</tbody>
</table>
6 Measuring the Need and Setting Targets

The aim of this chapter is to translate the derived requirements from the previous chapter to useful data and numbers.

6.1 Figures to Create Targets and Limit Values

It is of great importance to create targets and limit values through numbers and figures in order to define the requirements. The targets and limit values are used in a Quality Function Deployment which is further described in next chapter. Below presentations of the relevant numbers and figures follow.

- Weight and Fuel Consumption
A crucial part of the project task is to connect weight to fuel consumption in order to get comparative figures. The formula for this must describe the extent to which the weight of the design is affecting the fuel consumption. In this project the formula for this is based on assumptions and estimations about the reach stacker’s operation. The data is taken from readouts from Konecranes’ data system Truconnect. Truconnect logs data from every course of events in the reach stacker’s operation. The current fuel consumption amounts to 18-22 liter/hour. The target is to decrease this number to 15-18 liter/hour.

- Legislation and Safety
Paragraph 4.1.1 expounds the standards the reach stacker is governed by. A target value that can be derived from standards is the safety factor the telescopic boom should be calculated on. It is also important to take into account the corrosion class of the environment that the machine is operating in in order to determine sufficient surface treatment and material.

The safety factor of the telescopic boom is set to be not lower than 2. Calculations of safety factors are based upon figures given in the documents “Lug Boom Suspension” and the appendix A: “Calculations of the Lifting Boom”. According to calculations the internal boom is subjected to the greatest load of the two booms when operating 1st row container stacks.

27 See appendix I: Truconnect Logged Data
- Corrosion

The reach stackers are often used in highly corrosive environments. According to the corrosion class system the environment is in its worst scenario best described as C5\textsuperscript{28}. The recommended surface treatment\textsuperscript{29} for this scenario is Z275+300\textmu m alternatively stainless steel as material. Z275+300\textmu m means that plate should have a zinc protection of 275 gram/m\textsuperscript{2} plus a paint thickness of 300 \textmu m in order to withstand the rough conditions.

- Total Material Cost

The total material cost is depending on the price of the chosen material and on the total material consumption. Reportedly, the material consumption for the booms is distributed as 9 tons for the standard booms and 9.5-10 tons for the special booms. The special booms comprises of 10-12\% of the total amount of 130 details, this according to the statistics of the year of 2013. The total amount of consumed material is given below.

\[
\left( \frac{9.5 + 10}{2} \right) \times \left( \frac{0.10 + 0.12}{2} \right) + 9 \times \left( 1 - \frac{0.10 + 0.12}{2} \right) \times 130 = 1180,725 \text{ tons}
\]

The amount purchased steel of designation S500Q is today at least 1181 tons. The target is to reduce the use of material with 75\% which means 886 ton.

- Design

The current design allows a payload of 45 000 kg in 1\textsuperscript{st} row, 31 000 kg in 2\textsuperscript{nd} and 16 000 kg in the 3\textsuperscript{rd}. These range and maximum payloads are absolute requirements.

- Fatigue Strength

The reach stacker has an average life length of 200 000 hours. To resist the high stress the machine is subjected to during this time the number of load cycles the boom must endure $10^6$ cycles without risk of obtaining fatigue cracks.

\textsuperscript{28} (Schweitzer, 1996)
\textsuperscript{29} (Infosteel; Zinkinfo Benelux; VOM vzw the Belgian association for surface technologies of materials; he ‘Mobility and Public Works’ department of the Flemish Region, 2012)
- **Material Strength**

The current material, steel S500Q, incorporates yield strength of 500 N/mm$^2$. If the desire to reduce the material consumption shall be met the material strength must be greater than the current value.

- **Benchmark Competition**

In order to determine the appropriate approach to achieve the targets of the project the competitors’ solutions of the problems are mapped. This represents the benchmark competition. Competing manufacturers and their corresponding products are Liebherr LRS 645$^{30}$, Kalmar DRG450$^{31}$, Hyster RS45/46$^{32}$.

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$^{30}$ (Liebherr, 2014)  
$^{31}$ (Kalmar Industries, 2014)  
$^{32}$ (Hyster Big Trucks, 2014)
7 Quality Function Deployment

The purpose with the quality function deployment is to get a result that is satisfactory for the customer.

7.1 House of Quality

With the in the previous chapter gathered requirements and target values a House of Quality matrix is established. The conclusions of the matrix serve as basis for investigating and forming concepts. These concepts are evaluated in a Failure Mode and Effects Analysis, FMEA, where the result illustrates the most suitable concept. To measure the need and set targets the method of Quality Function Deployment, QFD, is used. From the Final Requirements in the previous chapter Demanded Quality is derived in the QFD that can be seen in appendix. The relative priority for each objective is measured in percentage and ranked from highest (1) to the lowest (14). The targets/limit values have their origin in voice of the customer, standard sheets and technical data sheets. Appendix D; House of Quality shows the established HoQ.

7.1.1 Ranked Product Objectives (Goals) from House of Quality

The results from the HoQ can be seen in table 13. The ranking in the left column mean that the most desirable product objective is listed as 1 followed by the rest in descending order.

Table 12. Ranked Product Objectives

<table>
<thead>
<tr>
<th>Product Objective</th>
<th>Target or Limit Values</th>
<th>Relative Priority (%)</th>
<th>Rank</th>
</tr>
</thead>
<tbody>
<tr>
<td>High fatigue strength construction material.</td>
<td>10^6 load cycles</td>
<td>15,8</td>
<td>1</td>
</tr>
<tr>
<td>Decreased fuel consumption.</td>
<td>&lt;18 liter/hour</td>
<td>12,9</td>
<td>2</td>
</tr>
<tr>
<td>A design with good accessibility to important components.</td>
<td></td>
<td>11,7</td>
<td>3</td>
</tr>
<tr>
<td>Low total material price.</td>
<td>886 ton</td>
<td>9,3</td>
<td>4</td>
</tr>
<tr>
<td>High strength design.</td>
<td>Payload</td>
<td>8,6</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>45/31/16 ton</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(QFD Online, 2010)
<table>
<thead>
<tr>
<th>Requirement</th>
<th>Description</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Short transportations for material and product.</td>
<td>Within Europe/ within Asia</td>
<td>8,6</td>
<td>6</td>
</tr>
<tr>
<td>Decreased material consumption.</td>
<td>&lt;9 ton/&lt;10 ton</td>
<td>6,7</td>
<td>7</td>
</tr>
<tr>
<td>A material possible to produce energy-efficiently.</td>
<td>Primary material 100 MJ/tons material</td>
<td>6,7</td>
<td>8</td>
</tr>
<tr>
<td>Safety Factor according to standard practice.</td>
<td>SF&gt;2</td>
<td>5,0</td>
<td>9</td>
</tr>
<tr>
<td>Few changes in existing infrastructure of manufacturing/assembling.</td>
<td></td>
<td>4,5</td>
<td>10</td>
</tr>
<tr>
<td>High strength construction material.</td>
<td>Corresponding to present design (S_y=500 MPa)</td>
<td>4,3</td>
<td>11</td>
</tr>
<tr>
<td>Anticorrosive surface.</td>
<td>Z275+300μm alt. stainless steel</td>
<td>2,4</td>
<td>12</td>
</tr>
<tr>
<td>Recyclable material.</td>
<td>Secondary material: 50 MJ/tons</td>
<td>2,2</td>
<td>13</td>
</tr>
<tr>
<td>Manufacturing of product in line with strict legislation regarding quality and environment.</td>
<td></td>
<td>1,2</td>
<td>14</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>100</td>
<td></td>
</tr>
</tbody>
</table>

7.2 Conclusions of Requirements, Objectives and Affecting Factors

In chapter 2, Theory, different material and design alternatives are presented. Based on the information gathered thus far and gained data it is possible to sort the materials suitable for this application.
From the in the HoQ weighted requirements can be inferred that the appropriate construction material for this project is a high strength steel with greater strength than the currently used material. Depending on the yield strength the material thickness can be decreased. The density is approximately equal for the steel alloys in question which mean that the relationship between the weight and the material thickness is straight.

Steel as construction material opens for a continued collaboration with Konecranes’ already established suppliers. Presumably, this affects the total price of the investment positively due to an already existing partnership.

