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**Abstract**

This document describes the sub systems of the Hermes bulk wagon, different ways to minimize its tare weight and the potential weight reduction.

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**Executive Summary**

In the Hermes project, a new railroad wagon will be developed. A major goal is to increase the load capacity by lowering the tare weight by 25-30 % compared to similar wagons and to increase the load capacity with >50 %. This report represents deliverable 1.1 of the project in which the task is to investigate ways of achieving this weight reduction by taking full advantage of advanced high strength steels (AHSS).

As a reference wagon the current version of Kiruna Wagon’s Side Dumper is used. It is designed for 25 ton axle load, 8 ton/m line load and 1 435 mm track gauge. The wagon tare weight is 28.6 ton and the load capacity 71.5 ton. A tare weight reduction of 25-30 % in this case means about 7 - 8.5 ton and a load capacity increase of 50 % means 36 ton.

Based on Kiruna Wagon’s previous experience in working with weight reduction of railroad wagons and SSAB’s experience with AHSS, several different ways of minimizing the tare weight have been found and investigated. Of these, some has higher potential and are more possible to actualize. Based on this, the following are chosen for further investigations in the project: weight reduction through general design, through smart use of advanced high strength steels (AHSS), by taking vibration dampening into account, by minimizing over dimensioning, and finally by using reducing the life span of selected parts of the wagon.

The total potential weight reduction through these methods is estimated to an astonishing 9.8 ton or 34 %, rendering in a tare weight of about 19 ton, the largest weight reductions stemming from redesigning the wagon to a more flexible, modular version and by a more extensive use of AHSS. This means that the weight reduction goal should be well in reach.

Since the total weight of the EU wagon is set, the load capacity can only be increased with the amount the tare weight is reduced, i.e. 9.8 ton, resulting in a load capacity increase of about 22 %. In other words, the load capacity goal of 50% reduction initially set in the project proposal has been shown by this study as not possible to reach.

Within the project, a prototype wagon will be made to suit the needs of Iberpotash Catalonia. In that case, the plan is to increase the permissible axle and line load of the tracks, which allows heavier wagons and a load capacity increase up to 100 % per wagon meter, i.e. well beyond the goal. The tare weight reduction goal of 25-30 % for the new wagons may however be a challenge since they have to be made much stronger than the current wagons in order for them to be able to handle the dramatically increased weight of the load.

With this report, the tasks and objectives of deliverable 1.1 have been reached.
1 Background

Within the Hermes project, a wagon for granular material will be developed for EU and Iberpotash respectively. A main objective is to increase the payload capacity by minimizing the wagon tare weight. In an internal report, Kiruna Wagon’s Side Dumper wagon type has been selected as being the most suitable type of wagon for the project. Kiruna Wagon has previously developed a first as well as a lighter second generation of the wagon type. In the Hermes project, a modular third generation of the wagon type will be developed.

The main objective of Hermes Work package 1 is to find methods to reduce the tare weight of bulk wagons with 25-30%. The goal is also to increase the load capacity with more than 50% per meter wagon. The weight reductions should be done by taking full advantage of AHSS, composites and joining techniques to meet the mechanical and corrosion requirements.

2 Introduction

This report represents deliverable 1.1 of the Hermes project wherein it is explored how to take full advantage of AHSS for the weight reduction of the wagon.

2.1 EU wagon

In this report, the current version of Kiruna Wagon’s Side Dumper wagon is, in general, used as a reference. This reference wagon is built for normal railroad gauge in EU, 1 435 mm. It has a tare weight of 28.5 tons and is designed for 25 ton axle load and 8 ton/m line load with a resulting length of 12.5 m. In the table below, the tare weight reduction goal is calculated. This weight reduction results in a corresponding increase in load capacity, which is also calculated.
Table 1: Goal weight reduction and resulting increase of load capacity of the EU wagon

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Max axle load</td>
<td>25 ton/axle</td>
</tr>
<tr>
<td>2</td>
<td>Number of axles</td>
<td>4</td>
</tr>
<tr>
<td>3</td>
<td>Max wagon weight (1 x 2)</td>
<td>100 ton</td>
</tr>
<tr>
<td>4</td>
<td>Max line load</td>
<td>8 ton/m</td>
</tr>
<tr>
<td>5</td>
<td>Minimum wagon length (3 / 4)</td>
<td>12.5 m</td>
</tr>
<tr>
<td>6</td>
<td>Current tare weight</td>
<td>28.5 ton</td>
</tr>
<tr>
<td>7</td>
<td>Current load capacity (3 - 6)</td>
<td>71.5 ton</td>
</tr>
<tr>
<td>8</td>
<td>Current load capacity (7 / 5)</td>
<td>5.7 ton/m</td>
</tr>
<tr>
<td>9</td>
<td>Weight reduction goal (25-30 % x 6)</td>
<td>7.125 - 8.55 ton</td>
</tr>
<tr>
<td>10</td>
<td>Goal tare weight (6 - 8)</td>
<td>20 - 21.4 ton</td>
</tr>
<tr>
<td>11</td>
<td>New load capacity (3 - 10)</td>
<td>78.6 - 80 ton</td>
</tr>
<tr>
<td>12</td>
<td>New load capacity (11 / 5)</td>
<td>6.3 – 6.4 ton/m</td>
</tr>
<tr>
<td>13</td>
<td>Load capacity increase (11 – 7)</td>
<td>7.1 – 8.5 ton</td>
</tr>
<tr>
<td>14</td>
<td>Load capacity increase (13 / 7)</td>
<td>10 – 12 %</td>
</tr>
</tbody>
</table>

The goal to reduce the tare weight with 7-8.5 tons seems possible. This would render in an increased load capacity of the same amount. Counted per meter wagon, the load capacity would increase from 5.7 to 6.4 ton/m, or 10-12 %, i.e. the same as for the whole wagon.

In this case, the maximum allowed axle load and maximum total wagon weight as well as the maximum allowed line load is set. This means that the minimum wagon length is also set. Therefore the increase in load capacity is strictly limited to the same amount as the reduction of the tare weight. This is far less than the goal to increase the load capacity with >50 %.

In order to reach >50 % increased load capacity, the tare weight would have to be reduced with more than 71.5 x 50 % = 36 ton, which is more than the original tare weight of the wagon, and hence impossible. The only way to reach such a dramatic increase in load capacity would be to increase the maximum allowed axle load, which is not within the scope of the current project.

**2.2 Iberpotash wagon**

In the Iberpotash case, however, such a change is planned. The current maximum allowed axle load of the tracks is 15 ton/axle. This is planned to be increased to 18 ton/axle. At the same time, the maximum allowed line load is planned to be increased from 5 to 8 ton/m. This improvement of the tracks allows for a dramatically increase of load capacity in addition to the increased load capacity that the planned tare
weight reduction allows. The table below shows a calculation of the goal tare weight as well as the increased load capacity due to the combined effect of the new tare weight and increased track capacity.

Table 2: Increased load capacity of Iberpotash’s wagons as a result of tare weight reduction and planned track upgrades

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Current max axle load</td>
<td>15 ton/axle</td>
</tr>
<tr>
<td>2</td>
<td>Number of axles</td>
<td>4</td>
</tr>
<tr>
<td>3</td>
<td>Current max wagon weight (1 x 2)</td>
<td>60 ton</td>
</tr>
<tr>
<td>4</td>
<td>Current max line load</td>
<td>5 ton/m</td>
</tr>
<tr>
<td>5</td>
<td>Current minimum wagon length (3 / 4)</td>
<td>12 m</td>
</tr>
<tr>
<td>6</td>
<td>Current actual wagon length</td>
<td>13.362 m</td>
</tr>
<tr>
<td>7</td>
<td>Current tare weight</td>
<td>15.3 ton</td>
</tr>
<tr>
<td>8</td>
<td>Current load capacity (3 - 7)</td>
<td>44.7 ton</td>
</tr>
<tr>
<td>9</td>
<td>Current load capacity (8 / 6)</td>
<td>3.35 ton/m</td>
</tr>
<tr>
<td>10</td>
<td>Tare weight reduction goal (25-30 % x 7)</td>
<td>3.8 – 4.6 ton</td>
</tr>
<tr>
<td>11</td>
<td>Goal tare weight (7 - 10)</td>
<td>10.7 – 11.5 ton</td>
</tr>
<tr>
<td>12</td>
<td>New max axle load</td>
<td>18 ton/axle</td>
</tr>
<tr>
<td>13</td>
<td>New maximum line load</td>
<td>8 ton/m</td>
</tr>
<tr>
<td>14</td>
<td>New max wagon weight (2 x 12)</td>
<td>72 ton</td>
</tr>
<tr>
<td>15</td>
<td>New minimum wagon length (14 / 13)</td>
<td>9 m</td>
</tr>
<tr>
<td>16</td>
<td>New load capacity (14 - 11)</td>
<td>60.5 – 61.3 ton/wagon</td>
</tr>
<tr>
<td>17</td>
<td>New load capacity (14 / 15)</td>
<td>4.5 – 4.6 ton/m</td>
</tr>
<tr>
<td>18</td>
<td>Load capacity increase (17 – 8)</td>
<td>15.8 – 16.6 ton/wagon</td>
</tr>
<tr>
<td>19</td>
<td>Load capacity increase (16 / 8)</td>
<td>35 - 37 % / wagon</td>
</tr>
<tr>
<td>20</td>
<td>Load capacity increase (17 – 9)</td>
<td>3.4 - 3.5 ton/m</td>
</tr>
<tr>
<td>21</td>
<td>Load capacity increase (17 / 9)</td>
<td>103-106 % / m wagon</td>
</tr>
</tbody>
</table>

The table shows that the load capacity per wagon would increase with 35 – 37 % as a combined result of the reduced tare weight and the increased max axle load of the railroad. The new wagon can, however,
be made shorter than the current one. If the goals can be achieved, the resulting increase in load capacity will be over 100 % per meter wagon and of the same order of magnitude counted per train. The increased railroad track capacity in terms of max axle and line load is what makes the biggest difference in this case, the reduced tare weight only contributing to a fraction of that. The goal of reducing the tare weight with 25-30 % is also very challenging, since the bogies makes up for a large part of the total wagon weight and they will be the same on the new wagon. In addition, the wagon has to be strong enough to handle the drastically increased load capacity, which adds to the tare weight.

