Master's thesis

NTNU
Norwegian University of Science and Technology
Faculty of Engineering
Department of Civil and Environmental Engineering

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# Investigation of Railway Switch and Crossing with 35 Tonnes Axle Load

Master's thesis in Railway Engineering
Supervisor: Albert Lau
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# Abstract

Increasing the gross volume and weight of freight on heavy haul lines is a way to increase the capacity and the cost efficiency. In addition, the increased axle load has an impact on the dynamic forces and the loads. The performance on track is also affected, especially in curves and more complex structures, such as switches and crossings.

The following study investigates known theory and previous studies through a literature study on the subject, including work on increasing the performance of switches and crossings. The effects of increasing axle load from 30 to 35 tonnes in a railway switch and crossing are simulated using the commercial Multibody Simulation software GENSYS. Through investigating the railway switch and crossing in all directions in the facing move, the contact patch area, the position and the stresses as well as the wear rate have been evaluated and compared through a 60EI-R760-1:15 switch and crossing divided into switch panel and cross panel. Further, cant and track gauge was altered to find effects of changes in performance and wear using the Pearce & Sherratt wear prediction method, through switch panel and crossing panel.

The obtained results show that a wider track gauge gives significant advantages in the through route in the switch panel. Fewer contact points and better steering in the through route in the switch panel were observed. In the diverging route in the switch panel, however, the the benefits are minor. Benefits in the crossing panel were not as significant, but in some cases it was observed that a wider gauge was beneficial. A narrower gauge was in general not favourable, resulting in more complex contact conditions and also flange contact in most cases in both switch panel and crossing panel. Introducing cant in the diverging route also altered the properties in both beneficial and non-beneficial ways. With more cant, an increased lateral movement in the diverging route was observed, especially in the switch panel, but the cant deficiency was somewhat eliminated. A shallow cant was found to perform better than a higher cant in many cases. Results also show that when introducing cant in switches and crossings thorough investigations into wheel-rail interaction is recommended.

The results and conclusions from this thesis could be used when designing switches and crossings for heavy haul lines in the future, as axle loads are expected to increase.

# Sammendrag

Økning av volum og vekt av godset som transporteres på tungtransport jernbanelinjer er en måte å øke kapasitet og kostnadseffektiviteten. I tillegg har økt aksellast innvirkning på dynamiske krefter og laster. Ytelse og oppførsel på sporet blir også påvirket, spesielt i kurver og i mer komplekse konstruksjoner, slik som sporveksler.

I denne studien undersøkes kjent teori og tidligere studier gjennom et litteraturstudium av emnet, inkludert arbeid gjort omhandlende forbedring av ytelse i sporveksler. Effektene av økt aksellast fra 30 til 35 tonn i en sporveksel er simulert ved hjelp av den kommersielle *Multibody Simulation*-verktøyet GENSYS. Gjennom undersøkelser av sporveksel i alle retninger fremover blir kontaktpunktets areal, dets posisjon, samt påkjenninger og slitasje evaluert og sammenlignet gjennom en 60EI-R760-1:15 sporveksel hvor bytte- og krysspanelet er separert. Videre er overhøyde og sporvidde endret for å finne effektene av endring i ytelse og slitasje ved bruk av Perace & Sherratts prediksjonsmetode for slitasje. Denne undersøkelsen blir gjort gjennom både bytte- og krysspanelet.

Resultatene som ble funnet viser at bredere sporvidde gir signifikante fordeler i det rette sporet i byttepanelet. Færre kontaktpunkter og bedre styring i det rette sporet i byttedelen ble observert. I avvikersporet i byttepanelet er fordelene derimot mindre betydelige. Fordelene i krysspanelet var ikke like signifikante, men i enkelte tilfeller ble det observert at bredere sporvidde var fordelaktig. Smalere sporvidde var generelt ikke fordelaktig, da det generelt resulterte i mer komplekse kontaktpunktstilstander samt hjulflens kontakt i de fleste tilfeller i både bytte- og krysspanelet. Introduksjon av overhøyde i avvikersporet endret også egenskapene på både fordel- og ufordelaktige måter. Mer overhøyde førte til økt sidevegs bevegelser i avvikersporet, spesielt i byttepanelet, men mangel på overhøyde ble til en viss grad eliminert. Det ble funnet ut at ved introduksjon av en beskjeden overhøyde ytelsen var bedre enn ved en høyere overhøyde i mange tilfeller. Resultatene viste også at før man introduserer overhøyde i sporveksler, er det anbefalt å undersøke hjul-skinne interaksjonenene som oppstår.

Resultatene fra denne avhandlingen samt konklusjonene som har blitt trukket kan bli brukt ved planlegging og prosjektering av sporveksler for tungtransport jernbanelinjer, da det er forventet at lastene vil øke.

# Acknowledgement

I want to express the greatest compliments to my advisor, Albert Lau at the Department of Civil and Environmental Engineering, for the support and guidance through my last year of studying. His extensive knowledge on railway engineering and Multibody Simulations has been utterly inspirational, and I want to express my gratitude for all the help I got when obstacles were encountered.

I also want to thank Ingemar Persson from DEsolver, who has always been available to answer my questions about GENSYS and making sure the model worked properly. In addition, I would like to thank the people at Bane Nor for providing requested information about switches and crossings, that has been really helpful in the work of this Masters thesis.

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# 1 Introduction

## 1.1 Problem Description, Objective and Scope

Switches and crossings (S&C) are important parts of the railway infrastructure, providing flexibility to train operations. The way in which they are built, combined with complex load and contact point situations, makes S&C prone to damage and failure. This can cause issues and delays in operations. Numbers from 2015 provided by Bane Nor (Norwegian National Rail Administration) show that $55\,\%$ of the faults reported in the entire infrastructure were related to S&C [1]. More recent numbers show that this is still the case, but with more prediction of failures there are ambitions of lowering these numbers [2]. In addition, in 2015 $35\,\%$ of the annual maintenance budget on superstructure in Norwegian was assigned to maintenance of S&C, although they only make up about $3\,\%$ of the total infrastructure length [3]. In 2019, the maintenance costs related to S&C had risen, even though the relationship between these costs and the total maintenance costs did not change notably [4]. In short, S&C are important but comes with a high cost. Thus, the goal of all infrastructure managers should be to reduce the costs of S&C maintenance and provide a healthy infrastructure for trains to run on.

In recent years, there has been an increased focus on sustainable transport. Politicians are looking to the railway sector, both in Norway and Europe, as an option to take on more of the transportation needed in today's society [5]. Big investments into the railway sector and measures such as the "fourth railway package" [6] for better interoperability between countries and a more healthy competition between operators are introduced, as well as focusing on rail travels in The European Year of Rail 2021 [7]. Many railway lines have reached full capacity in the number of daily passing trains. Thus, operators and infrastructure managers are looking for other ways to increase capacity to maximise the potential. On heavy haul lines, like the Ofoten Line in northern Norway, this objective is handled by increasing the amount of goods, meaning more mass and higher loads [8].

The higher axle loads from increased goods mass can lead to more wear on the infrastructure and on the wheels. This applies to the entire track structure from rail to ballast. Since S&C are prone to damage, an increase of axle load will most likely affect wear rate on the S&C more than on open track. Hence, an investigation of the effects in S&C with higher axle load will be performed in this thesis as well as a study of how it affects the wheel-rail interaction, specifically with an increase of mass from 30 to 35 tonnes. The investigation will be performed by simulating a wagon through a standard S&C in the commercial software GENSYS [9], which is a Multibody Simulation (MBS) tool.

The original research questions presented in the unpublished project assignment [10] was developed as a preparation for this thesis work. However, the questions have been changed to fit the objective of this thesis better. Through revision and changes, the research questions are now the following:

- What are the effects of increased axle load in switches and crossings?
- What can potentially be done to improve performance in switches and crossings?

The plan of this thesis is to look at two track parameters and change these to see differences in performance of the wagon. From a Life Cycle Cost perspective it is know that a optimisation of the curvature and contact/transition geometries are important to reduce the forces and prolong the lifetime of switches [11]. Previously, there has been done a significant amount of studies regarding optimisation of S&C with focus on different elements. Pålsson's doctor thesis [12] investigates track gauge optimisation of the switch panel track gauge, switch rails and crossing design. Other articles treat optimising crossing nose shape [13], switch rails [14], support stiffness [15], elastic track properties in the crossing [16], rail pad stiffness [17] and more.