The current design can be considered to be of uncomplicated shape and is therefore both easy and cheap to manufacture. A continued usage of steel opens for a cost effective solution in terms of design.

7.3 Generating Concepts

From the collected data it is possible to outpoint two strategies to gain reduced weight; material selection and design. These two strategies can obviously not be separated due to their dependence on each other. The conclusion drawn on the basis of the collected information is that a low cost solution is to be preferred in this project. A low cost solution means a minimal interference in the existing production infrastructure. With a design that is based on the existing design a minimal amount of actions and investments are required in the production line.

The thickness of the most affected plates in the most affected zone (row) is calculated from the premise of a safety factor of 2.

Steel manufacturers offers different types of steel alloys suitable for this application. Two products relevant for this boom in particular are Weldox 700 and Weldox 900 provided by SSAB\textsuperscript{34}. The numbers represent yield strength.

\textsuperscript{34} (SSAB)
Using steel with higher yield strength means that thinner dimensions will endure the loads. However, a telescopic in a similar configuration as current but with thinner dimensions will deflect down more which is not a desired consequence. Also, the stability is affected negatively. Therefore, these effects must be suppressed through the design.

7.3.1 High Strength Steel with Rectangular Design
A concept with a high strength steel like S690QL as structural steel opens for a cost efficient investment. This due to the possibility to avoid excessive interference in the production. The design can presumably remain similar to current.

7.3.2 Ultra High Strength Steel with Rectangular Design
In earlier chapters it is stated that reinforcement in form of for example latticework may be beneficial for use of thinner plates. Latticework gives strong structures but confines the room for components inside the boom, for instance the important hydraulic cylinder.
Use of ultra-high strength steel, like S890QL, gives opportunity to use thin plates. Unfortunately this means that the boom will deflect far more which means that reinforcements are needed to restrain the unwanted consequence. Latticework and ultra-high strength steel in combination probably have good potential to gain reduced weight as concept.

7.3.3 Ultra-High Strength Steel with U-shaped Design

Similar applications show that heavy duty cranes gain strength when corners are rounded. Especially mobile cranes that are subjected to great loads often incorporate this design. The rounded shape contributes to a beneficial stress distribution which possibly could be applied to a reach stacker. According to similar applications the material selection should be a high strength steel in order to sustain the heavy loads.
8 Failure Mode and Effects Analysis

*In the Failure Mode and Effects Analysis, FMEA, the concepts are examined from a perspective of possible failures.*

8.1 Conducting Failure Modes and Effects Analysis

By ranking “Probability of Occurrence of Failure”, Severity of Failure” and “Likelihood of Detecting Failure” from 1 to 5 where 1 represents the lowest risk and 5 the highest, and then multiply the factors a value is given. This value signifies the Risk Priority Number, RPN. By adding all RPN values that are derived from assumed possible failure a total RPN value is formed. Through the weighting system of RPN the concept that is predicted to pose the fewest risks gets the lowest score. The full FMEA of this project can be found in appendix E. In table 14 possible failures are listed. These are used in the full FMEA\(^{35}\).

The weighting of the RPN is undertaken by the student in consultation with the customer.

**Table 13. Possible Failures**

<table>
<thead>
<tr>
<th>Assumed Possible Failures</th>
</tr>
</thead>
<tbody>
<tr>
<td>The boom is deflecting too much.</td>
</tr>
<tr>
<td>The boom is not enough stable.</td>
</tr>
<tr>
<td>The boom is not light enough.</td>
</tr>
<tr>
<td>The boom does not resist fatigue.</td>
</tr>
<tr>
<td>The boom is not enduring the stress.</td>
</tr>
<tr>
<td>The boom is subjected to cracking weld joints.</td>
</tr>
<tr>
<td>The boom is subjected to buckling.</td>
</tr>
</tbody>
</table>

\(^{35}\) See appendix E; Failure Mode and Effects Analysis
9 Result

The chapter is presenting the results of the CAD simulations and the FMEA.

9.1 CAD Simulations

When screening the generated concepts the starting point is “worst case scenario”. The boom is subjected to its greatest stress when lifting 45 ton in the 1st row of containers. The external boom is less subjected to the stress due to its larger dimensions and supporting hydraulic cylinders. Therefore, the figures from CAD drawings illustrate the results of simulating the internal boom in SolidWorks application SimulationXpress.

The fixtures are set in the supported “inner” end and the load is applied to the outer end where the spreader is attached. The simulations shall be viewed as indicators of the reality. Figure 20 shows the current state. The red areas indicate where the safety factor, SF, reach a value below 2. In the reality of the current design, the SF is 2,4 at its lowest value. Because of that the SF is set to be greater than 2, attention should be given to red areas due to the risk of buckling.

Figure 20. Current State Boom; von Mises Stress
9.1.1 High Strength Steel; S 690 QL

The total weight of the boom amounts to 2974 kg which means a weight reduction of 26% compared to current design. Figure 22 shows the same boom but with 5 mm reduction of the dimensions.
9.1.2 Ultra-High Strength Steel; S 890 QL

Figure 24 and 25 show the same boom but with 5 mm reduction of the dimensions which means same weight as above: 2974 kg (26% lighter than current design). Notable is that the CAD drawing does not contain any reinforcements.
If the dimensions are even more reduced, 10 mm reduction of top and bottom plate and 8 mm of side plates the weight is reduced to 2269 kg (43% lighter than current).
Finally the result of a U-shaped boom is showed. Reinforcements are placed as ribs inside the boom to restrain the great stress. The weight here becomes 3793 kg which is a weight reduction of 5%. The figures 28, 29 and 30 show the concept.
Figure 29. Ultra-high Strength Steel 3; Safety Factor

Figure 30. Close-up U- shape
9.1.3 **Comparison Table**

Below a comparison table illustrates the results.

**Table 14. Concept Comparison**

<table>
<thead>
<tr>
<th>Material thickness side/top/bottom plates (mm)</th>
<th>S500Q (Default)</th>
<th>S690QL</th>
<th>S890QL (1)</th>
<th>S890QL (2)</th>
<th>S890QL (3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material thickness side/top/bottom plates (mm)</td>
<td>15/20/20</td>
<td>10/15/15</td>
<td>10/15/15</td>
<td>7/10/10</td>
<td>20/10 (U-shaped)</td>
</tr>
<tr>
<td>Weight (kg)</td>
<td>3992</td>
<td>2974</td>
<td>2974</td>
<td>2269</td>
<td>3793</td>
</tr>
<tr>
<td>Percentage of weight reduction</td>
<td>-</td>
<td>25,51%</td>
<td>25,51%</td>
<td>43,16%</td>
<td>5%</td>
</tr>
<tr>
<td>Results of simulation</td>
<td>Default</td>
<td>As default</td>
<td>Better than default</td>
<td>Worse than default</td>
<td>Worse than default</td>
</tr>
</tbody>
</table>

9.2 **FMEA**

From the FMEA\(^{36}\) the information visible below can be read. In terms of risk of failure concept 1 is weighted the lowest with a total RPN\(^ {37}\) of 214 while the concepts 2 and 3 share a total RPN of 246.

**Table 15. Accumulated Risk Priority**

<table>
<thead>
<tr>
<th>Concept</th>
<th>Risk Priority Number, RPN</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. High Strength Steel with Rectangular Design (S690QL)</td>
<td>214</td>
</tr>
<tr>
<td>2. Ultra-high Strength Steel with U-shaped Design (S890QL)</td>
<td>246</td>
</tr>
<tr>
<td>3. Ultra High Strength Steel with Rectangular Shape (S890QL)</td>
<td>246</td>
</tr>
</tbody>
</table>

\(^{36}\) See appendix E; Failure Mode and Effects Analysis

\(^{37}\) See abbreviations
9.3 Calculations of the Lifting Boom

From the guidance from CAD Simulations and the FMEA it has been assessed that concept 1 seems to be the most relevant concept for a continued exploration. Hence, the concept is calculated to investigate whether the design will endure the stress it is subjected to if the material is changed to S690QL and the dimensions are reduced with approximately 25%. The calculations are found in Appendix C; Calculations of Lifting Boom – Concept 1. The result from them is that concept 1 corresponds well to the current design in terms of stress. In the most affected position, internal boom 1st row, the safety factor value amounts up to 2.61.