3 Wagon design

In order to be able to judge a possible reduction of the tare weight, a first step is to take a look at the different sub systems of the wagon. The Side Dumper consists of the following sub systems from top to bottom: lid, body, side doors, frame, draw gear, footsteps, handrail, brake system and finally, bogies, see the figure below.

Some sub systems are standard purchased parts. In those cases, the lightest solutions are chosen. The other sub systems are mainly welded steel parts designed by Kiruna Wagon. These are the ones that are in focus for weight reduction in the Hermes project. The table below shows a list of the sub systems, their current weight and a note on what category they belong to.
### Table 4: Sub systems of the current Side Dumper, their weight and category

<table>
<thead>
<tr>
<th>Sub system</th>
<th>Weight</th>
<th>Category</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lid</td>
<td>1 015 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Mechanical system for lid</td>
<td>270 kg</td>
<td>Including pipes, hydraulic cylinder and brackets, Mainly purchased</td>
</tr>
<tr>
<td>Body</td>
<td>7 580 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Side doors</td>
<td>4 x 450 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Side door locking system</td>
<td>2 x 650 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Frame</td>
<td>5 547 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Draw gear</td>
<td>1 100 kg</td>
<td>Purchased</td>
</tr>
<tr>
<td>Footsteps</td>
<td>2 x 10 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Handrail</td>
<td>2 x 8 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Brake system</td>
<td>250 + 110 kg</td>
<td>Brake + parking brake, Purchased</td>
</tr>
<tr>
<td>Bogies</td>
<td>2 x 4 635 kg</td>
<td>Including compact brake, Purchased</td>
</tr>
<tr>
<td>Weld seams</td>
<td>300 kg</td>
<td>Welded</td>
</tr>
<tr>
<td>Total</td>
<td>28 578 kg</td>
<td>N/A</td>
</tr>
</tbody>
</table>

The purchased components make up for 11 000 kg or about 1/3 of the tare weight, whereof the bogies only weighs 9 270 kg. The sub systems in question for weight reduction represent a weight of 17 578 kg or about 2/3 of the wagon weight. To reach the goal of the project, the weight of these sub systems have to be reduced with 7 125-8 550 kg or 40 – 50 %.
4 Strength requirements

For a wagon to be approved it has to be shown that it is strong enough for the expected loads during the course of its lifetime. Strength calculations have to be made according to EN 12663-1 or -2 Railway applications - Structural requirements of railway vehicle bodies. This standard describes the standard load cases that have to be taken into account. The most challenging being the fatigue load case that should be applied to all parts of the wagon and for 10 million cycles:

- Lateral acceleration ± 0.2 g
- Vertical acceleration 1 ± 0.3 g

The standard also specifies that all wagon specific load cases have to be taken into account. These are typically the loads due to loading and unloading. If the standard is followed, there is normally only one load case that has to be verified through practical tests; a static vertical loading of the frame.

In case the standard is not followed, in-service tests have to be made to validate the durability of the wagon for these cases.

5 Weight reduction

On one hand, there are several potential methods to reduce the wagon weight. On the other hand, there are several EN-norms and standards that describe how railroad wagons should be designed. In order to take full advantage of different ways for weight reduction, it may in some cases be necessary to use methods that are not described in the standards. In those cases, it has to be proven that the alternative methods provide safe solutions when using the series manufactured versions of the wagons.

The weight should first of all be reduced by smart holistic as well as detail design. Taking full advantage of AHSS is another way. The “Design Handbook. Structural design and manufacturing in high-strength steel” by SSAB is an invaluable source for ideas in this case. Taking the dampening effect of the inertia of the fully loaded wagon also seems to be a way of reducing the weight. In cases when it is known what railroad lines the wagons will be used on, and where the tracks induce lower fatigue loads than stipulated by the standards, this can be used for reducing the wagon weight. To take the time not only to mitigate the critical stresses during the wagon design, but also to minimize any over dimensioning also seems like a good idea. Another way of reducing the wagon weight may be to reduce the calculated number of fatigue vibrations and introduce routines for crack detection and mitigation, a so-called damage tolerant method.
5.1 Weight reduction through general design

When unloading, the Side Dumper body tips to the side and while the current Side Dumper has hinges between the body and frame, the modular Side Dumper that is patented by Kiruna Wagon, will have a part of the hinges on the body and the other part on the unloading station, see the figure below.

![Diagram](image)

Figure 5: On the current Side Dumper (left) the hinges are integrated in the wagon. On the modular Side Dumper (right), the body has to be supported by the unloading station on the tipping side.

The shape of the modular underframe can in this way be made much simpler and lighter, basically in the form of a beam. Other changes are to have one instead of two side doors per side and that the door locking mechanism will be moved from the body gables to the body bottom. The EU wagon for gravel will not need any lid at all, while the Iberpotash wagon lid will be like the existing Iberpotash wagons. The figure below shows an exploded view of the modular Side Dumper for Iberpotash.
The planned design changes will result in a substantial weight reduction; see the estimations in the table below.

Table 7: Weight reduction through general design

<table>
<thead>
<tr>
<th>Sub system</th>
<th>Original weight</th>
<th>Estimated new weight</th>
<th>Estimated weight reduction</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Body</td>
<td>7 580 kg</td>
<td>6 280 kg</td>
<td>1 300 kg</td>
<td>No vertical beams between doors, simpler gables</td>
</tr>
<tr>
<td>Side doors</td>
<td>4 x 450 kg</td>
<td>2 x 800 kg</td>
<td>200 kg</td>
<td>Reduced number, optimized</td>
</tr>
<tr>
<td>Side door locking system</td>
<td>2 x 650 kg</td>
<td>2 x 250 kg</td>
<td>800 kg</td>
<td>Simplified, placed under the body</td>
</tr>
<tr>
<td>Frame</td>
<td>5 547 kg</td>
<td>1 750 kg</td>
<td>3 797 kg</td>
<td>Simplified</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td>6 097 kg</td>
<td></td>
</tr>
</tbody>
</table>

The planned design changes will result in estimated weight reductions of 6 100 kg or 21% of the current wagon weight of 28 500 kg, which will lead to a resulting tare weight of 22 400 kg.
5.2 Weight reduction through the use of AHSS

Taking the definition of "vehicles" for all kinds of transport means, such as cars, trucks, trains, ships or even airplanes, it is completely true to state that the lighter the vehicle, the more cost effective will be the transport. The demands, most specifically in the transport industry have been continuously more effective and productive transport means and through weight reduction, transport performance can be automatically increased.

Advanced high strength steels gives potential for considerable weight reductions and more cost-effective ways to produce energy efficient vehicles. Structural components can be upgraded with much lighter and impact resistant components, as well as more rational and cost effective manufacturing processes. These factors have broadened the applications of high strength steels. Conventional bending, cutting and joining techniques without special equipment can be used.

Considering that the initial upgrade can start from doubling the strength of the steel (such as from S355 to S700), there is a great weight reduction potential in introducing AHSS in many kinds of transport vehicles, and several years of development in steel manufacturing, design and processing techniques have created a solid basis for this upgrading.

The potential thickness reduction can be roughly estimated by a conservative “rule of thumb” when upgrading from conventional steels to high strength steel. For steel plates and parts, the following expression can be used as a first estimation of the required thickness of AHSS plates when replacing mild steels.

\[ t_{HS} = t_{MS} \sqrt{\frac{R_{e,MS}}{R_{e,HS}}} \]

Where:

- \( t_{HS} \) is the thickness of the High Strength Steel
- \( t_{MS} \) is the thickness of the Mild Steel
- \( R_{e,MS} \) is the yield strength of the Mild Steel
- \( R_{e,HS} \) is the yield strength of the High Strength Steel

If, for example, a 6 mm mild steel plate is replaced by a 700 MPa AHSS plate, the thickness of the new plate can be reduced to 4.3 mm, see calculation below.

\[ 6.0 \times \sqrt{\frac{355}{700}} = 4.3 \text{ mm} \]

The resulting weight reduction in this case is almost 30%. With higher strength AHSS, weight reductions up to 50% is possible. The effects of stiffness, stability, joining strength and fatigue related to the new design can be addressed by different strategies as explained in the following sections.
5.2.1 Stiffness

When plate thicknesses are reduced, resulting in lighter and slender parts with lower area moment of inertia, the stiffness properties will decrease and deflection increase. This is due to the elastic modulus being the same for AHSS as for mild steels and the stiffness is only dependent of the area moment of inertia, see the equation for calculating deflection below.

Figure 8: Equation for deflection of a beam

Where:
- \( B \) = maximum deflection
- \( F \) = applied load
- \( L \) = distance between supports
- \( E \) = elastic modulus
- \( I \) = area moment of inertia

If the deflection is critical, the shape of the new part can be modified to compensate for the lost stiffness. The following picture shows the original mild steel part, the loss of stiffness due to a reduction of plate thickness and an example how this can be compensated by a change of the shape of the cross section.

Figure 9: Original mild steel part (left), AHSS part with reduced thickness and stiffness (middle) and a modified part with regained stiffness.
5.2.2 Instability

Reduced material thickness and slender structural parts may lead to a risk of buckling. Thus, buckling in different configurations may have to be analyzed (see examples in the picture below). Local and shear buckling are instability phenomena which can appear for slender plate surfaces exposed to compression or shear stresses respectively. In these cases, the buckling is local, which means that the actual resistance of the plate can be significantly higher than the critical buckling stress.

The buckling effect has been exhaustively studied and its prevention by means of design artifices is very well known and widely applied, see examples in the figure below.
5.2.3 Static strength of weld joints

Despite appearing a critical point, the static strength of welded joints has been deeply developed by steel manufacturers, researchers and welding systems developers, and the result achieved has shown optimum levels of quality and reliability.