In most of these studies genetic algorithms have been used to solve the optimisation problems. This methodology will be presented in this thesis, but will not be utilised due to time constraints.

The approach of utilising a genetic algorithms has been used in studies and can be found in, among others, [18, 19, 20]. In [18], an optimisation for track gauge in the switch panel is performed, where MATLAB's [21] built in functions are used to solve the multi-objective problem. The function runs through several generations, picking the best from each and developing them further into the next generation. After each generation, the new design is implemented into the MBS model and simulations are run to calculate forces, loads, wear rates and other parameters that sets the limitations for the optimisation problem. When the requirements are met or the algorithm has converged, the chain is stopped and the best option from the last generation is picked as the optimum solution of the optimisation problem.

Even though many studies have investigated ways to improve and optimise different elements of S&C, there are no studies looking at the effects and possible measures when axle load is as high as 35 tonnes. On the Ofoten Line, there have been test runs with 32.5 tonnes axle load with duration of a year, with the goal to test the infrastructure for higher load. However, wheel-rail interactions at a microscopic level have not been evaluated and should therefore be investigated.

The objective of this thesis is to investigate what happens in wheel-rail-interaction when axle load is increased from 30 to 35 axle tonnes. An investigation on what alterations of track gauge and cant does when measuring wear, contact point positions and contact patch area will also be performed.

# 2 Theory

In this chapter a brief introduction to switches and crossings (S&C), its components and wheel-rail mechanics is presented. Also, explanations to different track design parameters that affect performance and interactions between wheel and rail. Then some information about the state of the art S&C is presented.

## 2.1 Switches and Crossings

There are multiple variations of S&C which answer the need at the location. From the standard S&C, as seen in Figure 1 where one track is divided into two, to big fans where many switches and crossings are connected to each other creating a chaotic layout of rails often seen in connection to big stations or maintenance facilities. The different S&C are presented in this list:

- Standard S&C with one track splitting into a main and a diverging track
- Symmetrical S&C where both new tracks are diverging
- One-sided double S&C where the track splits into three tracks on the same side
- Two-sided double S&C with one diverging track so each side of the main straight track
- A diamond crossing where two tracks meet but with no possibility of change of course
- Single slip where two tracks meet and the course could be changed in one direction
- Double slip where two tracks meet and course could be changed in both directions
- Single crossover where two parallel tracks are connected with the possibility to change track in one direction
- Double crossover where two parallel tracks are connected with the possibility to change track in both directions

- Series of successive S&C where one track is successively split into several tracks
- Track fan where tracks are successively split

[22]

What all of these have in common is the demand for compatibility and usability in all directions. In a standard S&C trains must be able to travel both in the diverging and straight routes, as well as in the facing and trailing moving directions. For the standard S&C this gives four possibilities of travel and for the more complex S&C the number of possible ways of travel depends on their layout.

Notations on S&C refers to the rail quality, radius of the diverging curve and crossing angle. Rail quality is normally given in weight of the rail per metre, with 54 and 60 kg/m being the most common for lines in Norway. The radius of the curve tells how sharp the turn is in the diverging route, where as in most S&C there is no transition curve, meaning that with small radius the speed must be low. The crossing angle of a S&C is the angle between the centre line of the straight track and the tangent line at the rear of the diverging track. Crossing angles from 1:7 to 1:20 are normal.

This crossing angle must not be confused with the tilt of the rails themselves which on open track lies around 1:40 in Norway. Rails in S&C are recommended to be placed vertically, meaning no rail inclination. Since normal rail has inclination the rails need to be twisted, and this happens outside the S&C itself. For new S&C constructions in Norway, the governing body's standard is 1:20 rail inclination if one uses 60EI rail, otherwise no rail inclination is standard for S&C [23].

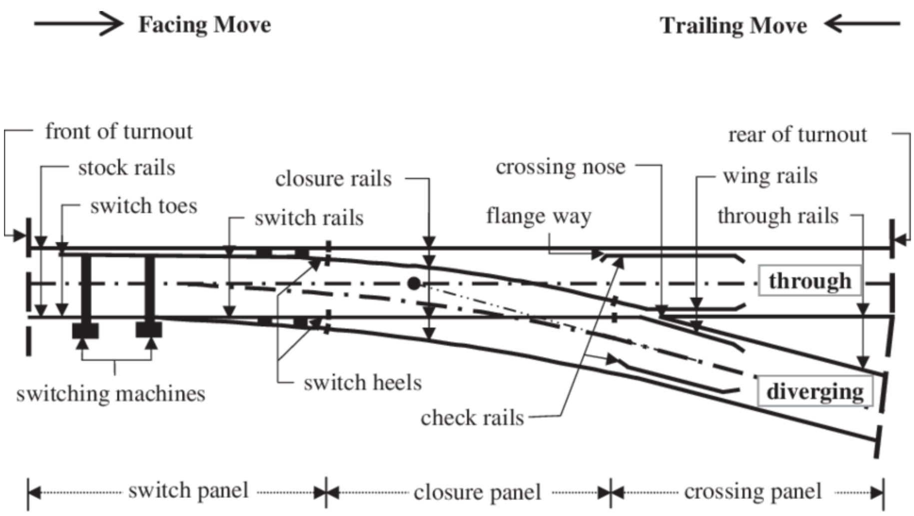



Figure 1: Standard Switches and Crossings with all components [24]

A standard S&C as can be seen in Figure 1 has three sections or panels, which again are built up of many smaller parts. As stated earlier, S&C are prone to damage and faults, and the common damage mechanisms are wear, rolling contact fatigue and large accumulated plastic deformations [25]. These as caused by the discontinuities and changing rail profiles in both switch panel and crossing panel which can be seen in Figure 2. For a better understanding of a standard S&C a presentation with its parts and wheel-rail interaction observed at different stages when a train runs through it is given.

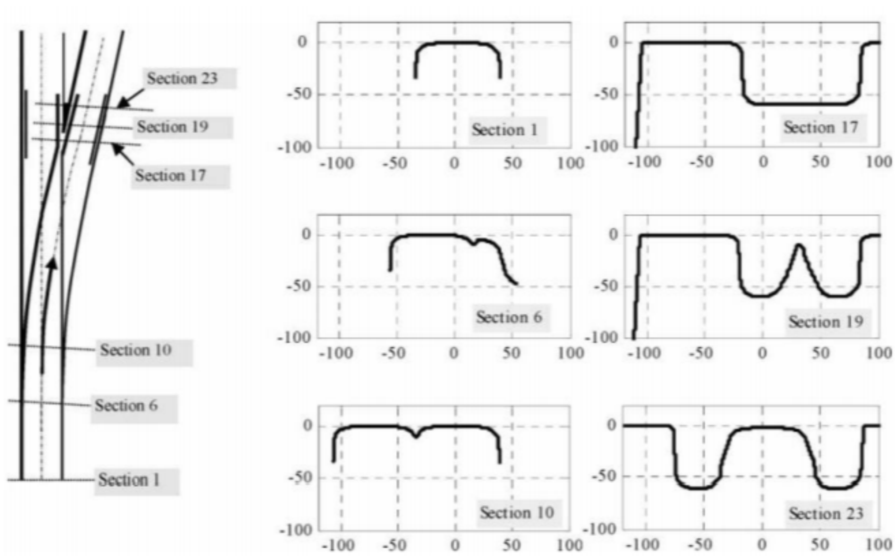



Figure 2: Switches and Crossings cross sections, adopted from [24]

## 2.2 Switches and Crossings Components and Track Design

Here, the most important components will be presented briefly. It is advised to look at Figure 1 for the exact placement of each component.

### 2.2.1 Switches and Crossings Components

Switch blades are what allows the train to change direction on the S&C. They are placed in the middle of track, with one diverging which takes the load of the wheel when the train travels in the diverging route, and one straight, which takes the load of the wheel when trains go straight. The geometry of the switch blades are not continuous, as seen in Figure 2. The two switch blades lie adjacent to the two stock rails, but only one is in contact at the time, leaving a gap for the wheel flange of the wheel to pass through without derailing. The switch blades are controlled by a switch machines that is operated automatically.