Table 16. Concept 1 Figures

<table>
<thead>
<tr>
<th>Basis of calculations</th>
<th>External</th>
<th>Internal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Section Modulus, Z</td>
<td>18501763 mm³</td>
<td>1063892 mm³</td>
</tr>
<tr>
<td>Centroidal distance, ŷ</td>
<td>435 mm</td>
<td>385 mm</td>
</tr>
<tr>
<td>Outer base dimension, B</td>
<td>880 mm</td>
<td>800 mm</td>
</tr>
<tr>
<td>Inner base dimension, b</td>
<td>860 mm</td>
<td>780 mm</td>
</tr>
<tr>
<td>Outer height dimension, H</td>
<td>870 mm</td>
<td>770 mm</td>
</tr>
<tr>
<td>Inner height dimension, h</td>
<td>825 mm</td>
<td>740 mm</td>
</tr>
<tr>
<td>Top/bottom plate thickness, t</td>
<td>22.5 mm</td>
<td>15 mm</td>
</tr>
<tr>
<td>Side plate thickness, t</td>
<td>10 mm</td>
<td>10 mm</td>
</tr>
</tbody>
</table>

9.3.1 Calculated Possible Weight Reduction

Table 18 shows the percentage of the decreased material thickness of concept 1.

Table 17. Decreased Material Thickness

<table>
<thead>
<tr>
<th></th>
<th>External</th>
<th>Internal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top/bottom plate thickness, t</td>
<td>25%</td>
<td>25%</td>
</tr>
<tr>
<td>Side plate thickness, t</td>
<td>33%</td>
<td>33%</td>
</tr>
</tbody>
</table>

The following table illustrates the weight of the external and internal booms that concept 1 will bring.
Table 18. External Boom Concept Weight

<table>
<thead>
<tr>
<th></th>
<th>Current weight (kg)</th>
<th>Concept weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Top plate</strong></td>
<td>1873,9</td>
<td>1405,4</td>
</tr>
<tr>
<td><strong>Bottom plate</strong></td>
<td>1754,2</td>
<td>1315,7</td>
</tr>
<tr>
<td><strong>Side plates</strong></td>
<td>2*840,4</td>
<td>2*554,7</td>
</tr>
<tr>
<td><strong>Total weight (kg)</strong></td>
<td>5309</td>
<td>3830,5</td>
</tr>
<tr>
<td><strong>Weight reduction in percentage</strong></td>
<td>-</td>
<td>28%</td>
</tr>
</tbody>
</table>

**Internal Boom**

<table>
<thead>
<tr>
<th></th>
<th>Current weight (kg)</th>
<th>Concept weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Top plate</strong></td>
<td>1174,4</td>
<td>880,8</td>
</tr>
<tr>
<td><strong>Bottom plate</strong></td>
<td>971,7</td>
<td>728,8</td>
</tr>
<tr>
<td><strong>Side plates</strong></td>
<td>2*789,3</td>
<td>2*520,9</td>
</tr>
<tr>
<td><strong>Total weight (kg)</strong></td>
<td>3725</td>
<td>2651,4</td>
</tr>
<tr>
<td><strong>Weight reduction in percentage</strong></td>
<td>-</td>
<td>29%</td>
</tr>
</tbody>
</table>

The total weight of the telescopic boom becomes 6481,9 kg which means that concept 1 will provide a weight that is reduced with 28% compared to current design.
10 Analysis

In this part the collected data is analyzed. It identifies and discusses the factors that have emerged during the work, how they interact, how possible solutions can be used, how a change can be developed and what potential there is for the future.

10.1 General Exploration

To obtain reduced weight while maintaining strength by increasing the structural steel’s yield strength is an acknowledged approach with scientific support. The U.S. Department treats the subject in their workshop report “Trucks and Heavy-Duty Vehicles Technical Requirements and Gaps for Lightweight and Propulsion Materials”\textsuperscript{38}. About high strength steel and advanced high strength steel (structural) they write:

“The combination of low material cost, high strength and stiffness (modulus), and an extensive modeling and design database make various grades of high-strength steels highly competitive materials for HDV\textsuperscript{39} applications.”

In the report the fuel-based cost saving is calculated, here presented in table 16. The report authors mean that if the weight of heavy duty vehicles is reduced with 10\% the cost saving may amount up to $8,30 per kilogram.

\textsuperscript{38} (U.S Department of Energy, 2013)
\textsuperscript{39} See abbreviations
### Table 19. Heavy-Duty Vehicle Chassis Metrics and Targets

<table>
<thead>
<tr>
<th>HDV Weight Reduction Metrics</th>
<th>2010 (Overall Baseline)</th>
<th>2025</th>
<th>2030</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Overall Weight Reduction</strong></td>
<td>Materials mostly steel, close to full optimization</td>
<td>10% lighter weight using advanced materials</td>
<td>10% weight reduction using advanced materials</td>
</tr>
<tr>
<td><strong>Fuel-Based Cost Tolerance</strong></td>
<td>–</td>
<td>$8,30/kg Saved*</td>
<td>$8,82/kg Saved*</td>
</tr>
<tr>
<td><strong>$/kg saved</strong></td>
<td>–</td>
<td>Lighter by 20%</td>
<td>Lighter by 20%</td>
</tr>
<tr>
<td><strong>Ladder frames</strong></td>
<td>–</td>
<td>Lighter by 10%</td>
<td>Lighter by 10%</td>
</tr>
<tr>
<td><strong>Wheels and tires</strong></td>
<td>–</td>
<td>Lighter by 5%</td>
<td>Lighter by 10%</td>
</tr>
<tr>
<td><strong>Axles</strong></td>
<td>–</td>
<td>Lighter by 10%</td>
<td>Lighter by 10%</td>
</tr>
<tr>
<td><strong>Brakes</strong></td>
<td>–</td>
<td>Lighter by 10%</td>
<td>Lighter by 10%</td>
</tr>
<tr>
<td><strong>Springs</strong></td>
<td>–</td>
<td>Lighter by 10%</td>
<td>Lighter by 10%</td>
</tr>
<tr>
<td><strong>Chassis accessories: fuel system, exhaust, battery systems</strong></td>
<td>–</td>
<td>Lighter by 15%</td>
<td>Lighter by 25%</td>
</tr>
</tbody>
</table>

*Represents Cost Neutrality for Class 8 trucks (heavy duty) based on Environmental Impact Assessment, EIA diesel fuel price projections, 4 year payback and annual Vehicle Miles Travelled, VMT, of 160 900 km.
The steel manufacturer Ruukki\textsuperscript{40} provides an efficiency calculator on their homepage where it is possible to calculate the estimated fuel saving when reducing heavy duty vehicle’s Eigen weight by using high strength steel as structural steel\textsuperscript{41}. According to the efficiency calculator and the selected data input a weight reduction of 30\% can result in a decreased fuel consumption of 3\%. The calculator is evidently not applicable for reach stackers but can serve as guidance.

The fully assembled application consists of several subassemblies which require suitable joining techniques. It is likely to assume that a maximized automation of possible welding is to strive for due to efficiency reasons. Also an automatized welding process is preferable with respect to the health aspect for the workers. This put demands on the welding methods as well as the material referring to both welding electrodes and steel. Without risking that heat accumulation affects the material negatively, the selected steel should not require preheating before welding in order to minimize inefficient holding times. Other methods to join details are by screw joints or adhesive bonds.

In an early state it has been clear that appropriate materials must have an extraordinary strength due to the boom’s heavy payload. The fatigue strength must also be high to endure the strain. Microstructures and chemical composition determines the properties of the material. Different defects in the structure can cause failures which places high demands on suppliers and their methods. Test of the fatigue behavior of butt welded joints in high strength steel have been carried out with good results. The steel used in the study was DOMEX 600 DC which is provided by SSAB. The report concludes following statement\textsuperscript{42}:

“The comparison of characteristic curves obtained for each welding condition and the fatigue class (FAT) indicates that, in general, fatigue strength of DOMEX 600 DC steel is significantly higher than fatigue strength recommended by IIW for conventional steels.”