The static strength of the complete joint depends on the properties of the individual parts of the joint, which in this case, means the characteristics of the heat affected zone (HAZ), the weld metal, the unaffected parent metal and their interactions with each other. Certainly, the part which deserves most attention is the HAZ, since each particular sub-zone of the HAZ is subjected to a unique “heat treatment” from the welding process, which causes the local mechanical properties to vary throughout the HAZ. This includes the static strength throughout the HAZ, where parts of the HAZ will typically have a lower static strength than the unaffected parent metal.

However, when the lowest strength of the joint is in the narrow HAZ, obtained by correct parameters, the tensile strength transverse to the complete joint can be higher than the part of the HAZ with the lowest strength. In simplified terms, the narrow lower strength section will interact with the surrounding areas of higher strength; strengthen the entire joint by means of local plasticizing effect and redistribution of stresses.

5.2.4 Fatigue of AHSS

Lighter and more optimized structures achieved through the use of AHSS entail better material efficiency and allows for higher stresses. This turns fatigue into a design parameter of a higher extent and it pays off to devote time and effort to perform accurate fatigue design. This is solely dependent of the increased stress levels which in turn are a pure consequence of the design which allies stronger materials and slender parts and not with the steel itself.

There is a huge difference in fatigue resistance of materials with none or mild mechanical notches (holes) versus materials with crack-like imperfections such as welded joints. Welded joints often have sharp notches at the weld toe and weld root resulting in rapid initiation of fatigue cracks and thereby dominated by crack growth. On the other hand, the fatigue mechanism of the parent material is dominated by the initiation phase, resulting in a much higher fatigue resistance, see the figure below.

Figure 12: Material with crack-like imperfections does not have a crack initiation period, resulting in much lower fatigue resistance.
For the parent material, the fatigue resistance increases with the yield strength. For welded steels, the crack propagation rate or fatigue strength is independent of the steel grade, see the figure below. The fatigue strength of welds can, however, be improved by post-weld treatments.

![Fatigue strength between parent material and welded component](image)

**Figure 13:** Fatigue strength between parent material and welded component.

### 5.2.4.1 The effect of joints on fatigue resistance

Considering that several kinds of joints are needed to transform parent material into a final product, it is important to know how to handle joints in the design to ensure a correct life length with a minimized weight. Therefore the design decisions always have to be evaluated based on the detail category applied to the structure. Each detail category has a defined fatigue strength value, the so-called FAT class. *EN 1993-1-9 Eurocode 3: Design of steel structures – Fatigue* has lists of different weld and screw joints and their detail categories.

The stress range that the joint will experience during its life time depends on its placement. The combination of the detail category and the stress range determines the life length of joints of normal steel constructions. The calculation of fatigue life is described in *EN 1993-1-9 Eurocode 3: Design of steel structures – Fatigue* and is calculated according to the equation below:

\[
N = 2 \cdot 10^6 \cdot \left( \frac{FAT}{\Delta \sigma} \right)^m
\]

where

- \( N \) – number of load cycles
- \( FAT \) – fatigue resistance value (MPa)
- \( m \) – slope of the S-N curve, \( m = 3 \) or \( 5 \)
- \( \Delta \sigma \) – stress range

![Equation for calculating the allowed fatigue life depending on load cycles and stress range](image)

**Figure 14:** Equation for calculating the allowed fatigue life depending on load cycles and stress range.
In this equation, instead of 2 million, the number of cycles should be 10 million for railroad vehicles according to EN 12663 Railway applications - Structural requirements of railway vehicle bodies. This means that the detail category has to be recalculated to lower numbers.

Examples of fatigue resistance due to welds and holes compared to the parent material for Domex 700 AHSS is shown in the figure below.

As the figure shows, a hole reduces the fatigue resistance with 90 % and welds with 96.5-98.3 % depending on the direction of the weld.

As far as known to Kiruna Wagon, there are no standards that describe detail categories for glued joints.
5.2.4.2 Mechanical joining

According to an example in EN 1993-1-9, a bolted joint has about 40% higher detail category than a similar welded joint, see the figure below.

This makes it interesting to study mechanical joining for weight reduction. The fatigue loaded chassis of modern Scania trucks are 100% bolted or hot riveted. This further reinforces the thought that mechanical joining is good for fatigue. A drawback with mechanical joining is that they in many cases require extra material which adds weight, for example for overlapping plates and bends to form the joints.

Holes with poor surface roughness or material changes for example due to thermal cutting, may introduce crack-like defects causing drastically decreased fatigue life. This means that the hole surface roughness and conditions are critical for the fatigue life. It seems like the standards assume a calculation based on hole edge stresses (bearing stresses), which makes the surface roughness important. An exception may be holes that are screwed with very high clamping forces. According to “Guide to Design Criteria for Bolted and Riveted Joints” by Geoffrey L. Kulak et al., the fatigue strength is improved when rivets are replaced by high-strength bolts. The high clamping force in the bolt results in a much better stress condition at the critical sections at the fastener holes. If sufficient slip resistance is provided, bearing stresses are eliminated and crack initiation and growth is not as critical at fastener holes. Questions for further literature studies and/or tests, that will be tackled next in Task 1.4:

- How much do different hole surface qualities affect the life length/need for additional weight? Laser cut/water jet cut/drilled/punched?
- How much does the detail category improve with high-strength bolts?
- Can the same effect be reached with huck bolts, with their high clamping forces, as with high strength bolts?
- What is the effect of possible corrosion on the clamping forces of high-strength and huck bolt joints, and what are the possible mitigation methods for that?
- Does hot riveting improve the detail category, and if so, how much?
5.2.4.3 Gluing

Welds and mechanical joints join parts along lines; weld seams or hole edges. This cause stress concentrations in the line of attachment. Glue on the other hand, is applied to surfaces. This distributes the stress in the joint which reduce the stress concentrations. In some cases this can be a big advantage. Glue is especially good for shear forces but not for peeling forces. Gluing requires clean surfaces and curing time. Questions for further literature studies and/or tests in task 1.4:

- How to calculate fatigue life of a glued joint?

5.2.4.4 Design for improved fatigue resistance

In order to pursue longer fatigue life for each detail of the structure, in a brief and simplified perspective, the task of the designer is to try to push all joints to zones with lower stress ranges while always trying to design the joints with better FAT classes. An example of this kind of smart design is shown in the picture below.

![Figure 17: The effect of moving joints to lower stress range zones can increase the life length several times.](image)

In addition to the inferior effect on fatigue life of the joint itself, many times there is also a rapid change of stiffness in or adjacent to the joints, causing extra stress concentrations. An example is when thick plates are joined with thin plates. One way of limiting this effect is to make the transition more gradual for example by adding a medium thick plate between the two plates, see the figure below.
Another kind of rapid dimensional change that cause stress concentrations is in sharp corners. A workaround in these cases can be to replace corners with rounded shapes, see example in the figure below.

More design suggestions are found in SSAB’s “Design Handbook. Structural design and manufacturing in high-strength steel”.
5.2.5 Post weld treatments

Welds introduce residual stresses and is considered to be a crack in the material. This is what reduces the fatigue life length of the welded parts. Post weld treatments aim at reducing the residual stresses, introduce pressure tensions and/or reshaping the weld toe to be more fatigue resistant. Ways of post weld treatments mentioned in EN 15085-3 Railway applications. Welding of railway vehicles and components - Design requirements, are peening, grinding and heat treatment. Another method is TIG dressing. Peening and grinding introduces pressure tensions and reshapes the weld toe. Heat treatment relieves the residual stresses. TIG dressing aims at reshaping the weld toe.

On page 26 of EN 15085-3 is says that post weld treatments in the form of peening, grinding and heat treatment makes it possible to allow either higher stresses or longer lifespan of the joints. It does, however, not tell how much. According to the international institute of welding, IIW, post weld treatments increase the detail category with 30 %, except peening that increases the detail category with 50 %.

Of the different post weld treatment methods, heat treatment is very expensive, especially for large wagon parts. Grinding is hard to do on AHSS due to their hardness.

High Frequency Mechanical Impact is a form of peening that allows for increasing the detail category of welds up to five steps for 700 MPa steel and an astonishing eight steps for 960 MPa steel, see figure below. This method is simple to learn and the size of the equipment is small enough to allow for treatment in every place that is accessible for welding.

High Frequency Mechanical Impact is a kind of peening that aims at reshaping the weld toe and introduce pressure tensions. The result is a ditch around the weld, see figure below. It is important that the ditch gets the correct width and depth, which is checked with a template.
With this method, it is important that the weld has the correct size and penetration, in order to avoid the crack starting in the weld root. A remaining question is what penetration is required to ensure that the crack starts in the weld toe?

![Image of weld](image)

Figure 21: The HFMI ditch around the weld has to have the right shape.

### 5.2.6 Example weight reduction using AHSS

To explore the weight reduction potential through use AHSS, a FEA was performed with the commercial code ABAQUS V6.13 (standard formulation) and two AHSS grades were compared DOMEX 700 and STRENX 960, see the table below.

<table>
<thead>
<tr>
<th></th>
<th>DOMEX 700 MCD</th>
<th>STRENX 960 MC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Yield Strength</strong></td>
<td>700 MPa</td>
<td>960 MPa</td>
</tr>
<tr>
<td><strong>Fatigue Limit</strong></td>
<td>420 MPa</td>
<td>630 MPa</td>
</tr>
</tbody>
</table>

Table 22: AHSS properties

The evaluation of weight reduction was focused on a side door of the current Kiruna Wagon Side Dumper, see the figure below. The reduction is expected to be similar for the other sub systems of the wagon when mild steels are replaced with AHSS.
The door was modelled by shells elements, the mesh and boundary conditions are shown in the figure below with the respective thickness of each of the steel types. The thickness reduction was initially calculated using the ratio between yield strength and also by less of the half of this ratio (37% and 15% respectively). The load on the side door was applied by hydrostatic pressure on all internal walls, up to reach the allowed mechanical limit of the material.