Stock rails have standard geometry and run through the whole S&C. In a standard S&C there is one straight stock rail and one diverging. In the closure panel both stock rails and switch blades will have the same geometry of standard rails.

In the crossing panel there is a gap where one wheel for a short duration is in the air with no rail support before hitting the crossing nose. The crossing nose is made of material with higher stiffness as the impacts here require it.

Guard rails or check rails are installed in the crossing panel to ensure safe passage through by forcing the wheel flange through the flange way, pointed out in Figure 1. If the contact points discussed previously are not favorable, the wheelset could change direction and might lead to derailment. The guard rails will prevent this by interlocking the wheel flange, and make it continue on in the right direction. Contact between wheel flange and guard rails are normal in S&C and the need for maintenance is also present for this component.

The wing rails are the extension of the two inner rails in the S&C and are shaped like wings. In the crossing these open up a gap for the wheel flange to pass through to provide the possibility for passing.

### 2.2.2 Rail Head Geometry

In standard S&C the switch blades will have a continuously changing rail head profile which starts with a width 30 mm in the switch toe and ends up with a width and profile like nominal rail in the switch heel. The thinner parts of the switch blade is also exposed to high lateral forces in the switch panel because of the cant deficit.

However, on German high speed lines a different rail geometry has been introduced in order to facilitate for higher speed through S&C. One part of the new geometry is the change of width of the rail head where wheel comes in contact with rail. As speed increases, dynamic forces increases and the wider rail head geometry is better at coping with higher loads [26]. In addition to change of rail geometry the system has other advantages which will be discussed in later chapters.

### 2.2.3 Sleepers

Sleepers influence the track properties in S&C, and they need to be made specifically to fit the increased width of the track. In the closure panel sleepers are lengthened to provide support for all rails in the different directions. Sleepers are recommended to have the same properties as in other parts of the track for the load to be evenly distributed and not have unwanted issues with abrupt change of track stiffness, as can found in connections with the transition to culverts, bridges or tunnels [27]. Some findings from the Norwegian database suggest that the introduction of concrete sleepers on the Ofoten Line is the reason for increased damage in S&C, when the rest of the track has wooden sleepers.

On the Ofoten Line wooden sleepers are used because of low elasticity in the ground, and Pandrol fastening systems are used to ensure that there is no movement when exposed to high forces during braking on the downhills towards Narvik. However, when axle loads are increased the governing body has decided that concrete sleepers will be used in the future, as the wooden sleepers does not have the same capacity. To start using concrete sleepers a thicker layer of ballast is needed for it to be stable and last [28]. The sleepers in S&C are recommended to be similar to the rest of the track so the track stiffness is the same [27], this to avoid the problems regarding transition zones of the track stiffness as can be seen in conjunction with bridges, culverts or tunnels with slab track.

### 2.2.4 Rail Pads and Track Stiffness

Stiffness in rail pads and track influence the bearing capacity and the performance in S&C, and can be optimised as has been done in Lau's research on S&C [19]. The study uses genetic algorithms to find the optimum rail pad stiffness in order to reduce wear in S&C with a better and more even track stiffness.

## 2.3 Wheel-Rail Mechanics

### 2.3.1 Switch Panel

Stock rails will guide trains into the S&C in the facing move, where the wheels hit the switch toe, marking the beginning of the S&C and the switch panel. Here the choice of direction is made, and the train will run in either the through or diverging route dependant on which switch blade is lying tangent and "hugging" the stock rail. The other switch blade is held at a distance to its adjacent stock rail so that the wheel flange can pass through without derailing.

When the wheel first makes contact with the switch blade multiple wheel-rail contact points will be present at the same time. Multiple contact points disturbs the travel of the wheelset through the switch panel. In the diverging route high contact forces and creepage are generated [29], this is due to the change of rail profile as well as the sudden change of curve radius as there are no transition curve or clothoid in a standard S&C. Compared to a curve on the open track, where the conicity of the wheels makes it easier for the train to run with better rolling radius for both wheels, the wheelset will have issues steering through the switch diverging curve. Equivalent conicity decides how much lateral movement is needed for the wheelset to find the optimum rolling radii through a curve, and the wheel with the lowest rolling radius dictates the direction of steering. In the diverging route it is however common that the outer wheel has a contact point on the straight stock rail which creates a rolling radius which is smaller than on the inner wheel, and it will try to steer the wheelset in the opposite direction of the curve. This induces creepage and thus wear, which will be explained in more detail in a later section.

The contact points when the train runs in the diverging route and in the facing move is shown in Figure 3 where the wheel makes contact with the thin rail head of the switch blade while still depending on the stock rail to take parts of the load. This is necessary as the switch blade rail head in standard S&C does not have the required h, often 38 mm for high speed lines [30] and up to 38 mm in standard rails on Norwegian lines [31].

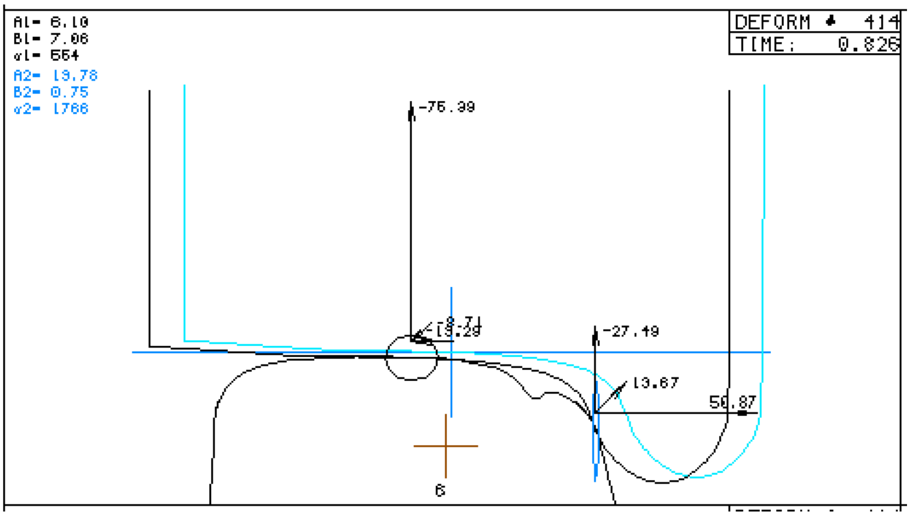


Figure 3: Two contact points in the diverging route in switch panel when transitioning from straight stock rail to diverging switch blade

However, it is not only in the diverging route there might be issues. When a train is guided in the through route it will be slightly shifted laterally due to the different contact point and rolling radius differences [30]. The lateral movement is common on open track especially after curves, where the wheelset move from side to side in a sinusoidal movement to find again optimum rolling radii. This sinusoidal movement can be seen in Figure 4 and is called hunting [32]. This hunting movement of the wheelset on the straight rails will again be affected and decided by the suspension systems of the wagon, meaning that the critical speed will be lower for a train with softer suspension compared to a train with hard suspension. This is discussed in detail in [30] and will also be discussed more closely in section 2.5 where a gauge widening is proposed as a solution to the hunting movement problems.

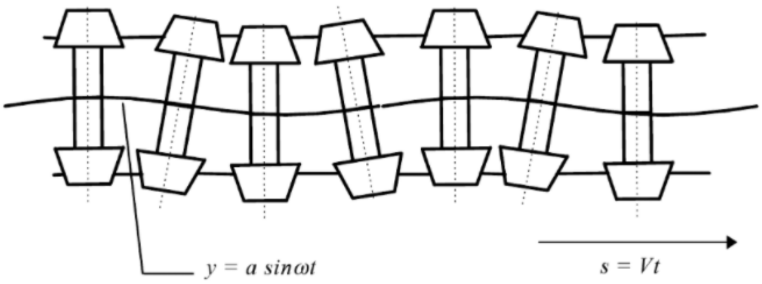



Figure 4: Sinusoidal hunting movement due to conicity of the wheels [32]

Multiple contact points are also common to happen in the straight route in the switch panel. These unfold in a similar way as in Figure 3 and also here can it be more than one contact between wheel and switch blade simultaneously. This is what causes the hunting motion explained in the previous sector. The contact point with the smallest rolling radius, which is between stock rail tread and outer part of the wheel, tries to steer the wheelset in one direction before the contact point vanishes as the diverging stock rail bends off and leaves only the straight switch blade and stock rail in contact with the wheelset.