\textsuperscript{40} (Ruukki, 2014)
\textsuperscript{41} See appendix H: Ruukki Efficiency Calculator
\textsuperscript{42} (Costa, Ferreira, & Abreu, 2010)
The assembling of the details in the application demands a certain level of machinability. The material will put demands on the manufacturing tools and equipment. If steel of high strength is selected the tooling must be of high quality. Long plates and high strength requires for instance press brakes with great capacity.

The sustainability aspect of the product needs to be considered. For this purpose the sustainability may be reflected in recyclable material as well as energy efficient manufacturing and production.

Repairability and maintenance are closely connected; the application requires easily accessible components and easily conducted repairing/maintenance to keep the cost down.

The material choice must also offer a lighter construction than the current design in order to reach the objectives of the project. This means two things; the material can either be dense as or even denser than the current material, but have a greater strength or the material must be less dense (lighter weight) than current material but compensate for the possible need for increased dimensions with strength. The report “Experimental study on high strain rate behavior of high strength 600–1000 MPa dual phase steels and 1200 MPa fully martensitic steels” supports the approach of using high strength steel in order to reduce vehicle’s weight and bring cost savings in terms of machinability:

“As one of high grade advanced high strength steels (AHSSs), dual phase (DP) steel sheets and fully martensitic (MS) steel sheets have been successfully used in automotive crash-resistance components for its great benefit in reducing vehicle weight while improving car safety as well as their advantage in cost saving through cold forming instead of hot forming.”

The environmental resistance means the material’s ability to resist all kinds of corrosion. The tough environment the reach stacker is operating in will need certain surface treatment. To achieve the required objectives it is to prefer to use a coating that is not solvent-based. Solvent-based paint demands purchase of emission allowance which is a short-term settlement badly compatible with a sustainable and eco-efficient LCA. Also solvent-based paint means an elevated health risk for the personnel that are

43 (Wang, Li, He, We, Wang, & Du, 2012)
in contact with the solvents. An interesting solution for that is sol-gelnanocoating where the sol-gel process is a method for producing solid materials from small molecules. The book “Corrosion Protection and Control Using Nanomaterials” (2012) describes the coating and the advantages of it:

“Sol–gel coatings are desirable for their environmental friendliness, high performance, compatibility with existing coating application technologies and for the on-demand simplicity of tailoring coating properties.”

Water-based paint is also a more sustainable alternative but this type of coating demands longer drying times which can cause problems in terms of cost efficiency. However, it is worthwhile to review the development of these coatings at regular intervals as they are constantly improved.

10.2 Analysis of Result

A material change where a structural steel of higher strength is selected seems to yield the desired objectives.

A low cost solution may be to use steel with yield strength around 700 MPa. If for example S690QL is used the design can be similar to current design which means that the overall handling do not need extensive and costly changes. The downside is that the possible weight reduction is quite small. The schematic CAD drawings show that the weight reduction of the internal boom will amount up to 28%. A rough estimate of the total amount of material consumption needed if the weight is reduced with 28% is approximately 850-900 ton. The number is based on 130 details and takes into account the 10-12% of details with special design. It is likely to state that the external boom should have a similar appearance due to reason of fitting.

More material can be reduced if a structural steel with ultra-high strength steel is used. Because such an application will mean a deflection of the boom to an unwanted extent reinforcements of the structure to counteract this are needed. Since an appropriate design of these requires closer investigation reinforcements are left out in the concepts.

---

44 See abbreviations.
45 (Pathak & Khanna, 2012)
Ultra-high Strength Steel 1 and 2. However, the simulations show that the internal boom is mostly affected in the one end that is inserted in the external boom. Reinforcements could be placed here to suppress buckling. In concept Ultra-high Strength Steel 3 ribs are supporting the U-shaped design in the most affected areas. Gain in reduced weight seems small. In this project a concept based on using ultra-high strength steel seem to leave too many loose threads in terms of uncertain consequences.
11 Conclusion

In the chapter of Introduction the question below was asked:

- Is a weight reduction motivated?

Based on the documentation that forms the basis for this product development it can be seen that a weight reduction of a heavy duty vehicle is motivated if the purpose is to gain a more cost efficient drive\textsuperscript{46}. This evidence enables a further exploration where the next question that was asked in the introduction receives a response:

- If a motivation for weight reduction is ensured, can the evidence form basis for a relevant concept?

The conclusion is that it seems possible to obtain a telescopic boom with reduced weight but with maintained strength as well as fatigue strength\textsuperscript{47} if the material S690QL is used as structural steel. The weight of the telescopic boom could likely be decreased with 28\%. Such reduction would open possibilities for downsizing the operating components, for example hydraulic cylinders and pumps.

The final concept is chosen from a perspective of safety. Although greater savings likely can be made with more unconventional materials it is precarious to select untested materials where impact assessments have not been thoroughly made\textsuperscript{48}.

Since the selected material and design of the final concept is well proven the risks of unpredictable consequences are presumably rare. The production line will probably need few major changes in the infrastructure; hence the concept will hopefully mean a cost efficient investment.

\textsuperscript{46} (U.S Department of Energy, 2013)
\textsuperscript{47} (Wang, Li, He, We, Wang, & Du, 2012)
\textsuperscript{48} See Appendix E; Technical Gaps Table
12 Discussion

To make a complete product development a more comprehensive work is required where every factor needs to be further investigated. The purpose of this report is that its conclusion can serve as input to such future work.

The reason that composite and aluminum is mentioned in the theory chapter is that those are materials with interesting and good advantages in terms of machinability and light weight. Another type of material interesting for a prospective product development within the field of reducing heavy duty vehicles’ weight are the advanced high strength steels, AHSS\(^49\). The AHSS steel is characterized by its high strength, ductility and energy-absorption which allow thinner components and improved safety if used in vehicles.

CAD programs often provide a function where the program shows where the stress approaches zero and it is therefore possible to cut material in the design in order to reduce weight. This is applicable to all designs.

\(^{49}\)(Kuziak, Kawalla, & Waengler, 2008)
References


http://www.ruukki.com/energy-efficiency-calculator/
Science Direct. (2014). Retrieved 03 03, 2014, from Science Direct:
http://www.sciencedirect.com.proxy.lnu.se/
Appendix A: Nomenclature

**Nomenclature** (units within brackets)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$I_x$</td>
<td>Moment of Inertia, x-axis</td>
<td>mm$^4$</td>
</tr>
<tr>
<td>$I_y$</td>
<td>Moment of Inertia, y-axis</td>
<td>mm$^4$</td>
</tr>
<tr>
<td>$B$</td>
<td>Base dimension, outer</td>
<td>mm</td>
</tr>
<tr>
<td>$b$</td>
<td>Base dimension, inner</td>
<td>mm</td>
</tr>
<tr>
<td>$H$</td>
<td>Height dimension, outer</td>
<td>mm</td>
</tr>
<tr>
<td>$h$</td>
<td>Height dimension, inner</td>
<td>mm</td>
</tr>
<tr>
<td>$A$</td>
<td>Cross sectional area</td>
<td>mm</td>
</tr>
<tr>
<td>$S_y$</td>
<td>Yield strength</td>
<td>N/mm$^2$</td>
</tr>
<tr>
<td>$S_u$</td>
<td>Ultimate strength</td>
<td>N/mm$^2$</td>
</tr>
<tr>
<td>$l_A$</td>
<td>Moment arm, external boom</td>
<td>mm</td>
</tr>
<tr>
<td>$l_B$</td>
<td>Moment arm, internal boom</td>
<td>mm</td>
</tr>
<tr>
<td>$l_H$</td>
<td>Height</td>
<td>mm</td>
</tr>
<tr>
<td>$F_B$</td>
<td>Force, boom</td>
<td>N</td>
</tr>
<tr>
<td>$F_A$</td>
<td>Force, attachment</td>
<td>N</td>
</tr>
<tr>
<td>$F_L$</td>
<td>Force, load</td>
<td>N</td>
</tr>
<tr>
<td>$g$</td>
<td>Acceleration of gravity</td>
<td>m/s$^2$</td>
</tr>
<tr>
<td>$\bar{y}$</td>
<td>Centroidal distance</td>
<td>mm</td>
</tr>
<tr>
<td>$Z$</td>
<td>Section modulus</td>
<td>mm$^3$</td>
</tr>
<tr>
<td>$\sigma_{\text{max}}$</td>
<td>Maximum normal stresses</td>
<td>N/mm$^2$</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Safety factor</td>
<td></td>
</tr>
<tr>
<td>$M$</td>
<td>Bending torque</td>
<td>Nmm</td>
</tr>
</tbody>
</table>
Appendix B: Safety Factor Calculations of the Lifting Boom

Model: SMV 4531 TB5

**Given Data**

Material: S 500 Q

Nominal thickness: 3-50 mm

\[ S_y = 500 \text{ N/mm}^2 \]

\[ S_u = 590-770 \text{ N/mm}^2 \]

\[ F_A = 9500g \text{ N} \]

\[ F_B = 5240g \text{ N} \]

\[ (5309 + 3725) \times 0.58 \times g = 5240g \text{ N} \]

Where 0.58 (58%) is the percentage of the boom weight distributed in front of the supporting pistons. This will constitute part of the total torque affecting the boom.