<table>
<thead>
<tr>
<th>PART</th>
<th>DOMEX 700 MCD</th>
<th>STRENX 960 MC</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Strength Criterion (-37%)</td>
<td>Fatigue Criterion (-15%)</td>
</tr>
<tr>
<td>Central Panel</td>
<td>4 mm</td>
<td>2,5 mm</td>
</tr>
<tr>
<td>Other parts</td>
<td>6 mm</td>
<td>3,7 mm</td>
</tr>
</tbody>
</table>

The next two figures show the stresses on the door after a plate thickness reduction of the two AHSS grades according to the thickness reduction rules “strength criterion” and “less than half ratio criterion” respectively.
If the strength criterion is used, the thickness changes from 4 to 2 mm and from 6 to 3.7 mm respectively. If less of the half ratio is used, the thickness changes from 4 to 3.4 mm and from 6 to 5.1 mm respectively. The left side of the figures show the stress using the DOMEX 700 limit value on the legend scale, and the right side shows the legend with STRENX 960 limit value. It can be seen that under the applied load, the stress distribution is similar when the reference is the stress limit of each material. With this thickness reduction, no failure will be expected.

The figure below shows the displacements of the doors with the different materials and thicknesses in loaded condition. The displacement is largest on the thinnest door, the one with the plates of the highest yield strength, due to the decreased stiffness associated with the thickness reduction of the plates.
In the same way, the natural frequencies decrease with the reduction of plate thickness, which is shown in the figure below, while the eigenmodes do not show large changes. To ensure proper sealing of the wagon body, the deflection of the doors should not increase due to the new material. This means that the lighter doors would have to be redesigned to be stiffer, what could possibly add some weight.

In summary, the weight of two side doors can be reduced with 340 kg from 880 kg to 540 kg plus possible extra weight to recreate the original stiffness of the doors. The wagon has four side doors, which means a total potential weight reduction of 680 kg of the doors by using the higher strength steel.

Assuming all of the wagon sub systems are made from 700 MPa steel and it is replaced by 960 MPa steel in the same way as for the side doors, the potential weight reduction of the whole wagon of the current design is 17 578 kg x 37 % = 6 500 kg according to the strength criterion. However, deeper analysis are
necessary to determine the design and resulting weight reduction potential for each particular sub system, considering also the joints of the parts and necessary reinforcements to retain stiffness and against buckling. An estimation would be that the theoretically possible weight reduction of 37 %, in reality would be 10 % less, i.e. 27 %. For fatigue exposed parts, this reduction will be lower.

In addition to the issues with stiffness and buckling when converting to thinner and stronger steel, the natural frequency of the different parts decrease. Care has to be taken in the design so that their natural frequencies in their assembled condition are higher than the frequencies of the railroad wagon in use.

5.3 Dampening

The standard EN 12663 Railway applications - Structural requirements of railway vehicle bodies specifies the amplitude and number of fatigue cycles that every part of railroad wagons has to withstand. In practice, the fatigue vibrations arise from the wheel – rail interaction and are translated up through the wagon. According to Kiruna Wagon’s previous experience, a fully loaded wagon has a substantial inertia that in practice will dampen the vibrations, more the farther away the part is from the bogies. This means that a wagon designed according to the standard will be over dimensioned, the farther the parts are from the bogies.

One way of minimizing the wagon weight would be to establish a model for the actual amplitude vibrations depending on where the parts are placed on the wagon.

This method does not work for standard purchased components like bogies and draw gear. It does not either give any effect on components directly attached to the bogies. If possible, these components should be moved from the bogie, in order to allow a weight reduction of them. The components in question for this investigation are hence mainly the components that today are welded, except the frame, which should be designed according to the standard. The table below shows a summary of the components in question and their weight on the current wagon.

Table 29: Potential parts for weight reduction due to dampening.

<table>
<thead>
<tr>
<th>Sub system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lid</td>
</tr>
<tr>
<td>Body</td>
</tr>
<tr>
<td>Side doors</td>
</tr>
<tr>
<td>Side door locking system</td>
</tr>
</tbody>
</table>

In order to make a correct estimation of the potential weight reduction when dampening is taken into account, a model has to be established for how to calculate the dampening effect on the fatigue vibrations for different parts of the wagon. Until this model is established, the weight reduction potential due to dampening is assumed to be 3.5 % on the parts in question. This model is under development to fulfil requirements for wagon design of Task 5.3
5.4 Optimize for real tracks

According to Kiruna Wagon’s previous experience, the vibrations described in EN 12663 Railway applications - Structural requirements of railway vehicle bodies causing the fatigue loads on wagons are significantly worse than the case with some wagons on particular tracks. The reason is mainly better bogie behaviour and higher quality tracks and track maintenance than assumed in the standard. In the cases where it is known that a particular wagon only will go on a particular railroad stretch, more realistic vibrations can be used as inputs for the strength calculations. In that way the vibration amplitudes can be reduced and as a consequence, the tare weight of the wagon. With today’s knowledge at Kiruna Wagon, this is however not applicable for wagons for general use in Europe. In order for that to be applicable, there would have to be established a realistic estimation of vibrations in some kind of “worst case” tracks of Europe, and the resulting vibrations on a specific wagon, which would require a lot more work. This task therefore is not reasonable to include in the Hermes project.

5.5 Minimize over dimensioning

In normal wagon development projects, the focus is on managing the critical spots to make them strong enough and little time is left to minimizing the weight of parts that are over dimensioned in preliminary designs. The weight reduction potential of this is estimated to 3 % of the parts that are possible for weight reduction.

5.6 Damage tolerant method

Another way of minimizing the wagon weight that Kiruna Wagon has considered is to reduce the technical lifespan of the wagon and introduce regular check-ups of cracks and methods for mitigation of crack propagation in order to extend the lifespan. This method is called Damage Tolerant Method in EN 1993-1-9. The standard suggests four ways of using this method (see page 11 in the standard):

- Selecting details, materials and stress levels so that in the event of the formation of cracks a low rate of crack propagation and long critical crack length would result
- Provision of multiple load path
- Provision of crack-arresting details
- Provision of readily inspectable details during regular inspections.

5.6.1 Fundamentals of Damage Tolerant methodology

Railway components are usually designed for infinite life based on the fatigue strength of the material first introduced by Wöhler (SN curves). This approach is in general sufficient, but a comparatively small number of failures occur in practice, a fact that, besides other reasons, is due to limitations and uncertainties of this methodology such as:

- The number of loading cycles in railway components experience over their service time, which is usually a multiple of $10^6$–$10^7$ cycles. For a duty of 400,000 km per year, which is typical of high speed railway systems, the number of load cycles of axles and wheels is about $2 \times 10^8$ [1] which refers to the range of so-called giga-cycle fatigue [2,3]
- Complex loading conditions, which are much more stochastic in nature than the hypothetical loads used in the design rules or even realised in accompanied component testing
- Deviations from the assumption of component surfaces free of flaws. Gravier et al. [4] report on ballast impacts to railway axles which may occur on high speed tracks. Such events are seldom but when they occur the consequence can be sharp angle notches which accelerate the nuclear-
tion and further propagation of fatigue cracks. In addition, protective coatings are locally destroyed promoting corrosion damage.

In order to compensate for these uncertainties usually generous safety factors are applied. These higher safety factors lead to oversizing the components, resulting in component extra weight.

For understanding the phenomena of fatigue it is important to understand, that the initiation phase of a fatigue crack is in reality a microstructural crack extension phase, the growth rate of which is decreasing with increasing crack size until the crack is arrested at a microstructural barrier such as misoriented slip systems due to grain boundaries, second phases, twin boundaries, etc.

Therefore, the fatigue limit is associated not only with a certain stress level but also with a certain crack length which corresponds to the distance the crack can growth until it is arrested [5]. For cracks larger than this distance, in the figure below, designated by P, the fracture mechanics based fatigue threshold provides a second fatigue limit. Note, that this limit, is also caused by the arrest of an originally smaller, extending crack. Both, the endurance limit and the fatigue threshold can, therefore, be regarded as different states of the fatigue limit controlled by crack arrest at structural barriers. Note, that overload events may cause the crack to overcome such a barrier. As the consequence there is a transition from one fatigue limit state to the other and finally to extended crack growth up to the failure of the component.

![Fatigue limit as a function of crack size (Kitagawa diagram).](image)

Thus, short crack fracture mechanics analyses can be applied to the determination of crack initiation SN curves. In principle the initial crack size from the manufacturing process can be adjusted such that the simulated number of load cycles up to a crack extension of 0.5 or 1 mm is identical to that obtained in the SN curve.

### 5.6.2 Fail-safe concepts

While safe-life concepts do not allow for any fracture up to the end of service life, the fail-safe concepts allow for local or limited fracture as long as global or catastrophic failure of the overall structure is excluded. Fail-safe concepts are redundant design (multiple load path) or design for crack arrest.

#### 5.6.2.1 Redundant design

The key task of a redundant design philosophy is the detection of local damage or partial fracture of one or more sub-structures. This can be realised by in situ monitoring of the service operation or by regular inspections. The essential requirement is that the partial failure will be detected before it can cause significant damage to the whole structure. Frequently, the residual lifetime of the still intact member will be very limited. In such a case the component must be repaired or retired as soon as possible. If a limited
continued service time is required from a service point of view it has to be guaranteed that it will not exceed the residual lifetime of the predamaged component, which has to be established by fracture mechanics analysis.

5.6.2.2 Design for crack arrest
Another fail-safe concept is design for crack arrest. An extending crack may arrest if the actual crack driving force decreases with the extending crack, or if the crack advances into a region of increased crack resistance. As a rule, the decrease in crack driving force is due to certain design features such as crack stoppers or stress shielding. Thermal or residual stress fields may have a similar effect. If a structure is cyclically loaded a crack may continue to extend even at load levels below crack arrest, which means that the structure is safe only for a limited time. During that time span intervention from outside is necessary to stop the crack before it reaches a new critical size. A fracture mechanics based prediction of crack extension after the arrest incident is necessary for establishing the residual lifetime.