### 2.3.2 Closure Panel

Between the switch and crossing panel the closure panel with closure rails is located. It can be seen in Figure 1 in the transition from switch to crossing panel, and in Figure 2 from cross section 10, and it continues until the start of the crossing panel. In the closure panel the switch blades have "developed" standard rail heads and have the same bearing capacity as standard rails. All loads from the train is carried by the switch blade and opposite stock rail, and only one contact point between the wheel and switch rail is observed. This behaviour reminds strongly about a curve in the open track, but with cant deficiency there are more lateral force being induced.

In the closure panel sleepers are lengthen to provide support for all rails in the different directions. Sleepers are recommended to have the same properties as in other parts of the track for the load to be evenly distributed and not have unwanted issues with abrupt change of track stiffness as can be seen in connections with the transition to culverts, bridges or tunnels [27].

### 2.3.3 Crossing Panel

After the closure panel lies the crossing panel. It comprises of wing rails, crossing nose (often referred to as frog) and guard rails, all with crucial tasks securing trains passing through.

In the crossing panel it is common to observe multiple wheel-rail contact points, and steering mechanisms will be disturbed. In common crossing there is also a discontinuity in the rails meaning that for a short amount of time the wheel is in the air with no contact with rails, thus leading to high impact forces on the crossing nose itself. This is why the nose is often made out of steel with higher stiffness than normal rail, manganese steel is the most common but other options are also being used. On heavy haul and high speed lines a movable crossing nose could be introduced to reduce this high impact and damage to the nose. The movable nose will like the switch blades be pressed towards the active closure rail so that when the train runs over it there will be no gap and a continuous rail. The fixed and movable crossing nose can be seen in Figure 5 Movable wing rail with a fixed nose is another option to eliminate the gap in the crossing panel [27].

On the Ofoten Line work has been done from 2014 to change old S&C so they will be equipped with movable crossing noses, but there are still many lines with heavy axle loads running on standard fixed crossing noses. Crossings with movable noses needs more maintenance, but are necessary when the dynamic loads increases. In Norway, lines with expected axle loads exceeding 22.5 tonnes is recommended to have either movable crossing nose or or wing rails [23]

(a) Fixed crossing nose [33]     (b) Movable crossing nose [34]

Figure 5: Fixed (a) and movable (b) crossing noses

Wheel contact with wing rails are common in fixed crossings where the contact point with the wheel tread it on the wing rail at the same time as the wheel hits the crossing nose, which may lead to steering difficulties and are the reason for the existence of guard rails. Contact point with wing rail and crossing nose is shown in Figure 6.

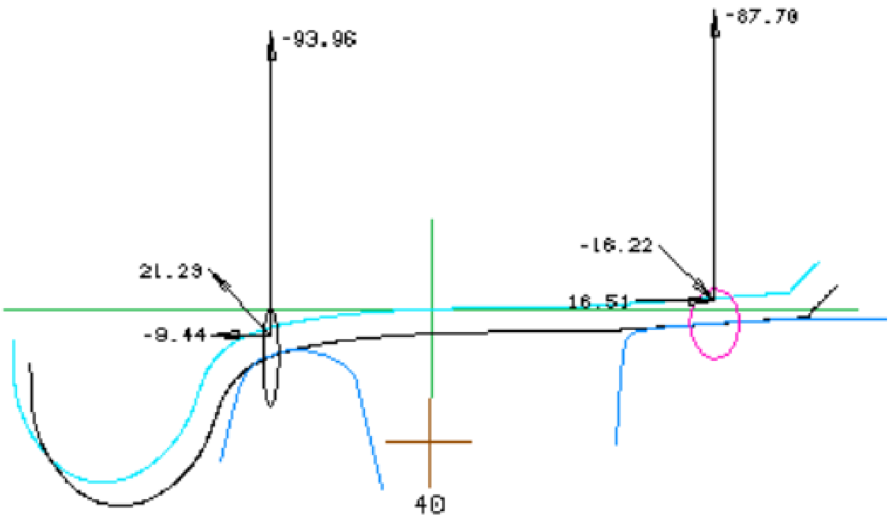



Figure 6: Contact point with wing rail and crossing nose simultaneously in the crossing panel

### 2.3.4 Steering and Rolling Radius of the Wheels

Railway wheels have been conically shaped for a long time, which help the train steer through corners without too much slip when the wheels are fixed to one axle. In curves the wheelset moves laterally so the two wheels' effective rolling radius is changed, so even though the rotational speed is the same for each wheel the outer wheel travels longer since the rolling radius is bigger [32]. As explained in section 2.3.1 the lateral movement needed is decided by the equivalent conicity of the wheels, where more conical wheels needs less movement, but a shallower wheel, or in a wider track gauge case where both contact points are on the tread more lateral movement is required for the wheels to find their optimum rolling radius. The situations in the switch and crossing panel where more than one contact point on one wheel is present, the rolling radius is disturbed and the difference on one wheel messes up the steering. Multiple point contact situation leads to poor steering, creepage which induces wear [18].

From Pålsson's thesis [29] it is found that equivalent conicity is the wheel parameter that corresponds best to damage in switches and crossings. The equivalent conicity is calculated as seen below, and it shows the relationship between lateral displacement and rolling radius difference in the steering mechanism of the wheelset.

$$\lambda_{eq} = \frac{r_r - r_l}{2\Delta y} = \frac{\Delta r_r - \Delta r_l}{2\Delta y} \tag{1}$$

In [12] and in other literature it is shown that the equivalent conicity is a non-linear function of lateral displacement, which can lead to sudden jumps in the contact point position on the wheel. This effect could also be enlarged if the wheels are worn and rolling radii does not correspond to the lateral position of the wheel.

Nominal rolling radius is the rolling radius a wheel has when the contact point with the rail hits in the middle of the tread. This is regarded as the 0-point for contact point position on the wheel and any offset from this point will make the wheelset turn, with turning magnitude being decided from the equivalent conicity. If both wheel-rail contact points are equal to the nominal running circle on a straight track, and conditions are optimal in terms of no irregularities in track or wheel, there will be no lateral movement and the train will run in a perfect straight line. In Figure 7 the nominal running circle and its contact position on the wheel is shown. Different wheels are geometrically different with altering width and angle of conicity, but the concept of nominal running circle is similar on all wheels.

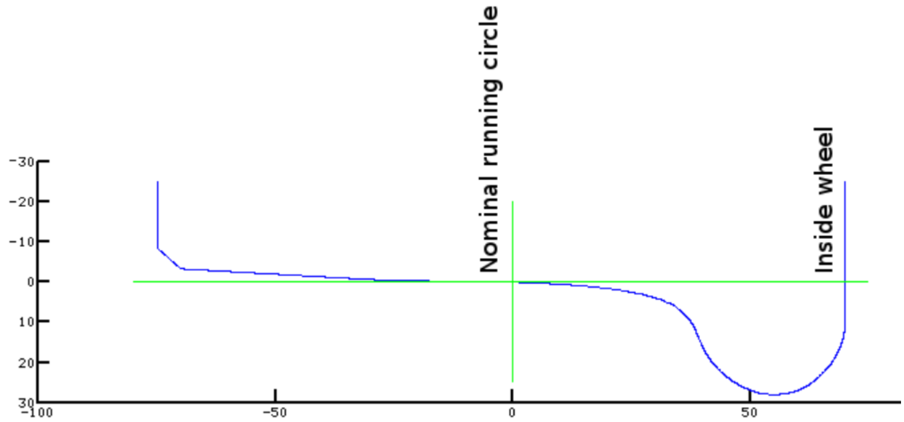



Figure 7: Contact point position of the nominal running radius [9]

## 2.4 Design and Layout of S&C that Affects Performance

### 2.4.1 Cant

Even though S&C are curved there are usually no cant to help the vehicle cope with the increased lateral acceleration in the diverging route. This makes the uncompensated lateral acceleration greater and forces trains to travel at lower speed through standard S&C. In some S&C lowering the inner rail of the diverging track has been done to create a cant, but this is not standard practise, as it might effect performance of the vehicle in the latter part when the track is leveled again, or if the diverging tracks immediately turns back to make it parallel to the straight track. The planar nature of the S&C makes it difficult for cant to be present [12]. On some high speed lines however a cant is introduced to reduce the uncompensated lateral forces and increase quality of riding in the diverging track, but there are difficulties related to this that needs to be solved [35], like space and moving parts. Cant with lowering the diverging stock rail is often limited to only 15 mm of superelevation, so the effect of cant is said to be not significant.