**Formulas**

- Moment of Inertia, \( I \)

\[ I = \frac{BH^3 - bh^3}{12} \]

- Section Modulus, \( Z \)

\[ Z = \frac{I}{\bar{y}} \]

- Bending Torque, \( M \)

\[ M = F \times l \]

- Maximum Normal Stress, \( \sigma_{max} \)

\[ \sigma_{max} = \frac{M}{Z} \]
• Safety Factor, $\eta$

$$\eta = \frac{S_y}{\sigma_{max}}$$

The most affected positions are dimensioning. For the safety factor calculations following information is given:

<table>
<thead>
<tr>
<th>Current State Figures</th>
<th>External</th>
<th>Internal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Section Modulus, $Z$</td>
<td>24474672 mm$^3$</td>
<td>14217167 mm$^3$</td>
</tr>
<tr>
<td>Centroidal distance, $\bar{y}$</td>
<td>435 mm</td>
<td>385 mm</td>
</tr>
<tr>
<td>Outer base dimension, $B$</td>
<td>880 mm</td>
<td>800 mm</td>
</tr>
<tr>
<td>Inner base dimension, $b$</td>
<td>850 mm</td>
<td>770 mm</td>
</tr>
<tr>
<td>Outer height dimension, $H$</td>
<td>870 mm</td>
<td>770 mm</td>
</tr>
<tr>
<td>Inner height dimension, $h$</td>
<td>810 mm</td>
<td>730</td>
</tr>
<tr>
<td>Top/bottom plate thickness, $t$</td>
<td>30 mm</td>
<td>20 mm</td>
</tr>
<tr>
<td>Side plate thickness, $t$</td>
<td>15 mm</td>
<td>15 mm</td>
</tr>
</tbody>
</table>
Safety Factor Calculations

- External Boom

• Moment of Inertia

\[
I_y = \frac{BH^3 - bh^3}{12} = \frac{880 \times 870^3 - 850 \times 810^3}{12} = 10646482500 \text{ } mm^4
\]

\[
I_x = \frac{BH^3 - bh^3}{12} = \frac{870 \times 880^3 - 810 \times 850^3}{12} = 795328250 \text{ } mm^4
\]

• Centroidal Distance

\[
\bar{y}_{top} = \bar{y}_{bottom} = \bar{y} = 435 \text{ } mm
\]

• Cross Section Area

\[
A = 2(30 \times 880) + 2(15 \times (870 - 60)) = 77100 \text{ } mm^2
\]

• Section Modulus

\[
Z_E = \frac{I_y}{\bar{y}} = \frac{10646482500}{435} = 24474672 \text{ } mm^3
\]

1st Row (45 ton):

Prerequisites:

\[
I_A = 6800 \text{ } mm
\]

\[
I_H = 14900 \text{ } mm
\]

\[
F_L = 45000g \text{ } N
\]

\[
F_{tot} = F_B + F_A + F_L = (5240+9500+45000)g = 586049 \text{ } N
\]

• Bending Torque

\[
M = F_{tot} \times I_A = 586049 \times 6800 = 3,985133 \times 10^9 \text{ } Nmm
\]

• Maximum Normal Stress

\[
\sigma_{max} = \frac{M}{Z_E} = \frac{3,985133 \times 10^9}{24474672} = 162,827 \text{ } N/mm^2
\]
• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{max}} = \frac{500}{162,827} = 3,071 \]

\[ \eta_u = \frac{S_u}{\sigma_{max}} = \frac{590}{163,536} = 3,623 \]

2nd Row (31 ton):

**Prerequisites:**

- \( l_A = 8100 \text{ mm} \)
- \( l_H = 13200 \text{ mm} \)
- \( F_L = 31000g \text{ N} \)

\[ F_{tot} = F_B + F_A + F_L = (5240+9500+31000)g = 448709 \text{ N} \]

• Bending Torque

\[ M = F_{tot} \cdot l_A = 451260 \cdot 8100 = 3,634546 \cdot 10^9 \text{ Nmm} \]

• Maximum Normal Stress

\[ \sigma_{max} = \frac{M}{Z_E} = \frac{3,634546 \cdot 10^9}{24474672} = 148,502 \text{ N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{max}} = \frac{500}{148,502} = 3,367 \]

\[ \eta_u = \frac{S_u}{\sigma_{max}} = \frac{590}{148,502} = 3,973 \]

3rd Row (16 ton)

**Prerequisites:**

- \( l_A = 9800 \text{ mm} \)
- \( l_H = 10300 \text{ mm} \)

\( F_L = 16000g \text{ N} \)

\[ F_{tot} = F_B + F_A + F_L = (5240+9500+16000)g = 301559 \text{ N} \]
• Bending Torque

\[ M = F_{tot} \times l_A = 301559 \times 9800 = 2,955278 \times 10^9 \text{Nm} \]

• Maximum Normal Stress

\[ \sigma_{\text{max}} = \frac{M}{Z_E} = \frac{2,980278 \times 10^9}{24474672} = 120,748 \text{N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{500}{120,748} = 4.141 \]

\[ \eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{590}{120,748} = 4.886 \]

Transport Position (45 ton)

**Prerequisites:**

\[ l_A = 4400 \text{ mm} \]
\[ l_H = 6400 \text{ mm} \]
\[ F_L = 45000 \text{ g N} \]
\[ F_{tot} = 586049 \text{ N} \]

• Bending Torque

\[ M = F_{tot} \times l_A = 586049 \times 4400 = 2,578616 \times 10^9 \text{Nm} \]

• Maximum Normal Stress

\[ \sigma_{\text{max}} = \frac{M}{Z_E} = \frac{2,578616 \times 10^9}{24474672} = 105,359 \text{N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{500}{105,359} = 4.746 \]

\[ \eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{590}{121,770} = 5.600 \]
- Internal Boom

• Moment of Inertia

\[
I_y = \frac{BH^3 - bh^3}{12} = \frac{800 \times 770^3 - 770 \times 730^3}{12} = 5473609167 \text{ mm}^4
\]

\[
I_x = \frac{BH^3 - bh^3}{12} = \frac{770 \times 800^3 - 730 \times 770^3}{12} = 5080909167 \text{ mm}^4
\]

• Centroidal Distance

\[
y_{top} = y_{bottom} = \bar{y} = 385 \text{ mm}
\]

• Cross Sectional Area

\[
A = 2(20 \times 800) + 2(15 \times (770 - 40)) = 53900 \text{ mm}
\]

• Section Modulus

\[
Z_l = \frac{I_y}{\bar{y}} = \frac{5473609167}{385} = 14217167 \text{ mm}^3
\]

1st Row (45 ton)

Prerequisites:

\[
l_B = 4800 \text{ mm}
\]

\[
l_H = 14900 \text{ mm}
\]

\[
F_L = 45000 \text{ g N}
\]

\[
F_{tot} = F_B + F_A + F_L = (5240 + 9500 + 45000) \text{ g} = 586049 \text{ N}
\]

• Bending Torque

\[
M = F_{tot} \times l_B = 588600 \times 4800 = 2,813035 \times 10^9 \text{ Nmm}
\]

• Maximum Normal Stress

\[
\sigma_{max} = \frac{M}{Z_l} = \frac{2,813035 \times 10^9}{14217167} = 197,862 \text{ N/mm}^2
\]