5.6.3 Damage tolerance concepts
Within the frame of the damage tolerance concept the possibility of fatigue crack extension is basically accepted. The aim is to prevent the crack to grow to its critical size during the lifetime of the component. Different philosophies such as periodic inspections, overload tests or periodic removal of cracks can be realised within a damage tolerance concept.

5.6.3.1 Classical damage tolerance concept (periodic inspections)
Periodic non-destructive inspection (NDI) in combination with fracture mechanics analyses forms the classical damage tolerance concept. Classical damage tolerance covers five steps:

Step (1) Establishment of an initial crack shape and size for the further analysis. Within a damage tolerance concept the initial crack size, \( a_0 \), is not identical to the size of a real flaw from the manufacturing process but is a fictitious size, which usually refers to the detection limit of the NDI technique.

The basic idea is that the largest crack that could escape detection is presupposed as existent. Note, that under industrial conditions the NDI detection limit is a statistical quantity rather than a fixed value, since it refers to that crack size which will be detected with a sufficiently high probability. This is dependent on a number of factors such as the applied NDI methodology (visual inspection, dye penetrant inspection, ultrasonic testing, magnetic particle inspection, etc.), the degree of automation, the skill of the operator with respect to the test method and with respect to the component, redundancy, the accessibility of the potential defect site, and the component material. The initial crack shape (semi-elliptical or semi-circular surface crack, corner crack, through crack, etc.) is based on experience from experiments or failure analyses. Note, that the initial crack size as well as the initial crack shape will be of significant influence on the residual lifetime determined at the subsequent analysis step.

Step (2) Simulation of sub-critical crack extension. The initial crack can extend due to various mechanisms such as fatigue, stress corrosion cracking, high temperature creep, or combinations of these mechanisms. This kind of crack growth is designated as sub-critical since it will not lead to immediate failure until a critical length of the crack is reached. For railway applications the common mechanism is fatigue but a possible contribution of corrosion to damage should always be kept in mind. For semi-circular or semi-elliptical surface cracks, crack extension has to be calculated simultaneously in the thickness direction and along the surface.

In fatigue, crack extension is expressed as a function of a stress intensity factor range, \( \Delta K \), and the crack extension rate, \( da/dN \), whereby \( da \) denotes an infinitesimal crack extension due to an infinitesimal number of loading cycles, \( dN \). There exist numerous equations for describing the \( da/dN - \Delta K \) relationship, the best known of which is the Paris Law, equation 1.
\[
\frac{da}{dN} = C \Delta K^m
\]

This equation describes the data in the so-called Paris range (range 2) where the curve in the double logarithmic plot constitutes a straight line. An upper bound to many empirical curves for steel is provided by reference values to the fit parameters \( C = 1,6475 \times 10^{-11} \) for \( da/dN \) in m/cycle and \( K \) in MPa √m, and \( m = 3 \) [6].

The possibility to use reference values for certain material classes such as steels is due to the fact, that factors such as the mean stress \( \sigma_m \) in a load cycle, the stress ratio \( R = \sigma_{\min}/\sigma_{\max} = K_{\min}/K_{\max} \) and the environment are of minor influence on the curve in range (1). The same is true with respect to the microstructure of the material. Note, however, that the threshold value \( \Delta K_0 \) shows a strong \( R \)-dependence. A conservative fatigue threshold value is given in [6] as \( \Delta K_0 = 2 \) MPa √m for steels.

**Step (3)** Determination of the critical crack size for component failure. The sub-critical crack extension is determined by the failure of the component. This may occur as brittle fracture or as unstable ductile fracture. Critical states may, however, also be defined by other events such as stable ductile crack initiation or the break-through of a surface crack through the wall.

**Step (4)** Determination of the residual lifetime of the component. The residual lifetime is that time or number of loading cycles which a crack needs for extending from the initial crack size, \( a_0 \), (step 1) up to the allowable crack size, \( a_c \), established in step (3). As already mentioned, besides all the factors, which affect the crack extension rate (step 2) the residual lifetime strongly depends on the choice of the initial crack size and shape (step 1). This is also a crucial point for the final step, the establishment of an inspection plan.

**Step (5)** Establishment of inspection intervals or formulation of demands for non-destructive testing. The constitution of an inspection plan is the aim of a damage tolerance analysis. From the requirement that a potential defect must be detected before it reaches its critical size it follows immediately that the time interval between two inspections has to be smaller than the residual lifetime. Sometimes inspection intervals are chosen to be smaller than half this time span. The idea is to have a second chance for detecting the crack prior to failure if it is missed in the first inspection. It is, however, also obvious that frequent-
ly even two or more inspections cannot guarantee the crack being detected since this would require a 100% probability of detection. In reality the probability of detection (POD) depends not only on the NDI technique but also on the actual crack size, as illustrated by the example in the figure below.

![Figure 32. Probability of crack detection as a function of the crack size and various methods of NDI.](image)

POD-crack size curves can be used to establish inspection intervals on a statistical basis. The probability of detection is increasing from inspection to inspection since the crack becomes larger during the time in-between. In addition, each new inspection increases the cumulative probability of all inspections that the crack will be found in time. The cumulative probability can easily be determined when the crack size versus time or cycle dependency (step 2) and the POD-crack size curve for the NDI method are known.

Establishing inspection intervals is essentially a statistical task acting as a link between fracture mechanics analysis, non-destructive testing and the constraints of industrial practice. It is important to realise that the cumulative probability of detection depends on both:

- Initial crack size which refers to a statistical NDI detection limit under industrial conditions
- Inspection interval, the time span between two inspections

Therefore, if an operator wants to save expenses by extending inspection intervals he has to improve his NDI technique having in mind that this requirement refers to the largest crack that could escape detection and not to the smallest crack which can be found. Conversely, if the inspection interval is fixed due to constraints from operation, the relation between crack extension, critical crack size and the given inspection interval can be used to formulate commitments on minimum requirements for NDI.

Finally it has to be noticed that classical damage tolerance can only be applied if the critical crack size is large enough to be reliably detected by NDI and if the time span between its potential detection and failure is large enough to establish a maintainable inspection plan under the constraints of industrial application. If these conditions are not given, alternative versions of damage tolerance such as overload tests or periodic removal of cracks can be carried out.

### 5.6.3.2 Overload testing

Periodic overload or proof testing is an alternative damage tolerance concept. A structure is subjected to overloading as compared to the service conditions. Since this is done in a special proof test facility potential failure happens in a controlled manner. If the component survives the test, then it is assumed to be safe under the more moderate design load. The problem of this concept is that the proof test by itself may create or enforce some damage. To prevent this is the aim of the accompanied fracture mechanics
analysis. A critical crack size is determined according to the overload, which refers to the maximum crack size that just does not cause fracture during the proof test. With that value as a potentially existing or initial crack size, $a_0$, after the overload test the further analysis is performed as in the case of classical damage tolerance. Fatigue crack extension and failure are determined as described above for the applied service load. The residual lifetime is, however, not used for establishing an inspection plan but for determining the time span after which the next overload test has to be carried out. If the component passes this new test it can be operated for an identical time interval until the next test, etc. The concept of overload testing is in particular useful in cases where non-destructive testing is not reliable because of small critical cracks or other factors.

5.6.3.3 Periodic removal of the crack

Periodic removal of the crack is also an alternative assessment philosophy in cases where the critical crack size is potentially smaller than realistic detection limits of common NDI techniques. A special problem is the determination of a representative initial crack size and shape from the manufacturing process. This information has to be provided experimentally. Since the initial crack size will usually show a certain scatter band, statistical processing will be necessary in many cases. As in damage tolerance, the extension of the initial crack will be simulated until the crack reaches its critical size. The residual lifetime then defines the time span between the periodic removal of the crack, e.g., by grinding. The fracture mechanics analysis has to be performed on the basis of elastic–plastic crack tip parameters such as the J-integral or the CTOD since small cracks are outside the range of applicability of linear elastic fracture mechanics. An important engineering application could be removal of surface cracks of railway rails.

5.6.4 Durability design versus damage tolerance

As a rule, durability design based on the SN concept and damage tolerance are not in a relation of competition. In contrast, they form two complementary safety levels with a common aim. Durability design can be regarded as a basic safety level. There are, however, a few cases where the concept is not sufficient to prevent the development and extension of fatigue cracks. The aim of damage tolerance as a second safety level is to hinder those cracks to reach a critical size, which would cause fracture. In common damage tolerance concepts fracture mechanics is used to establish inspection intervals or to provide non-destructive testing with information on the crack that has to be detected with high probability.

The damage tolerance approach however, makes it possible using less safety factors than durability design, which will result also in a lighter design.

The aim of this section is to demonstrate that fatigue design following classical damage tolerant approach allows weight reduction of components. This demonstration will take place from a couple of fatigue design examples of equivalent components.

5.6.5 References in section 5.6


5.7 Stress-Life approach versus Damage Tolerant methodology

5.7.1 Life estimation using Stress-Life (SN) approach

A notched part made from DOMEX700MC sheet is shown in the figure below. The part should withstand a maximum fatigue load of 100kN with load ratio of 0.1. The Stress-Life methodology will be used to establish the minimum thickness that must have the DOMEX700MC steel made specimen to support this cyclic effort.

![Figure 33. Sheet specimen with a central circular hole.](image)

First of all it is necessary to know that notches concentrate stresses and strains. The degree of concentration is measured by the elastic stress concentration factor, $K_t$, defined as the ratio of the maximum stress, $\sigma$, or strain, $\varepsilon$, at the notch (local) to the nominal stress, $S$, or strain, $e$.

\[ K_t = \frac{\sigma}{S} = \frac{\varepsilon}{e} \quad (1) \]

$K_t$ depends on the ratio of hole diameter to sheet width. The figure below shows $K_t$ plotted versus the ratio of hole diameter to sheet width [1]. Two curves are shown. In the upper curve the nominal stress is defined as load divided by total or gross area ($w \times t$), whilst in the lower curve by net area, the area remaining after the hole has been cut out. In this example the net area will be used.
According to figure 35 and specimen dimensions (figure 34), the stress concentration factor following the relationship based on net section is 2.3.