### 2.4.2 Clothoid & Transition Curve

Clothoids or transition curves are curves that has a linear change of curvature, meaning that the change of curvature is constant. At the entry and exit of curves on the open track a clothoid is placed to reduce the jerk (time derivative of side acceleration) which increases comfort for passengers and reduces stress in wheel-rail contact points. This is however not normal in S&C often due to lack of space, and the sudden change of curve radius induces high jerk.

In some high speed lines clothoids are in use, and with special designs, one has seen a significant performance increase and that S&C which incorporates clothoids are superior to those with constant curve radius [36].

In Norway one finds transition curves in connection to S&C on the high speed line to Oslo airport Gardermoen, where high speed trains travel through S&C at 100 km/h [27]. Clothoids take up a lot of room, and in general the issue with jerk is solved by reducing the speed through the diverging route, reducing the dynamic forces with lower speed.

### 2.4.3 Track Gauge

In Norway, as well as in most European countries track gauge is standardized at 1435 mm. In some old Soviet states the gauge is bigger. The Norwegian Railway government have given limits for the different panels and when maintenance is acquired. The limits is shown in Table 1

From the table for switch and closure panel one can see that it recommends a minimum track gauge of 1432 for the highest quality and 1430 mm for the lower quality classes, which is relatively close to standard 1435 mm. Quality classes is decided from the line speed from track geometry. When it comes to widening it is given that maintenance is needed when the gauge is from 6 - 25 mm wider than normal, depending on the quality and strictness of standards on the line. The limitation for maintenance is stricter through the crossing panel. Limits for crossing panel are maximum 3mm narrower and 4 - 15 mm wider, all values can be seen in Table 1. The different quality classes are given with respect to the speed at the given line. On the Ofoten Line speeds are low, but on heavy haul lines a high standard is needed because loads are high.

| Quality Class | Maximum limit Switch and Closure [mm] | | Minimum Limit Crossing [mm] | |
|---|---|---|---|---|
| | Minimum | Maximum | Minimum | Maximum |
| K0 | 1430 | 1441 | 1432 | 1439 |
| K1 & K2 | 1432 | 1445 | 1432 | 1443 |
| K3 | 1432 | 1450 | 1432 | 1443 |
| K4 | 1432 | 1455 | 1432 | 1447 |
| K5 | 1432 | 1460 | 1432 | 1450 |

Table 1: Norwegian maintenance requirements for track gauge in S&C [31]

Track gauge in S&C are shown to have a large influence on the dynamics and the damage characteristics. Meaning that changing and optimising track gauge with objective to minimize damage could be economically beneficial for the infrastructure managers responsible for maintenance. In his doctoral thesis, Björn Pålsson used genetic algorithms to optimise track gauge through S&C using a switch blade with stock rail characteristics. He found that in general a widening of the gauge will reduce damage and need for maintenance, and give longer life to crucial components [18].

Track gauge alterations will influence the wheel-rail contact interaction and the steering. The lateral movement will differ with different track gauges as the wheelset might need to move more or less to find the optimum rolling radii to navigate through the diverging route. In the through route it has been found benefits from widening the gauge in the switch panel to guide the wheelset passed the switch blade that is not active, and eliminating unwanted contact points as well as hunting [30], these designs will be presented in a later chapter.

From field measurements, which later was used to create a digital model of a S&C [25], there was shown that track gauge altered slightly through the S&C. With a maximum widening of 3 mm and a narrowing of 2 mm. Small irregularities are common both in S&C and in open track, but the limit for when maintenance is needed are more strict in S&C. This is due to the already high maintenance cost and track irregularities will induce even more damage.

## 2.5 State of the Art S&C

As already stated S&C are the cause of many faults and delays in the railway sector. Many researches have made it their goal to make S&C perform better with fewer faults and less need for maintenance, with reduced forces and better ride quality. Much of the work has gone into improving high speed S&C where the diverging route speed is the limiting factor. By increasing this speed the overall speed of operating trains could be higher. Also, a lot of work has been put into reducing the wear and damage to wheels and rails by changing vehicle and track parameters to find optimum solutions [18, 19, 20, 36, 37].

### 2.5.1 FAKOP

The German company Voestalpine [26] specialises in S&C and in 1991 a new design was introduced, based on BWG's previous design. Fahrkinematische Optimierung (FAKOP) or Kinematic geometry optimisation (KGO), where the idea is to reduce dynamic forces in the switch panel. The system is used on many high speed lines. The conical shape of the wheels are exploited to make the wheelset steer away from the switch blade in the through route by reshaping the straight stock rail creating a wider track gauge. The widening is shown in Figure 8 below, with the lateral difference in contact point position also shown. By doing this the are fewer contact points in the transition from stock to switch blade in the through, in addition it also helps with steering in the diverging route. Because the gaps between wheel flange and switch blade rail are greater with this design also the rail heads of the switch blade can be wider and thicker which makes them more robust and capable to deal with the loads. The main goal of the FAKOP design is to reduce the stresses experienced in the through route and eliminate the negative effects discussed in section 2.3.1. The FAKOP design have shown to be effective when it comes to reducing forces [30]. In Oswald's paper on the KGO/FAKOP system, simulations showed that FAKOP design with clothoid curve (curve with linear variation of curvature) also is superior compared to S&C without clothoids [36].

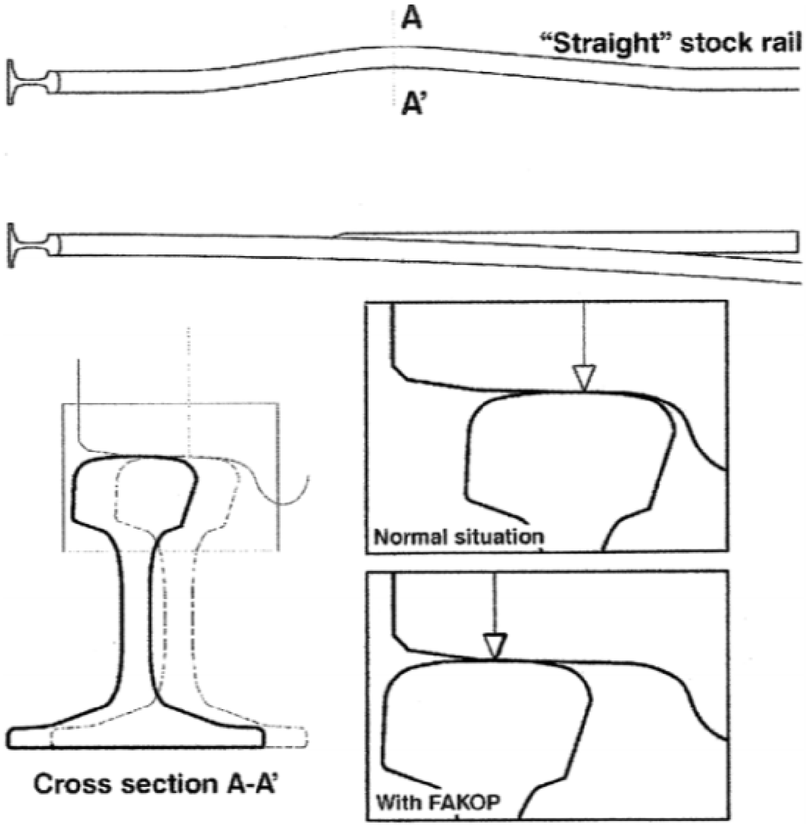



Figure 8: FAKOP designed switch panel with laterally displaced contact point [30]

### 2.5.2 CAFTERSAN

The CAFTERSAN design is based around the same idea as the FAKOP/KGO, with changing the lateral position of the wheel-rail contact point utilising the equivalent conicity of the wheels guiding them in the right way. However, the CAFTERSAN method does this in a different way, which is possible to incorporate for already existing S&C. By grinding away the inner parts of the stock rail the contact point is moved outward as shown in Figure 9 and the same effect of moving the contact point is accomplished.