• Safety Factor

\[
\eta_y = \frac{S_y}{\sigma_{max}} = \frac{500}{198,723} = 2,527
\]

\[
\eta_u = \frac{S_u}{\sigma_{max}} = \frac{590}{163,536} = 2,982
\]
2nd Row (31 ton):

**Prerequisites:**

\[ l_B = 5600 \text{ mm} \]
\[ l_H = 13200 \text{ mm} \]
\[ F_L = 31000 \text{ g N} \]

\[ F_{\text{tot}} = F_B + F_A + F_L = (5240 + 9500 + 31000) \text{ g} = 448709 \text{ N} \]

- **Bending Torque**

\[ M = F_{\text{tot}} \times l_B = 448709 \times 5600 = 2,512770 \times 10^9 \text{ Nmm} \]

- **Maximum Normal Stress**

\[ \sigma_{\text{max}} = \frac{M}{Z_I} = \frac{2,512770 \times 10^9}{14217167} = 176,742 \text{ N/mm}^2 \]

- **Safety Factor**

\[ \eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{500}{176,742} = 2.829 \]

\[ \eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{590}{176,742} = 3.338 \]

3rd Row (16 ton)

**Prerequisites:**

\[ l_B = 6600 \text{ mm} \]
\[ l_H = 10300 \text{ mm} \]
\[ F_L = 16000 \text{ g N} \]

\[ F_{\text{tot}} = F_B + F_A + F_L = (5240 + 9500 + 16000) \text{ g} = 301559 \text{ N} \]

- **Bending Torque**

\[ M = F_{\text{tot}} \times l_A = 301559 \times 6600 = 1,990289 \times 10^9 \text{ Nmm} \]

- **Maximum Normal Stress**

\[ \sigma_{\text{max}} = \frac{M}{Z_I} = \frac{1,990289 \times 10^9}{14217167} = 139,992 \text{ N/mm}^2 \]
• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{max}} = \frac{500}{141,176} = 3,572 \]
\[ \eta_u = \frac{S_u}{\sigma_{max}} = \frac{590}{141,176} = 4,215 \]

*Transport Position (45 ton)*

**Prerequisites:**

- \( l_B = 1000\text{mm} \)
- \( l_H = 6400 \text{mm} \)
- \( F_L = 45000\text{g N} \)
- \( F_{tot} = 586049 \text{N} \)

• Bending Torque

\[ M = F_{tot} \times l_B = 586049 \times 1000 = 0.586049 \times 10^9 \text{Nmm} \]

• Maximum Normal Stress

\[ \sigma_{max} = \frac{M}{Z_i} = \frac{0.586049 \times 10^9}{14217167} = 41,221 \text{N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{max}} = \frac{500}{41,221} = 12,130 \]
\[ \eta_u = \frac{S_u}{\sigma_{max}} = \frac{590}{41,221} = 14,313 \]
Appendix C: Calculations of the Lifting Boom – Concept 1

Material: S 690 QL (Weldox 700)
Nominal thickness: 3-50 mm

\[ S_y = 690 \text{ N/mm}^2 \]
\[ S_u = 770-940 \text{ N/mm}^2 \]

\[ F_A = 9500 \text{ g N} \]
\[ F_B = 3760 \text{ g N} \]

\[ (3831,5 + 2651,4) \times 0,58 \times g = 3760 \text{ g N} \]

<table>
<thead>
<tr>
<th>Concept 1 Figures</th>
<th>External</th>
<th>Internal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Section Modulus, Z</td>
<td>18501763 mm³</td>
<td>10638892 mm³</td>
</tr>
<tr>
<td>Centroidal distance, ( \bar{y} )</td>
<td>435 mm</td>
<td>385 mm</td>
</tr>
<tr>
<td>Outer base dimension, B</td>
<td>880 mm</td>
<td>800 mm</td>
</tr>
<tr>
<td>Inner base dimension, b</td>
<td>860 mm</td>
<td>780 mm</td>
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<tr>
<td>Outer height dimension, H</td>
<td>870 mm</td>
<td>770 mm</td>
</tr>
<tr>
<td>Inner height dimension, h</td>
<td>825 mm</td>
<td>740 mm</td>
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<td>Top/bottom plate thickness, t</td>
<td>22,5 mm</td>
<td>15 mm</td>
</tr>
<tr>
<td>Side plate thickness, t</td>
<td>10 mm</td>
<td>10 mm</td>
</tr>
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</table>
- External Boom

- Moment of Inertia

\[
I_y = \frac{BH^3 - bh^3}{12} = \frac{880 \times 870^3 - 860 \times 825^3}{12} = 8048266875 \text{ mm}^4
\]

\[
I_x = \frac{BH^3 - bh^3}{12} = \frac{870 \times 880^3 - 825 \times 860^3}{12} = 5677870000 \text{ mm}^4
\]

- Centroidal Distance

\[
\tilde{y}_{\text{top}} = \tilde{y}_{\text{bottom}} = \tilde{y} = 435 \text{ mm}
\]

- Cross Section Area

\[
A = 2(22.5 \times 880) + 2(10 \times (870 - 45)) = 56100 \text{ mm}^2
\]

- Section Modulus

\[
Z_E = \frac{I_y}{\tilde{y}} = \frac{8048266875}{435} = 18501763 \text{ mm}^3
\]

1st Row (45 ton):

Prerequisites:

-\( l_A = 6800 \text{ mm} \)
-\( l_H = 14900 \text{ mm} \)
-\( F_L = 45000 \text{ g N} \)

\[
F_{\text{tot}} = F_B + F_A + F_L = (3760 + 9500 + 45000) \text{ g} = 571531 \text{ N}
\]

- Bending Torque

\[
M = F_{\text{tot}} \times l_A = 571531 \times 6800 = 3,88641 \times 10^9 \text{ Nmm}
\]

- Maximum Normal Stress

\[
\sigma_{\text{max}} = \frac{M}{Z_E} = \frac{3,88641 \times 10^9}{18501763} = 210,06 \text{ N/mm}^2
\]
• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{max}} = \frac{690}{210,06} = 3,28 \]

\[ \eta_u = \frac{S_u}{\sigma_{max}} = \frac{770}{215,390} = 3,67 \]

2nd Row (31 ton):

Prerequisites:

\( l_A = 8100 \text{ mm} \)
\( l_H = 13200 \text{ mm} \)
\( F_L = 31000g \text{ N} \)

\[ F_{tot} = F_B + F_A + F_L = (3760+9500+31000)g = 434191 \text{ N} \]

• Bending Torque

\[ M = F_{tot} * l_A = 434191 * 8100 = 3,516944 \times 10^9 \text{ Nmm} \]

• Maximum Normal Stress

\[ \sigma_{max} = \frac{M}{Z_E} = \frac{3,516944 \times 10^9}{18501763} = 190,09 \text{ N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{max}} = \frac{690}{190,09} = 3,63 \]

\[ \eta_u = \frac{S_u}{\sigma_{max}} = \frac{770}{190,09} = 4,05 \]

3rd Row (16 ton)

Prerequisites:

\( l_A = 9800 \text{ mm} \)
\( l_H = 10300 \text{ mm} \)
\( F_L = 16000g \text{ N} \)

\[ F_{tot} = F_B + F_A + F_L = (3760+9500+16000)g = 287041 \text{ N} \]
• Bending Torque

\[ M = F_{\text{tot}} \cdot l_A = 287041 \cdot 9800 = 2,813000 \cdot 10^9 \text{ Nmm} \]

• Maximum Normal Stress

\[ \sigma_{\text{max}} = \frac{M}{Z_E} = \frac{2,813000 \cdot 10^9}{18501763} = 152,04 \text{ N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{690}{152,04} = 4,54 \]
\[ \eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{770}{161,081} = 5,06 \]

Transport Position (45 ton)

Prerequisites:
\l_a = 4400 \text{ mm} \\
\l_H = 6400 \text{ mm} \\
F_L = 45000g \text{ N}

\[ F_{\text{tot}} = 571531 \text{ N} \]

• Bending Torque

\[ M = F_{\text{tot}} \cdot l_A = 571531 \cdot 4400 = 2,514736 \cdot 10^9 \text{ Nmm} \]