In fact, the stress concentration factor is needed to calculate the fatigue notch factor, $K_f$. Thus, the effect of the notch in stress-life approach is taken into account by modifying the unnotched SN curve through the use of fatigue notch factor, $K_f$.

Values of $K_f$ for $R = -1$ generally range between 1 and $K_t$ (where $K_t$ is defined based on the net area), depending on the notch sensitivity of the material, $q$, which is defined by:

$$q = \frac{K_f - 1}{K_t - 1}$$  \hspace{1cm} (2)

A value of $q = 0$ (or $K_f = 1$) indicates no notch sensitivity, whereas a value of $q = 1$ (or $K_f = K_t$) indicates full notch sensitivity. Neuber [2] developed the following approximate formula for each notch factor and $R = -1$ loading:

$$q = \frac{1}{1 + \sqrt{\rho/r}} \text{ or } K_f = 1 + \frac{K_t - 1}{1 + \sqrt{\rho/r}}$$  \hspace{1cm} (3)

Where $r$ is the radius at the notch root. The characteristic length, $\rho$, depends on the material. Values of $\rho$ for steel alloys are shown in the figure below [3].
According to the figure above and equation 3, the geometry shown in the first figure in this section made of DOMEX700MC steel, with an ultimate strength of 865MPa, the fatigue notch factor is 2.2.

This fatigue notch factor is applicable only if the load ratio is -1 (fully reversed fatigue). Generally, the real parts are not subjected to this load ratio, being due to use Modified Goodman Diagram for notched parts.

To plot Modified Goodman Diagram is necessary to know the Ultimate Tension Strength (865MPa), Yield Stress (820MPa), Fatigue Stress at R = -1 or other load ratio (203MPa at R = 0 and 2000000 cycles) and fatigue notch factor (2.2). With the example data it is possible to draw the Modified Goodman Diagram for smooth and notched parts shown in the figure below.

Using the Modified Goodman Diagram shown in the figure above is possible to determine than the maximal admissible stress for a specimen made of DOMEX700MC with a central hole (see the first figure in
this section) loaded with a load ratio of 0.1 is 231 MPa (127 MPa of mean stress + 104 MPa of stress amplitude). If the maximum fatigue load that the specimen has to withstand is 100 kN, the minimal thickness of the specimen is 14 mm.

5.7.2 Re-design with classical damage tolerance approach

An equivalent geometry of previous example is shown in the figure below. In this case, a pair of cracks growing from a circular hole are drawn, given that the application of damage tolerance approach requires the existence of previous cracks.

![Figure 37. Sheet specimen with a pair of cracks growing from circular hole.](image)

The first stage to design the part following the damage tolerance concept is to determine the stress intensity factor. The stress intensity factor for a pair of cracks growing from a circular hole in a wide plate can be approximated by [1, 4]:

\[ K \approx 1.12 (K_t S_{net}) \sigma \sqrt{\pi l} \]  \hspace{1cm} (4)

Where \( K_t \) is the stress concentration factor based on the net section stress, \( S_{net} \). The value 1.12 is the free edge correction factor [1].

Once stress intensity factor \( K \) is expressed in function of applied load or stress and specimen geometry, the fatigue crack growth life, \( N_f \), can be found by integrating the Paris Law equation:

\[ \frac{da}{dN} = C \Delta K^m \]  \hspace{1cm} (5)

Where \( m \) is the slope of the line and \( C \) the coefficient found by extending the straight line to \( \Delta K = 1 \text{MPa} \sqrt{\text{m}} \), known as Paris Law parameters.

Under cyclic stresses, the stress intensity factor, \( K \), becomes stress intensity factor amplitude, \( \Delta K \):
\[ \Delta K \sim Y \Delta \sigma \sqrt{\pi l} \quad \text{where } Y = 1.12(K_t S_{net}) \quad (6) \]

Integrating equation 5 with equation 6 data, fatigue crack growth life, \( N_f \), is obtained:

\[ N_f = \frac{t_f \left( -\frac{m}{2^+1} \right) - t_i \left( -\frac{m}{2^+1} \right)}{\left( -\frac{m}{2^+1} \right) C (\Delta \sigma)^m \pi^{m/2} \gamma^m} \quad (7) \]

To use the equation 7 to determine the fatigue growth life is necessary to have the fatigue crack kinetics of steel. In this example data from literature are used. Thus, for bainitic/martensitic steels \( C \) values are around \( 2 \times 10^{-12} \) and \( m \) values around 2.25. To determine the crack length from which it starts to propagate, the threshold \( \Delta K_{th} \) is needed. To know the critical crack length, the material toughness, \( K_{IC} \), is needed. In steels with mechanical characteristics similar to DOMEX700MC \( \Delta K_{th} \) values are around 6 MPa√m and \( K_{IC} \) values are around 45 MPa√m.

All this data feeds equation 7, the results of which are presented in the table below, which relates the thickness of the specimen subjected to a maximum fatigue load of 100kN with load ratio of 0.1, with the crack length at the propagation start and the critical length. The figure below represents these results. The starting thickness was calculated with Stress-Life methodology.

<table>
<thead>
<tr>
<th>Thickness, mm</th>
<th>Start propagation length, mm</th>
<th>Critical length, mm</th>
<th>Cycles to critical length</th>
</tr>
</thead>
<tbody>
<tr>
<td>14</td>
<td>0.163</td>
<td>9.2</td>
<td>4298139</td>
</tr>
<tr>
<td>10</td>
<td>0.083</td>
<td>4.7</td>
<td>1728346</td>
</tr>
<tr>
<td>8</td>
<td>0.053</td>
<td>3.0</td>
<td>921315</td>
</tr>
<tr>
<td>6</td>
<td>0.030</td>
<td>1.7</td>
<td>394191</td>
</tr>
</tbody>
</table>
5.7.3 Results in function of applied methodology

According to the Stress-Life approach example, to obtain infinite life with the specimen shown in the first figure 34, the thickness should be a minimum of 14 mm. This methodology would ensure that with this specimen geometry no cracks will appear in 2 000 000 cycles.

To apply damage tolerance design cracks must already exist, given that this approach can only be used if cracks appear and these are long enough.

In this sense, the crack length at which it starts to propagate (the critical length), for a given material (given \( \Delta K_{th} \) and \( K_{IC} \)) and an applied load, will be a function of the material thickness. For a lower specimen thickness, the length at which a crack starts to propagate before critical size is smaller, which results in less time between inspections, as shown in the figure below.
Thus, a specimen thickness of 14 mm will probably reach infinite life. Thickness reduction enables smaller cracks or defects to propagate by fatigue, being necessary to establish regular inspections, which should be planned following the damage tolerance methodology.

5.7.4 References in section 5.7


5.8 Damage tolerant method - summary

In an attempt to estimate the weight reduction potential by a damage tolerant method, Kiruna Wagon has performed an internal study. In the study, the required thickness (and weight) of plates that would withstand reduced number of fatigue cycles was compared with one that would withstand the full number of cycles according to EN 12663. Calculations were made for full lifespan, 50 % lifespan and 20 % lifespan. It turned out that the weight reduction would be greater for parts exposed to tension and compression stresses compared to parts exposed to bending, see the table below.

Table 41: Potential weight reduction due to reduced life span.

<table>
<thead>
<tr>
<th></th>
<th>Tension/compression</th>
<th>Bending</th>
</tr>
</thead>
<tbody>
<tr>
<td>50 % lifespan</td>
<td>15 %</td>
<td>10 %</td>
</tr>
<tr>
<td>20 % lifespan</td>
<td>50 %</td>
<td>25 %</td>
</tr>
</tbody>
</table>

In a wagon, most of the parts are exposed to bending. For an estimation of weight saving, the lower number should be counted on for an estimation of the total weight reduction. The fewer the calculated number of vibrations, the more frequent the crack inspections would have to be, so it would have to be judged what would be most economical.

Due to the idea of the modular wagon with chassis with long life span and exchangeable bodies, the chassis should not be designed with a damage-tolerant method. The parts to focus on for these methods are shown in the table below.
Table 42: Wagon parts suitable for a damage tolerant method.

<table>
<thead>
<tr>
<th>Sub system</th>
<th>Comment</th>
<th>To be counted</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lid</td>
<td>Hard to inspect so high up</td>
<td>No</td>
</tr>
<tr>
<td>Body</td>
<td></td>
<td>Yes</td>
</tr>
<tr>
<td>Side doors</td>
<td></td>
<td>Yes</td>
</tr>
<tr>
<td>Side door locking system</td>
<td>Expensive parts</td>
<td>No</td>
</tr>
</tbody>
</table>

The parts in question make up for a large part of the wagon weight, and even with only 10% weight reduction, the potential of using this method is significant. The implementation requires knowledge of where the cracks eventually will form on all of the hundreds of joints in the parts of the wagon. These knowledge requires extensive work, initially not foreseen in the project.

5.9 Potential tare weight reduction

In the table below, the estimated potential weight reductions from the different methods are summarized for the Side Dumper.