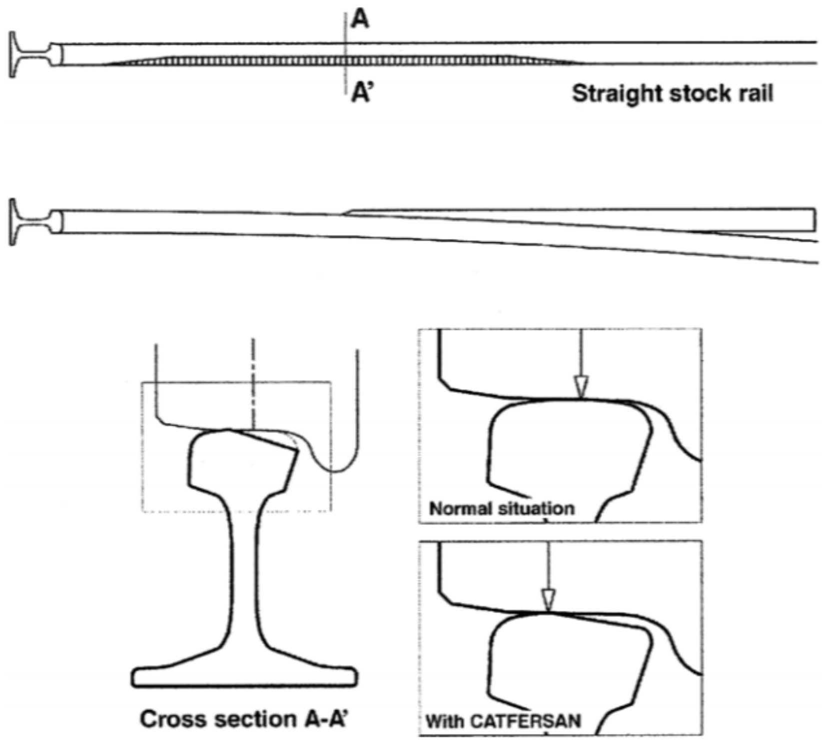



Figure 9: CAFTERSAN designed switch panel with laterally displaced contact point [30]

Also for this design studies have found improvements when it comes to performance compared to standard S&C in the through route, but there are some implications in the diverging route due to the new rail head profile. However, since speed is lower in the diverging route the benefits are bigger than the drawbacks compared to standard S&C [30].

# 3 Methodology

## 3.1 Numerical Model

To simulate dynamic wheel-track interaction the commercial Multibody Simulation software GENSYS [9] was used. In the software masses and springs represent the different parts of the wagons and track, and couplings make up the connections between them. The suspension system of wagons are represented by springs with different stiffness. GENSYS was developed in Sweden in the 1990's following the need for better simulation and calculation tools. The software has been used in a great amount of studies such as [18, 20, 24, 25, 38, 39] and many more. GENSYS allows time-step simulation and calculation of wheel-rail dynamics, and could also be used to visualise wheel-rail interaction on a microscopic level.

SIMPACK is another Multibody simulation program that is used for instance in [40], but because NTNU has an agreement with DEsolver and there was great knowledge and competence in GENSYS at the institute it was decided to use GENSYS.

### 3.1.1 Vehicle Model

A freight vehicle model with three piece bogie configuration, similar to the Fanoo040 wagons used on the Ofoten Line, is used in the simulations. One wagon has two bogies and a total of four axles.

The vehicle model has been validated by researchers at KTH in Sweden on tangent and curved track, using GENSYS simulation and comparing the results with measurements done on track in Sweden [41]. The wagon is built up by masses and springs. For the simulations, only one wagon was used for all runs, the focus of this study is to look at the dynamics and interactions at the first wheelset in addition to a need to keep the simplicity of the model.

The speed was constant at 60 km/h for all simulations. All simulation scenarios were run with weights of both 120 tonnes and 140 tonnes in total. Distributed over the four axles this corresponds to 30 tonnes and 35 tonnes in axle load. The wheels used for all simulations are standard S1002 wheels that are not worn. These wheels have also been measured using the MiniProf tool.

The suspension characteristics were kept the same through all simulations, and is something others might look into. From articles reporting about test running with heavier axle load on the Ofoten Line it is stated that the wheel stiffness has been changed when trains are run with higher axle load [42, 43]. This was not done in the simulations in this thesis. The model has a two stage suspension system, with primary suspension between wheel and bogie, and secondary suspension between bogie and vehicle.

### 3.1.2 3-P Bogie

Three-piece bogies have a simple and robust design, making them easy to maintain and the initial cost is also low [41]. A conventional three-piece bogie has weaknesses when it comes to stabilising hunting motion on straight track and in curves the steering mechanisms are not good [44]. The model of the bogie is built up by rigid bodies with mass connected with couplings that has properties of dampers and springs.

The bogie is connected to the frame with linear spring-damper elements [24].

### 3.1.3 Track Model

The modelled turnout is a standard 60EI-R760-1:15, meaning nominal rail profile 60EI without rail inclination, curve radius of 760 metres and a turnout angle of 1:15. The model is similar to the one used in [18, 19, 24, 25] and is based on measurements on a turnout in Sweden where the measuring tool MiniProf was used [45]. Some 80 cross sections throughout the turnout was measured before interpolating between the values to create a continuous model of track [25]. The measurements were made using Miniprof [45] tool. In Figure 10 a similar interpolation is shown using MATLAB [21].

For all simulations there were not included any track irregularities and a perfect track is assumed.

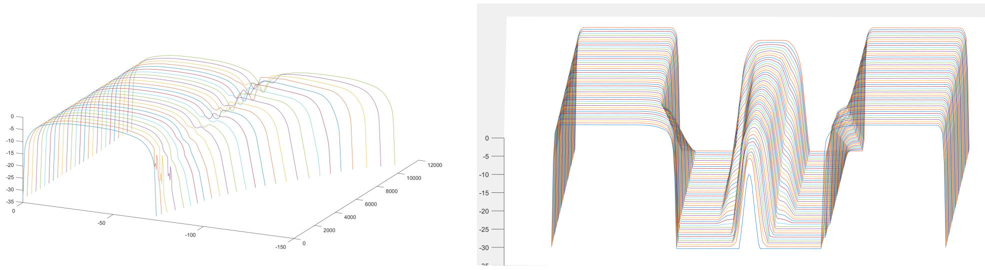
The model was split into switch panel and crossing panel. This is due to the difference in weight of the rails, as the crossing nose, where two rails are combined has double the weight of the stock rails. Therefore the simulations was done for each panel separately. This could have been solved and a combination of the two panels would have been possible, by changing the weight continuously while the wagon runs through, but this would make the simulation time increase significantly.

### 3.1.4 Wheel-rail Contact Model

In previous studies were GENSYS was used to evaluate wheel-rail interaction a pre-process for contact point function (KPF) was needed. This was done by calculating each cross section separately, laterally moving it across the wheel to prepare wheel-rail interaction before running the model itself. This was a time consuming job, which was not necessary in this thesis following the introduction of the "creep_fasim_4" contact model which evaluated contact between wheel and rail simultaneously as the model were running. The new contact model also allowed for 10 contact points, which was helpful when simulating S&C where multiple contact points are present at the same time.

During the preparation to this thesis a project assignment was submitted in the winter semester of 2020, this explains the basic concepts and a plan for this master thesis [10]. The project assignment suggests that optimisation using genetic algorithms was the best way to answer the research questions, and to find the best solution for switches and crossings in the future when train's weights increased. Due to time constraints this was not performed following a lack of competence in the modelling software, which made setting the model up and making all scenarios run a more time consuming task than planned. It was therefore decided on a more simple approach focusing on two variables and looking for the effects of changing these two.

To make the model work in all directions the S&C model needed some coding. Initially the plan was to extract and manipulate all cross sections, making simulations of running in the through route possible, as well as trailing moves. This was attempted done in MATLAB by changing the data points of the cross sections to eventually flip the S&C to make the train run straight through. These manipulated files were fed into GENSYS for simulations. In Figure 10 the cross sections with linear interpolation is shown. However, this was not the most efficient method as one could simply change a couple of lines of code to make the train run straight. The only downside was that now the S&C was effectively a left handed S&C, but this did not matter as it was symmetrical and the same effects were observed in left and right turning S&C. When evaluating results this had to be taken into account.