• Maximum Normal Stress

\[ \sigma_{\text{max}} = \frac{M}{Z_E} = \frac{2,514736 \cdot 10^9}{18501763} = 135,92 \text{ N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{690}{135,92} = 5,08 \]
\[ \eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{770}{139,371} = 5,67 \]
- Internal Boom

- Moment of Inertia

\[
I_y = \frac{BH^3 - bh^3}{12} = \frac{800 \times 770^3 - 780 \times 740^3}{12} = 4095973333 \text{ mm}^4
\]

\[
I_x = \frac{BH^3 - bh^3}{12} = \frac{770 \times 800^3 - 740 \times 780^3}{12} = 3589293333 \text{ mm}^4
\]

- Centroidal Distance

\[\hat{y}_{top} = \hat{y}_{bottom} = \hat{y} = 385 \text{ mm}\]

- Cross Sectional Area

\[A = 2(15 \times 800) + 2(10 \times (770 - 30)) = 38800 \text{ mm}\]

- Section Modulus

\[Z_I = \frac{I_y}{\hat{y}} = \frac{4095973333}{385} = 10638892 \text{ mm}^3\]

1st Row (45 ton)

**Prerequisites:**

- \(l_B = 4800 \text{ mm}\)
- \(l_H = 14900 \text{ mm}\)
- \(F_L = 45000 \text{ g N}\)

\[F_{tot} = F_B + F_A + F_L = (3760 + 9500 + 45000) \text{ g} = 571531 \text{ N}\]

- Bending Torque

\[M = F_{tot} \times l_B = 571531 \times 4800 = 2,743347 \times 10^9 \text{ Nmm}\]

- Maximum Normal Stress

\[
\sigma_{\text{max}} = \frac{M}{Z_I} = \frac{2,743347 \times 10^9}{10638892} = 257,86 \text{ N/mm}^2
\]

- Safety Factor

\[
\eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{690}{257,86} = 2,68
\]

\[
\eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{770}{257,86} = 2,99
\]
2nd Row (31 ton):

**Prerequisites:**
- $l_B = 5600 \text{ mm}$
- $l_H = 13200 \text{ mm}$
- $F_L = 31000\text{g N}$

$F_{tot} = F_B + F_A + F_L = (3760+9500+31000)g = 434191 \text{ N}$

- Bending Torque
  
  $$M = F_{tot} \cdot l_B = 434191 \cdot 5600 = 2,431467 \cdot 10^9 \text{ Nmm}$$

- Maximum Normal Stress
  
  $$\sigma_{max} = \frac{M}{Z_l} = \frac{2,431467 \cdot 10^9}{10638892} = 228,55 \text{ N/mm}^2$$

- Safety Factor
  
  $$\eta_y = \frac{S_y}{\sigma_{max}} = \frac{690}{228,55} = 3,02$$
  
  $$\eta_h = \frac{S_u}{\sigma_{max}} = \frac{770}{236,187} = 3,37$$

3rd Row (16 ton)

**Prerequisites:**
- $l_B = 6600 \text{ mm}$
- $l_H = 10300 \text{ mm}$
- $F_L = 16000\text{g N}$

$F_{tot} = F_B + F_A + F_L = (3760+9500+16000)g = 287041 \text{ N}$

- Bending Torque
  
  $$M = F_{tot} \cdot l_A = 287041 \cdot 6600 = 1,894468 \cdot 10^9 \text{ Nmm}$$
• Maximum Normal Stress

\[ \sigma_{\text{max}} = \frac{M}{Z_l} = \frac{1,894468 \times 10^9}{10638892} = 178,07 \text{ N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{690}{187,077} = 3,87 \]

\[ \eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{770}{187,077} = 4,32 \]

*Transport Position (45 ton)*

**Prerequisites:**

- \( I_B = 1000 \text{mm} \)
- \( I_H = 6400 \text{ mm} \)
- \( F_L = 45000 \text{g N} \)
- \( F_{tot} = 571531 \text{ N} \)

• Bending Torque

\[ M = F_{tot} \times l_B = 571531 \times 1000 = 0,571531 \times 10^9 \text{ Nmm} \]

• Maximum Normal Stress

\[ \sigma_{\text{max}} = \frac{M}{Z_l} = \frac{0,571531 \times 10^9}{10638892} = 53,72 \text{ N/mm}^2 \]

• Safety Factor

\[ \eta_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{690}{53,72} = 12,84 \]

\[ \eta_u = \frac{S_u}{\sigma_{\text{max}}} = \frac{770}{53,72} = 14,33 \]
# Appendix D: House of Quality

## Title:
Weight Reduction of Rear Stabilizer

## Author:
[Redacted]

## Date:
[Redacted]

## Notes:
[Redacted]

## Diagram

### Quality Characteristics
- Key Performance Indicators (KPIs)
- Customer Expectations

### Demanded Quality
- Functionality
- Durability
- Ease of Use
- Aesthetics

### Performance Requirements
- Reliability
- Safety
- Efficiency
- Cost

### Target or Ideal Values
- Weight
- Torsional Rigidity

### Difficulties
- (0) Easy to Achieve
- (1) Slightly Difficult
- (2) Difficult
- (3) Very Difficult

### Weight/Importance
- 112.5
- 132.5
- 232
- 332
- 34.5
- 62.5
- 115
- 135
- 155
- 175
- 45
- 90
- 135
- 180

### Relative Weight
- 6.7
- 11.9
- 4.5
- 6.5
- 1.6
- 1.7
- 5
- 6.4
- 6.6
- 5.7
- 3.3
- 1.2
- 2.3
## Appendix E: Failure Mode and Effects Analysis

<table>
<thead>
<tr>
<th>Project Leader</th>
<th>Date (YY/MM/DD)</th>
<th>Follow-up</th>
<th>Remarks</th>
<th>Process-FMECA</th>
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<td>Linda Ekdahl Norling</td>
<td>2014/05/05</td>
<td>2014/05/24</td>
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<th>Nr</th>
<th>Concept</th>
<th>Application</th>
<th>Possible failure</th>
<th>Failure effect</th>
<th>Failure reason</th>
<th>Probability of occurrence of failure</th>
<th>Severity of failure</th>
<th>Likelihood of detecting the failure</th>
<th>Risk Priority Number RPN</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>High Strength Steel with Rectangular Design (S690QL)</td>
<td>Telescopic Boom</td>
<td>The boom is deflecting too much.</td>
<td>Leads to fatigue and/or unstable drive.</td>
<td>Low degree of Rigidity.</td>
<td>2</td>
<td>2</td>
<td>2</td>
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<td>The boom is not enough stable.</td>
<td>Hazardous drive due to swaying.</td>
<td>Design that does not suit the purpose.</td>
<td>3</td>
<td>3</td>
<td>1</td>
<td>9</td>
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<td></td>
<td>The boom is not light enough.</td>
<td>No gain.</td>
<td>Product Development did not lead to wanted results.</td>
<td>4</td>
<td>1</td>
<td>1</td>
<td>4</td>
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<tr>
<td></td>
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<td></td>
<td>The boom does not resist fatigue.</td>
<td>Boom may break when loaded.</td>
<td>The material is not enough resistant.</td>
<td>2</td>
<td>5</td>
<td>5</td>
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<td>Issue</td>
<td>Failure Mode</td>
<td>Root Cause</td>
<td>RPN (Risk Priority Number)</td>
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<td>The boom is not enduring the stress.</td>
<td>Boom breaks when loaded.</td>
<td>The design failed; too weak</td>
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<td>weak design.</td>
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<td>The boom is subjected to cracking weld joints.</td>
<td>Boom may break when loaded.</td>
<td>Insufficient welding/welding</td>
<td>75</td>
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<td>The boom is subjected to buckling.</td>
<td>Boom may break when loaded.</td>
<td>High concentration of stress in</td>
<td>48</td>
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<td>certain points/too weak design.</td>
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Total Value of RPN | **214**
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<th>Possible failure</th>
<th>Failure effect</th>
<th>Failure reason</th>
<th>Probability of occurrence of failure</th>
<th>Severity of failure</th>
<th>Likelihood of detecting the failure</th>
<th>Risk Priority Number RPN</th>
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<tbody>
<tr>
<td>2.</td>
<td>Ultra High Strength Steel with Rectangular Shape (S890QL)</td>
<td>Telescopic Boom</td>
<td>The boom is deflecting too much.</td>
<td>Leads to fatigue and/or unstable drive.</td>
<td>Low degree of Rigidity.</td>
<td>4</td>
<td>2</td>
<td>2</td>
<td>16</td>
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<td>The boom is not enough stable.</td>
<td>Hazardous drive due to swaying.</td>
<td>Design that does not suit the purpose.</td>
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<td>The boom is not light enough.</td>
<td>No gain.</td>
<td>Product Development lead to wanted results.</td>
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<td>The boom is not enduring the stress.</td>
<td>Boom breaks when loaded.</td>
<td>The design failed; too weak structural steel and/or too weak design.</td>
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<td>Boom may break when loaded.</td>
<td>Insufficient welding/welding methods.</td>
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<td>Failure reason</td>
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<td>Severity of failure</td>
<td>Likelihood of detecting the failure</td>
<td>Risk Priority Number RPN</td>
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<td>2.</td>
<td>Ultra-high Strength Steel with U-shaped Design (S890QL)</td>
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<td>The boom is deflecting too much.</td>
<td>Leads to fatigue and/or unstable drive.</td>
<td>Low degree of Rigidity.</td>
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<td>The boom is not enough stable.</td>
<td>Hazardous drive due to swaying.</td>
<td>Design that does not suit the purpose.</td>
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<td>The boom is not enduring the stress.</td>
<td>Boom breaks when loaded.</td>
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<td>---</td>
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<td>----</td>
<td></td>
<td></td>
</tr>
<tr>
<td>The boom is subjected to buckling.</td>
<td>Boom may break when loaded.</td>
<td>High concentration of stress in certain points/too weak design.</td>
<td>4</td>
<td>4</td>
<td>3</td>
<td>48</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Total Value of RPN</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td><strong>246</strong></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
## Key Technical Gaps for Materials for Trucks and HDVs