Table 43: Total estimated weight saving potential for the Hermes EU wagon compared with the current Side Dumper

<table>
<thead>
<tr>
<th></th>
<th>Potential weight reduction</th>
<th>Resulting tare weight</th>
<th>Comment</th>
<th>Include in the Hermes project</th>
</tr>
</thead>
<tbody>
<tr>
<td>Current SD wagon</td>
<td>N/A</td>
<td>28 600 kg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>General design</td>
<td>6 100 kg / 21%</td>
<td>22 400 kg</td>
<td></td>
<td>Yes</td>
</tr>
<tr>
<td>Dampening</td>
<td>&gt; 400 kg*</td>
<td>22 000 kg</td>
<td>3.5% of (22 400 – 11 000) kg</td>
<td>Yes?</td>
</tr>
<tr>
<td>Optimize for real tracks</td>
<td>Unclear</td>
<td>22 000 kg</td>
<td>Unclear</td>
<td>No</td>
</tr>
<tr>
<td>Minimize over dimensioning</td>
<td>&gt; 300 kg*</td>
<td>21 700 kg</td>
<td>3% x (22 000 – 11 000) kg</td>
<td>Yes</td>
</tr>
<tr>
<td>AHSS</td>
<td>2 900 kg**</td>
<td>18 800 kg</td>
<td>(37 – 10) % x (21 700 – 11 000) kg</td>
<td>Yes</td>
</tr>
<tr>
<td>Reduced lifespan</td>
<td>800 kg – N/A</td>
<td>18 800 kg</td>
<td>10 % x (18 800 – 11 000) kg</td>
<td>No, too advanced</td>
</tr>
<tr>
<td>Total reduction in plan</td>
<td>9 800 kg / 34%</td>
<td>18 800 kg</td>
<td>-</td>
<td>N/A</td>
</tr>
</tbody>
</table>

*) Estimated weight reductions that has to be evaluated.
The calculation of the potential weight reduction by the use of AHSS is based on the strength criterion minus an estimated 10% for stiffeners; see section 5.2.6. It is also based on that most of the other methods are implemented before this change and that the welded sub systems of the wagon are converted from 700 to 960 MPa steel.

As the table above shows, the tare weight of the current Side Dumper can be reduced by an estimated total of 34%, from 28.6 ton to about 19 ton by applying all of the selected methods described in this report. This means that the project goal of reducing the tare weight by 25 – 30% should be within reach. Since the total weight of the EU wagon is set, the load capacity can only be increased with the amount the tare weight is reduced, i.e. 9.8 ton, resulting in a load capacity increase of about 9.8 / 71.5 = 14%.

In the Iberpotash case, the plan is to increase the axle and line load, which gives the possibility to increase the load capacity well beyond the goal. The tare weight reduction goal of 25–30% for the new wagons will however probably be a challenge since the new wagons have to be made stronger than the current wagons due to the dramatically increased weight of the load.

6 Fatigue properties of AHSS

Part of the strategy to reduce weight in the new wagon design involves the use of Advanced High Strength Steel (AHSS), which can have a thinner gauge, and theoretically reduce weight. However, the standards already specify the fatigue limit, and if a higher fatigue limit is used during design, then a proof of the new fatigue limit must be demonstrated. Standards do not consider the higher strength or fatigue of AHSS’s, therefore fatigue tests must be performed on the proposed Strenx 960 MC steel, and the results compared to steels, like the ones used in the current Side Dumper G1, which have maximum resistance of 355 MPa, and 650 MPa. Stress versus the number of cycles (or S-N) plots should demonstrate how many cycles until failure the proposed Strenx 960 MC withstand. The type of joining method may also add weight, and have a distinct S-N behavior. If further weight savings are desired, then the life of a cracked wagon part needs to be estimated, in order to create a lighter design of finite life with programmed repairs. For the tests the reference steel will be a Domex 700 MCD steel, which is similar to the one that is used on the current wagon. This section starts with a description of fatigue in metals, and the influence fatigue has on design. Then preliminary results are shown of the finite-life fatigue behavior of the proposed and reference steels, both from the SSAB EMEA portfolio. These fatigue tests form part of the procedure to characterize the new steel in order to prove the new design life.

6.1 Stress classified fatigue behaviors

The fatigue behavior of a component, or a material, is its ability to resist degradation during a series of repetitive loads. As the number of cycles increases, the fatigue response evolves, thus:

- At high stresses, the material supports a limited number of cycles before fracture (oligocyclic Fatigue).
- At lower stresses, the fatigue life increases and the fatigue failure occurs after a limited number of cycles (faster at higher loads). If the stress is low enough the fracture will not occur, and can be considered to have an infinite life, if the value known as the fatigue limit is not exceeded.

6.2 Classification of fatigue tests

Component design requires the estimation of the fatigue limit when loads are cyclical. There are 3 fatigue testing approaches currently used:
1. Based on stresses, where the fatigue limit of the material is determined from tests in which the stress amplitude is fixed, obtaining Wöhler or S-N curves. The nominal stress-life (S-N) model was first formulated between 1850s and 1870s.

2. Based on strains, the strain amplitude is fixed to construct ε-N curves. The local strain-life (ε-N) model was first formulated in the 1960s.

3. The study based on the Linear Elastic Fracture Mechanics (LEFM) using the crack propagation kinetics. The fatigue crack growth model, first formulated in the 1960s.

These 3 approaches generate different strategies to design components subjected to fatigue. In general, the approaches based on stresses and strains are those with a wider application in engineering. The strategy based on fracture mechanics, the approach in which the damage in the component is tolerated, is based on the study of pre-existing crack growth and is widely used in aerospace industry to fully optimize the component design and to reduce its weight.

6.3 Fatigue during design

There are several strategies (or criteria) for the design of components subjected to fluctuating loads (fatigue):

1. Infinite-life design is the oldest criterion, and requires local stresses, or strains to be essentially elastic, and safely below the fatigue limit. For parts subjected to millions of cycles. Ignores the effect of variable amplitude loading and requires oversized parts or components, which makes them not competitive. The design data comes mainly from stress-cycles to failure, S-N, tests.

2. Finite-life design is based on the assumption that the part is initially flaw-free and has a finite life in which to develop a critical crack. This approach is the most commonly used in the study of components with stress concentrators or subjected to loads near the yield strength of the material. The data required for finite-life design involve mainly strain-cycles to failure, ε-N tests.

3. Damage-tolerant design requires a fracture mechanics study to assure fatigue cracks will not lead to fracture before they can be detected and repaired. The assumption is that fatigue cracks will exist in an engineering structure. The damage-tolerant design philosophy was developed in the aircraft industry, where weight penalizes the use of safety factors. This design employs multiple-load paths and crack stoppers with rigid regulations and criteria for inspection and detection of cracks. The fracture mechanics techniques require \( \frac{da}{dN} - \Delta K \) tests, or crack propagation, \( da \), per cycle, \( dN \), versus the change in stress intensity factor \( K \).

6.4 Infinite-life design

Infinite-life design is based S-N or Wöhler curves. The component is designed to withstand an infinite number of cycles, if the maximum stress is below the fatigue limit. In the materials that do not have a well-defined fatigue limit (such as aluminum alloys) the fatigue limit is defined as the stress amplitude that the material can withstand for 100 million cycles (see the figure below).
The S-N curves are obtained from cyclic tests with relatively large constant stress amplitude (2/3 of UTS). Cycles to failure are counted, and the test is repeated, but with decreasing amplitudes. The results are represented in a diagram of stress amplitude versus cycles to failure, with a logarithmic or semi-logarithmic scale (see the figure above).

The infinite-life design approach is based on the mean stress applied during the fatigue tests of a component, as the following figure depicts. The main fatigue parameters are defined as follows.

\[ S_m = \frac{(S_{\text{max}} + S_{\text{min}})}{2} \]  
\[ R = \frac{S_{\text{min}}}{S_{\text{max}}} \]  
\[ S_a = \frac{\Delta S}{2} = \frac{(S_{\text{max}} - S_{\text{min}})}{2} \]  
\[ S_a = \left(S_f\right)^{2N_f} \]  

Where \( S_m \) is the mean stress, \( R \) is the stress ratio, \( S_a \) is the alternating stress, \( S'_f \) is the fatigue strength coefficient, \( b \) the fatigue strength exponent, and \( 2N_f \) is the number of reversals to failure (2 reversals = 1 cycle). Eq.4 is also known as the Basquin equation [1].

![Figure 45. Depiction of mean stress, stress ratio, and alternating stress in fatigue loading.](image)
The combinations of alternating and mean stresses that result in the same finite-life to failure can be plotted with one of several models, whose accuracy depends on the toughness of the metal. Goodman proposed a model that takes into account the ultimate tensile strength of the material $S_u$ (see the figure below). In the figure below, where the ordinates are $S_m$ values and the abscissas are the $S_a$ values, $S_e$ is the completely reversed stress amplitude for a given life (i.e., $10^6$, $10^5$, etc.).

![Goodman Model Diagram](https://example.com/goodman_diagram.png)

Figure 46. Goodman model that takes into account the ultimate tensile strength of the material $S_u$.

The $S_m$ and $S_a$ combinations that lie below the Goodman line are considered to have an infinite-life, a criterion which is conservative for ductile metals, but good for brittle metals. In the figure below the yield line ($S_a = S_{ys}$, $S_m = S_{ys}$) is compared to the Goodman model.

![Yield Line Diagram](https://example.com/yield_diagram.png)

Figure 47. The yield line compared to Goodman model.