(a) Switch blade (b) Crossing nose

Figure 10: Switch (a) and crossing nose (b) plotted using MATLAB

A lot of effort was put into making the simulation work for all directions, which in the end turned out to be eight, because of the division of the S&C into separate switch- and crossing panel. For each panel both diverging and straight in both facing and trailing move were simulated, which gave a great amount of results to be evaluated. There were some issues with the trailing move simulations and uncertainty towards the accuracy of these results.

In total there were performed 144 simulations with altering track gauges and 32 simulations with cant alterations. There were only looked at cant in the diverging route.

The results that were focused on are the wheel-rail contact point conditions, contact forces and the wear index. It has especially been looked at what happens to the contact patch properties and its path when weight is added to the wagon. Later, the investigation began and included going through all simulations systematically, and looking at what had happened when track properties were changed. In the trailing move simulations, a lot of sway and longitudinal movement of the car body on the bogies made for too much uncertainty to include the results in this thesis that were not anticipated. The movement and sway is most likely caused by the suspension being too soft and also the 3P-bogie is known for not being the most stable. It was decided to not present any results from the trailing move simulations, as this needs further work to be presented with less uncertainty.

In a validation study by Kassa and Nielsen [25] it was shown that the simulations struggled to reproduce the dynamic interaction with high frequencies. Because one has seen that the frequencies of the dynamics in crossings are very high, the simulations in the crossing panel was performed with possibilities to register frequencies up to 2500 Hz.

In the switch panel one does not expect to have the high frequency dynamics as in the crossing panel and for time saving purposes the frequency registration was at a much lower level.

The wheel-rail contact point positions shown in the results chapter are compared to the nominal running circle contact point, which is discussed in section 2.3.4.

## 3.2 Wear Estimation

In many of the studies concerning performance and damage in S&C, one of the main factors that are given a lot of attention is wear. Wear is one of three main damage mechanisms happening in S&C, in addition to rolling contact fatigue and plastic deformations accumulated over time [29]. However, there are more than one way to estimate wear from simulation results, and different methods have been used in different studies.

As described in Wang's book on high speed S&C [35], it is important to design and plan S&C based on wear, as wear will be a driver for more damage on track and wheels which will lead to more noise, greater risk of derailment and less motion stability [24].

In a study by Pombo et al. [46], three functions for predicting wear is presented as the most common, and they are applied to calculate the worn material of railway wheels. The article discusses the advantages and disadvantages of all three methods. In general, wear prediction functions are based on using tangential and normal contact forces as well as creepage (relative velocity normalized by the rolling velocity [47]) as input data, and so are the three presented in the study. A thorough presentation of the different methods and their benefits and usage can be found in [46].

Different studies uses different methodologies of estimating wear. For instance, in Pålsson's study of damage in S&C [29] the Pearce and Sherratt method [48] is utilised, where only creepage and creep forces are investigated. The estimation of wear i performed in a sophisticated way with a great number of simulations and updated wheel and rail profiles using finite element method for more accurate wheel-rail profiles along with deterioration in simulations. This method is similar to other parts of his thesis, where the wear estimation are run in loops to evaluate and find a optimum track gauge with the use of genetic algorithms [18]. Wang et al. [40] and Johanson et al. [38] use a different method for wear estimation based on Archard's wear, which also is presented in the study by Pombo et al. [46].

### 3.2.1 Wear Function from Pearce and Sherratt

In 1991 Pearce and Sherratt introduced a method for predicting wear which is based on the idea that material lost is proportional to the energy dissipation in the contact area, meaning the product of creepage and creep forces [48]. Material lost is dependant on the severity of the wear either mild, severe or catastrophic. From Equation 2 it is shown how $T\gamma$ is found, which is the wear index. In switches and crossings with multiple contact points the $T\gamma$ needs to be summed for all contact point's wear index. This value is given with unit of Joule/metre and is a measurement of energy dissipated per metre. In the equation cp represent each contact point that is present in the given situation. "Fnx" is longitudinal creep forces, "nux" is longitudinal creep, 2Fny" is lateral creep forces and "nuy" is lateral creep.

$$T\gamma = cp_{Fnx} \cdot cp_{nux} + cp_{Fny} \cdot cp_{nuy} \tag{2}$$

Table 2 shows how the amount of material lost for different severity of wear is calculated. Material lost is related to whether the wear is mild, severe or catastrophic, and in cases with flange contact one is often in the catastrophic realm.

Pearce and Sherratt

| | | | |
|---|---|---|---|
| T$\gamma$ <100 | [J/m] | materialloss 0,25 T$\gamma$ / D | $[mm^2/km]$ |
| 100 <T$\gamma$ <200 | [J/m] | materialloss 25 / D | $[mm^2/km]$ |
| T$\gamma$ >200 | [J/m] | materialloss (1,19 T$\gamma$ - 154) / D | $[mm^2/km]$ |

Table 2: Material loss from Pearce and Sherratt [46]

For wear estimations in this study the Pearce and Sherratt method, with the wear index formula seen above with wear index being dependant on creepage and creep forces was used. In one part of Pålsson's thesis were the damage in switches and crossings were investigated a similar method for calculating the wear index was used, as it was found that it was not much difference in peak wear from other methods and this was chosen for its simplicity [29]. The comparison to other wear estimation methods in his study was done by comparing the Pearce and Sherratt method to methods including the contact patch area.

All $T\gamma$ present in the simulations for all contact points were later summed to the total wear index between the given wheel and rail. For simulations through both the switch and crossing panel up to four/five contact points were observed throughout the simulation and added together. The area under the wear graphs in the results chapter are related to energy dissipated.

## 3.3 Cant and Track Gauge Alterations

Most standard S&C does not have cant due to the geometrical layout and moving parts which makes it difficult. In the diverging route this leads to a cant deficiency, meaning that the lateral acceleration through the curve is not compensated for, as it is in a curve on the open track.

In this thesis two cant scenarios are compared to standard cant which is 0 mm on most S&C. The two cant heights chosen is 37 mm and 75 mm. The different cant scenarios are simulated by gradually changing the height difference of the rails to 37 mm and 75 mm. On high speed S&C in China there is a limit for how much cant deficiency the track might have from lateral acceleration, with 75 mm as the maximum limit [35]. This is the reason for choosing 75 mm as the highest cant in these simulations. A short transition ramp was placed before the switch panel, as is usual for curves on the open track. 37 mm cant is chosen as a more shallow and an alternative that is easier to introduce on track. There was not included a clothoid in the simulations, so jerk in the diverging route will still be relatively high. The reasoning behind this is that the switches and crossings usually does not have clothoids, apart from some high speed S&C.

As previously stated, the crossing panel and switch panel are run in separate simulations in this thesis. Therefore, a cant in the switch panel will not affect the lateral displacement in the crossing panel when running through the S&C in a facing move. When separating the two panels the simulations will initiate with the same lateral displacement as if there were no cant. To make sure that the effect of cant is present in the crossing panel, the simulations are started early in the closure panel for the wheelset be laterally offset similarly to the exit of the switch panel. This also gave the wheelset time to find its optimum contact positions and in this way the cant effect from the switch panel is accounted for in the crossing panel.

In the scenarios with different track gauge there were done changes by 2 mm intervals, from 8 mm narrower to 8 mm wider gauge, relative to standard of 1435 mm. The reason for altering track gauge has to do with the steering mechanisms of the wheelset through the diverging and through route. By widening the track it has been seen from previous studies that the wheels easier find the optimum rolling radii and navigate better through the switch panel and into the closure- and crossing panel. FAKOP and CAFTERSAN designed S&C makes use of this kind of method to reduce damage and enhance performance in switches and crossings, and in this thesis it has been investigated whether similar effects could be achieved by a general widening with heavier axle loads (35 tonnes). The reasoning behind narrowing the gauge is to see what effects it has on wheel-rail interaction, and whether it could be in some cases beneficial. From section 2.4.3 it was shown that narrower gauge is not wanted S&C. By including an investigation into scenarios with narrower track gauge it could give a better understanding of S&C dynamics with different gauges.