<table>
<thead>
<tr>
<th>Structural Materials</th>
<th>Weight Reduction Potential</th>
<th>Three Most Significant Technical Gaps Impeding Widespread Implementation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon Fiber Composites</td>
<td>50–70%</td>
<td>Lack of low-cost precursors and energy efficient conversion processes for carbon fiber</td>
</tr>
<tr>
<td>Magnesium</td>
<td>30–70%</td>
<td>Cost effective, environmentally friendly process for magnesium production does not exist.</td>
</tr>
<tr>
<td>Aluminum</td>
<td>30–60%</td>
<td>Processes for joining Al alloys to dissimilar materials and welding of 7000 series Al are inadequate.</td>
</tr>
<tr>
<td>Glass Fiber Composites</td>
<td>25–35%</td>
<td>Lack of technologies to improve properties</td>
</tr>
<tr>
<td>Advanced High Strength Steels</td>
<td>10–30%</td>
<td>Understanding of structure/property relationships is insufficient to guide development of improved properties</td>
</tr>
<tr>
<td>Steel and Cast Iron (Propulsion)</td>
<td>0-15%</td>
<td>Manufacturing processes (forging, casting, etc.) are not cost effective or are inadequate</td>
</tr>
</tbody>
</table>

51 (U.S Department of Energy, 2013)
### History of Konecranes Lifttrucks AB

<table>
<thead>
<tr>
<th>Year</th>
<th>Event</th>
</tr>
</thead>
<tbody>
<tr>
<td>1947</td>
<td>Founding of SMV (Silverdalens Mekaniska Verkstad).</td>
</tr>
<tr>
<td>1959</td>
<td>Production and delivery of our first fork lift truck.</td>
</tr>
<tr>
<td>1994</td>
<td>Appointment of new management. The platform of the reach stacker is developed.</td>
</tr>
<tr>
<td>1995</td>
<td>Opening of a new production plant in Markaryd, Småland, southern Sweden. A complete new truck range was introduced, including fork lifts, container lift trucks and reach stackers.</td>
</tr>
<tr>
<td>1998</td>
<td>Delivery of the world’s largest reach stacker for handling containers on river barges. Factory and office extension.</td>
</tr>
<tr>
<td>2000</td>
<td>Introduction of a new fork lift truck concept, the SPECTRA range, 10-16 tons and 18-25 tons. Introduction of a new range of empty container handlers, stacking 5-8 high, 9 tons capacity, single and double stacking.</td>
</tr>
<tr>
<td>2004</td>
<td>SMV became part of KCI Konecranes of Finland through acquisition. SMV became a business unit of Konecranes with the legal name Konecranes Lifttrucks AB. Introduction of a new range of reach stackers for laden container handling &amp; intermodal handling, the TB &amp; CB series.</td>
</tr>
<tr>
<td>2006</td>
<td>KCI Konecranes launched a new global master brand strategy and identity, and dropped &quot;KCI&quot; from the brand name. Konecranes Lift Trucks business unit introduced a new range of 10-16 ton fork lift trucks with higher performance and a new range of empty container handlers with 4-6 high single stacking and 8 tons capacity. Factory and offices were expanded for an increased annual production capacity.</td>
</tr>
<tr>
<td>2007</td>
<td>Production facility was opened in Lingang (Shanghai), China.</td>
</tr>
<tr>
<td>2011</td>
<td>Major investment in facilities in Markaryd, Sweden to meet growing demands.</td>
</tr>
<tr>
<td>2013</td>
<td>Introduction of the world’s first hybrid reach stacker, the SMV 4531 TB5 HLT for container handling, with a lifting capacity of 45 tons. Konecranes and Linde Material Handling sign agreement in container handling lift truck business.</td>
</tr>
</tbody>
</table>
Appendix H: Ruukki Efficiency Calculator

Input:

<table>
<thead>
<tr>
<th>Waste compactor</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle class</td>
<td>Waste collection vehicle (selected because of its with reach stacker similar driving pattern; many starts and stops</td>
</tr>
<tr>
<td>Driving distance</td>
<td>2000 km/year</td>
</tr>
<tr>
<td>Operational life</td>
<td>20 years</td>
</tr>
<tr>
<td>Weight</td>
<td>7000 kg (old: 10000 kg)</td>
</tr>
<tr>
<td>Urban driving kilometers of total</td>
<td>0,0%</td>
</tr>
<tr>
<td>Empty load kilometers</td>
<td>45,0%</td>
</tr>
<tr>
<td>Fuel price (€/liter)</td>
<td>1,51 € (Swedish current price)</td>
</tr>
<tr>
<td>Fuel savings due to lower air resistance</td>
<td>0,0%</td>
</tr>
</tbody>
</table>

Conclusion: Same Payload

<table>
<thead>
<tr>
<th>Product Life Cycle</th>
<th>New Design</th>
<th>Old Design</th>
<th>Savings</th>
<th>Savings %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel Consumption (liter)</td>
<td>72082</td>
<td>74400</td>
<td>2318</td>
<td>3,12</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Product Life Cycle</th>
<th>New Design</th>
<th>Old Design</th>
<th>Savings</th>
<th>Savings %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel Consumption costs</td>
<td>108916</td>
<td>112418</td>
<td>3502</td>
<td>3,12</td>
</tr>
</tbody>
</table>

The calculation tool is based on the content of the following references and homepages.

WTT Report Version 2c March 2007

LIPASTO: http://lipasto.vtt.fi
TEAM: : http://www.ecobalance.com
IPCC: : http://ipcc.ch/ipccreports/assessments-reports.htm
WRI: : http://www.wri.com
ISWA: : http://www.iswa.org
Appendix I: Truconnect Logged Data

Readout Truconnect
Machine ID: S/n C20540_14842CN
Location: TPT, Thailand
Data logged: 2013-07-26 – 2014-02-14

Load Collective

SMV4531TB5, s/n C20540. Load Collective. (2013-07-26 -- 2014-02-14)
Number of Containers per Day

SMV4531TB5, s/n C20540. No of Containers / day. (2013-07-26 -- 2014-02-14)
Total Load per Day

SMV4531TB5, s/n C20540. Total Load/ day. (2013-07-26 -- 2014-02-14)