The Gerber, Soderberg and Morrow lines are other criteria used that consider the mean stress effect on fatigue (see the following two figures). The Morrow line considers the effect that tensile or compressive mean stresses would have on the value of the fatigue strength coefficient, $S'_f$, in the stress-life relationship (Eq. 4), where the new fatigue strength coefficient is $S' = S'_f - S_m$. A tensile mean stress reduces the fatigue life or decreases the stress amplitude that the component is able to withstand. If the mean stress is compressive, then both the fatigue life and the permissible stress amplitude increase. The quantity $S'_f$ is approximately equal to the true fracture strength from a tension test, so that it is larger than $S_u$, except for low-ductility metals, where it has a value close to $S_u$. The value of $S'$ has a larger value than $S'_f$ for compressive mean stresses, because $S_m$ is negative, whereas $S'$ has a lower value for tensile mean stresses, because $S_m$ is positive.
Deliverable D1.1

6.5 Finite-life design

Safe-life design is also based on failure at a finite number of cycles using the strain-life curves (ε-N). These are often called low-cycle fatigue (LCF) curves, because much of the data is obtained in less than 10 thousand cycles. When fatigue occurs at a relatively low number of cycles, the stresses that produce failure often exceed the yield strength. Even when the gross stress remains elastic, the localized stress at a notch is inelastic. Under these conditions it is better to carry out fatigue tests under fixed amplitude of strain (strain control) rather than fixed amplitude of stress. The total strain amplitude \( \Delta \varepsilon/2 \) is defined as follows [2, 3, 4] (see also the two following figures).

\[
\frac{\Delta \varepsilon}{2} = \frac{\Delta \varepsilon_e}{2} + \frac{\Delta \varepsilon_p}{2}
\]

\[
\frac{\Delta \varepsilon}{2} = \left(\frac{S'_f}{E}\right)^b + \varepsilon'_f\left(2N_f\right)^c
\]

Eq. 5

Where \( \Delta \varepsilon_e/2 \) is the elastic strain amplitude, \( \Delta \varepsilon_p/2 \) is the plastic strain amplitude, \( S'_f \) is the fatigue strength coefficient, which is divided by \( E \), the elastic modulus, \( b \) is the fatigue strength exponent, \( \varepsilon'_f \) is the fatigue ductility coefficient, and \( c \) is the fatigue ductility exponent. Note the use of \( \Delta \varepsilon/2 \) for total strain amplitude instead of \( \Delta \varepsilon \), the strain range. The term that describes the plastic strain amplitude is known also as the Coffin-Manson equation.
A plot of the stress at the tip of the stress-strain loop (point B) for various strain amplitudes yields a cyclic true stress-strain curve, which can be fitted with the following constitutive equation. Where \( k' \) and \( n' \) are the values to be iterated.

\[
\frac{\Delta \varepsilon}{2} = \frac{S_a}{E} + \left(\frac{S_a}{k'}\right)^{1/n'}
\]

Eq. 6

When \( \Delta \varepsilon / 2 \) and \( 2N_f \) are plotted on log-log coordinates, this yields a straight line where \( b \) is the slope. Likewise the log-log plot of \( \Delta \varepsilon / 2 \) and \( 2N_f \) provide data to determine the slope \( c \) and the intercept \( \log \varepsilon' \).

The strain-life curves plot the total strain amplitude versus the number of reversals (1/2 cycle) to failure, see the figure below. Both the elastic and plastic curves are straight lines. At small strains or long lives, the elastic strain predominates, and at large strains or short lives the plastic strain is predominant.

The low-cycle fatigue approach is used to predict the life until crack initiation at notches in machine parts where the nominal stresses are elastic, but the local stresses, and strain, at the notch root are plastic.
As in the case of infinite-life design approach, in order to use the finite-life design method in more realistic design situations, it is necessary to correct the results with respect to the presence of compressive or tensile mean stresses, and to treat less regular cycles of fatigue stress, as the figure below shows.

Figure 52. Mean-stress modification to the strain-life curve, $\sigma_0$ represents the mean stress $S_m$ [5].

The general strain amplitude versus reversal to failure behavior of metallic materials can be described in terms of how strong, tough or ductile the metal is. In the figure below, the curves show that for higher strain amplitude values, the most ductile metal withstands more reversals to failure, but the behavior is inversed for lower strain amplitude values. However, the test results of the tougher and stronger Strenx 960 MC do not show an inversion of reversals to failure when compared to the Domex 700 MCD steel. The fatigue properties of these two steels are compared in the next section.

Figure 53. Generalized strain amplitude versus reversals to failure behavior of metals in accordance to its toughness [5].
6.6 Strain-controlled fatigue tests of Strenx 960 MC and Domex 700 MCD

The low cycle fatigue tests show that for the same strain amplitude, the number of reversals to failure is larger on the Strenx 960 MC steel than on the Domex 700 MCD steel, as can be observed in the figure below. The effect that sharp edges or punching burrs may have on the results was minimized by polishing the edges. Axial-strain-controlled fatigue tests were performed in accordance with ISO 12106:2003. The reliability R and confidence C were both 50%. As an example, at 0.004 of $\Delta \varepsilon /2$, the Domex 700 MCD steel fails at 8,000 reversals approximately, whereas the Strenx 960 MC steel fails at around 20,000 reversals. At that same strain amplitude both the Domex 700 MCD, and Strenx 960 MC, were holding near 800 MPa.

![Figure 54](image.png)

Figure 54. The $\Delta \varepsilon /2$-$2N_f$ tests of Strenx 900 MC and reference Domex 700 MCD steels with polished edges, and according to ISO 12106:2003.

The difference in monotonic and cyclic-constructed true stress-true strain curves of both steels can be observed in the figure below. The Hollomon equation, $\sigma = k\varepsilon^n$, was used to fit the stress-strain data, and eq. 8 was used to create the cyclic stress-strain curve. The chemical composition and mechanical properties according to the data sheets of SSAB are shown in the two tables below, respectively. The fit of eq. 5 for each steel and the mechanical properties allow constructing the finite-life models for a given number of cycles to failure.
Figure 55. Experimental and fitted stress-strain curves of both steels.

Table 56. Chemical composition of both AHSS steels according to data sheets.

<table>
<thead>
<tr>
<th>Steel</th>
<th>C %</th>
<th>Si %</th>
<th>Mn %</th>
<th>P %</th>
<th>S %</th>
<th>Al_{tot} %</th>
<th>Nb %</th>
<th>V %</th>
<th>Ti %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strenx 960 MC*</td>
<td>&lt;0.12</td>
<td>&lt;0.25</td>
<td>&lt;1.30</td>
<td>&lt;0.020</td>
<td>&lt;0.010</td>
<td>&gt;0.015</td>
<td>&lt;0.05(1)</td>
<td>&lt;0.05(1)</td>
<td>&lt;0.07(1)</td>
</tr>
<tr>
<td>Domex 700 MC</td>
<td>&lt;0.12</td>
<td>&lt;0.10</td>
<td>&lt;2.10</td>
<td>&lt;0.025</td>
<td>&lt;0.010</td>
<td>&gt;0.015</td>
<td>&lt;0.09(2)</td>
<td>&lt;0.20(2)</td>
<td>&lt;0.15(2)</td>
</tr>
</tbody>
</table>

*Strenx 960 MC is grain refined.
(1) Sum of Nb, V, and Ti is maximum 0.18%.
(2) Sum of Nb, V, and Ti is maximum 0.22%.

Table 57. Chemical composition of both AHSS steels according to data sheets.

<table>
<thead>
<tr>
<th>Steel</th>
<th>Yield strength $R_{eH}$^{(1)} Min MPa</th>
<th>Tensile strength $R_m$ MPa</th>
<th>Elongation $A_5$ Min % Sheet thickness $t \geq 3$mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strenx 960 MC</td>
<td>960</td>
<td>980-1250</td>
<td>7</td>
</tr>
<tr>
<td>Domex 700 MC</td>
<td>700*</td>
<td>750-950</td>
<td>12</td>
</tr>
</tbody>
</table>

– The mechanical properties are tested in the longitudinal direction.
(1) If $R_{eH}$ is not applicable then $R_{p 0.2}$ is used.
*For thicknesses > 8 mm, the minimum yield strength may be 20 N/mm2 lower.

6.7 References in section 6
7 Required tests in tasks 1.2 and 1.4

On this document it has been deduced that to be able to save weight with the Strenx 960 MC steel, attention has to be focused on alternative joining methods, and experimental demonstrations that verify the higher fatigue FAT class, than the FAT classes already existent in the standards for steel joints. Apart from the joints, a comparative S-N plot is needed where the Low and High Cycle Fatigue behaviors of both the Domex 700 MCD, and Strenx 960 MC can be studied. A preview of those S-N plots has been explained on a previous section.

An array of preliminary tests is needed to determine the best (cycles to failure value) of several influencing factors on different types of joint specimens: (1) lap, (2) “T”, and (3) butt joints. The cycles to failure on lap joints may be influenced whether the joint is: (1) laser lap, (2) laser lap joined and High Frequency Mechanical Impact post treatment, (3) Conventional lap weld and TIG dressing post treatment, (4) adhesive lap joined, or (5) Huck bolted lap joint. Two different levels, or types of conditions, will be tested for each influencing factor. The levels may be, for example, different welding speeds, different Huck bolting settings or different glue layer thicknesses.

The Huck bolted joints require to make holes, which may reduce the FAT class, and may introduce another influencing factor, the hole quality. A different set of tests are needed to determine the best (cycles to failure value) of specimens with two holes, one hole cut with certain conditions and another hole cut with other conditions, but using the same cutting technique. The configuration of two holes in the same specimen will be adopted. The influencing factors on the tests with specimens with two holes are: (1) laser cutting, (2) plasma cutting, and (3) mechanical cutting or punching. The outer straight edges of the specimens should be deburred and polished to favor the failure on one of the holes. There is a possibility that hole quality may not be an issue, because contact with the Huck bolt may modify the edges of the hole in their own manner, regardless of the technique used to make the hole.

The influencing factors to be tested in the “T” joined specimens will be the same ones as in the lap joined specimens. The objective with the lap, and “T”, joined specimen tests is to determine which condition maximizes the cycles-to-failure value at a fixed value of stress. The best conditions will then be studied further to establish their fatigue FAT class, and demonstrate with fatigue tests the improvement respect to the charts on the standards. The fatigue tests should involve a complete S-N plot for the chosen joint specimens. The information from the joint specimen S-N plots should help mature the wagon design, and then specific wagon-joint prototypes should be put to a test with a high enough fatigue stress, to further demonstrate the predicted cycles-to-failure.

The tests required to implement damage-tolerant design philosophies in the new wagon with Strenx 960 MC involve testing fracture mechanics specimens, according to standards such as ASTM E399 or ISO 12737 for the determination of plane-strain fracture toughness in mode I. Determination of the stress...
intensity factor on a wagon part will require Finite Element Analysis to determine the geometric influence. The tests for fracture mechanics should involve: (1) the base material with an adequate thickness, and (2) welded material. The integrated Paris equation will used to predict the number of cycles once a crack has been detected. The verification of the cycles predicted requires a test on a prototype joint or wagon part.