## 3.4 Post-process of Data

GENSYS generates about 1300 arrays of information for each simulation ran as results. To be able to analyse and evaluate this data a script using the Octave program was used. Result files were extracted from GENSYS and the information needed was stored as variables in lists and cells. Then a search through the data stored was needed to find, extract and plot the necessary information for the specific case.

Simulation time was generally not a big problem for each individual simulation. However, since so many simulations were performed the total time was longer than anticipated. With the use of a remote workstation with high capacity and 36 processors the computation time for a single simulation was in the area between 5 to 15 minutes, depending on whether it was in the switch or crossing panel. The simulation in the crossing panel took longer time as the frequency was set significantly higher there. When running multiple simulations simultaneously, for instance 18 scenarios with different track gauge and weight, the time could go up to 3-5 hours, also depending on if the simulations were run in switch or crossing panel.

The use of a remote workstation with a powerful engine and 128 GB with 36 processors made the computational time half compared to a 16 GB RAM computer with 8 processors.

All results for all possible contact points are calculated and given to the user, even though the values are equal to zero. A significant amount of effort has therefore gone into evaluating results and present them in the most clear and understandable way for the reader.

# 4 Results

The focus of this thesis is to look at results in the first wheelset as this is most critical because it steers and leads the rest of the wheelsets through the S&C. After simulation was done the results from the second wheelset as well as the first wheelset of the second bogie was evaluated and it was not found any significant differences and it was decided to only present results from the first wheelset of the leading bogie in this thesis.

For the results, an investigation of whether the changes of track gauge in the whole S&C and cant in the diverging route has any effect when it comes to reducing the dynamic forces, impact loads and rate of damage when axle load is increased from 30 to 35 tonnes. The three main parameters that will be evaluated in this chapter are the contact patch area and its time of occurrence, as well normal contact forces between wheel and rail and wear index. Wear index is estimated with a method taking in creepage and creep forces. As simulations have been run in both directions through both the switch panel and the crossing panel we will look at these separately with most focus to what happens on the switch blades in the switch panels, as it will be most prone to damage and highest dynamic loads are observed here, and the crossing nose in the crossing panel. The trailing moves have not been taken into consideration as these simulations were insufficient.

Initially, in the comparison between simulations with loaded and unloaded trains we can see a changes in both the forces, wear index and changes in the contact patch path along the S&C. It it important to show the difference as this lays the foundation for further investigations and alterations to the track model.

## 4.1 30 Tonnes Axle Load vs. 35 Tonnes Axle Load

In the switch panel there are differences in both the diverging and through route when the axle load is increased. From simulation results the value for estimated wear increases on the switch blade in both track directions, in the facing move. Figure 11 shows the two cases in the diverging facing route on the switch blade, with 30 tonnes and 35 tonnes axle load. The difference is significant and tells us that the creepage and creep forces are higher here when the axle load is increased.

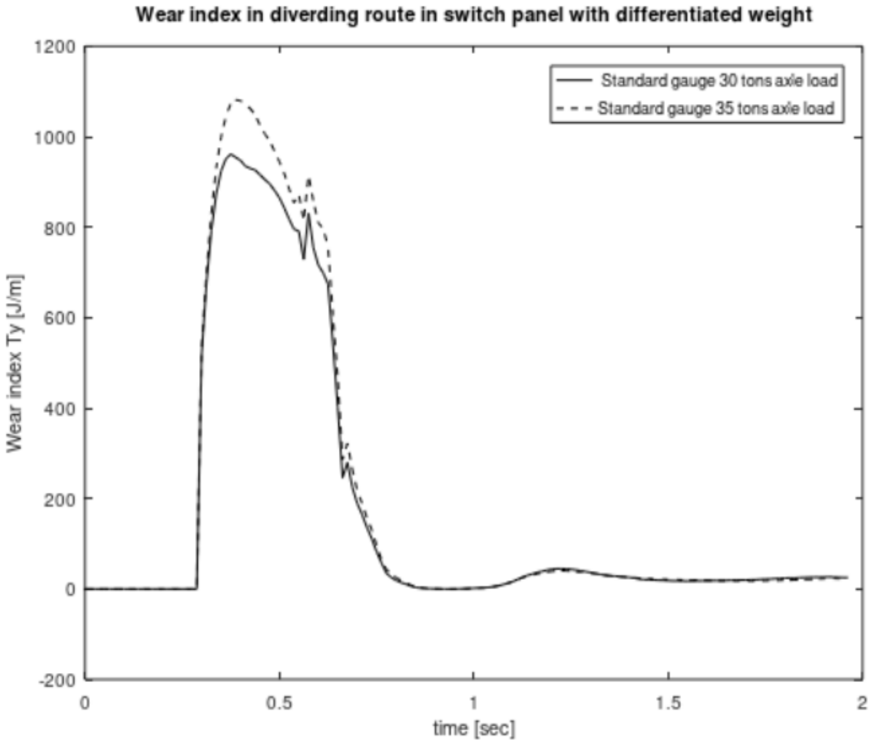



Figure 11: Difference in wear rate on the switch blade in the diverging route and facing move when axle load is increased from 30 to 35 tonnes

In the diverging facing move there is also a difference in the contact point position when the load is increased which can be seen in Figure 12. This implies that when the load is changed, different wheel-rail dynamics comes into play and different stresses will be experienced with the new loading situation.

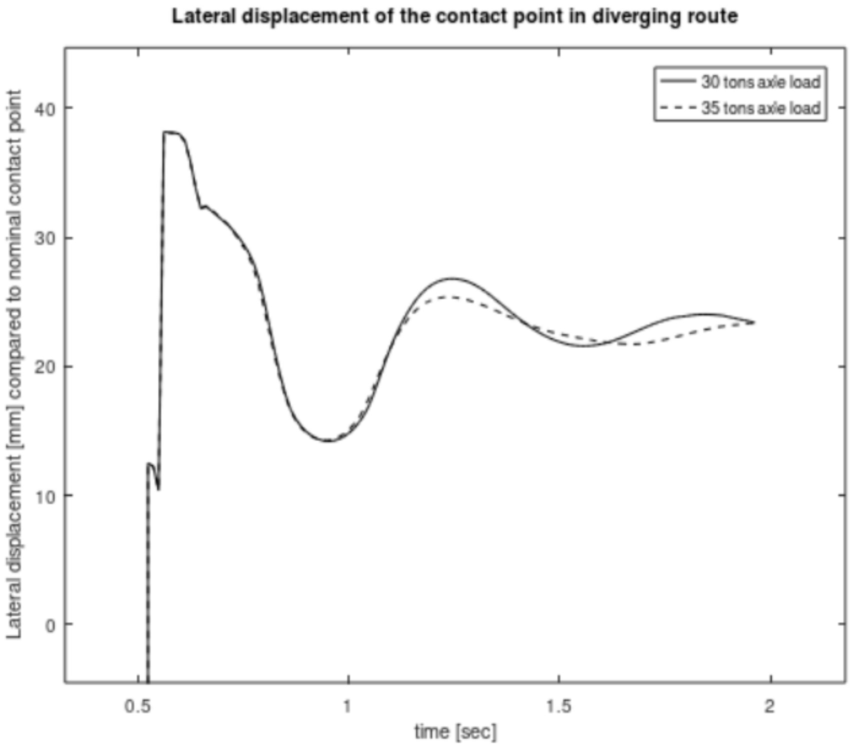



Figure 12: Position of contact point with 30 tonnes and 35 tonnes axle load. Offset from 0 on y-axis is lateral displacement relative to nominal contact radius

The geometrical characteristics of the contact patch is also changed with higher load. As seen in Figure 13 the average radius of the elliptical contact patch is significantly higher when the load is increased. The ratio between the two semi-axis does not differ, but the magnitude does and thus the average radius is different. This could be due to the change in contact point position that with higher load the wheel and rail makes contact at position with higher grade of conformity. It may also be the case resulting simply by the higher load creating more downward facing force deforming wheel and rail slightly more in the contact point.

The average radius of the contact patch, c, is calculated in the following manner in GENSYS [9], where a and b are the two semi-axis in respectively longitudinal and lateral direction. A simple way of calculating the contact patch area is also presented here:

$$\begin{aligned} c &= \sqrt{a \cdot b} \\ A_e &= \pi \cdot c^2 \end{aligned} \tag{3}$$

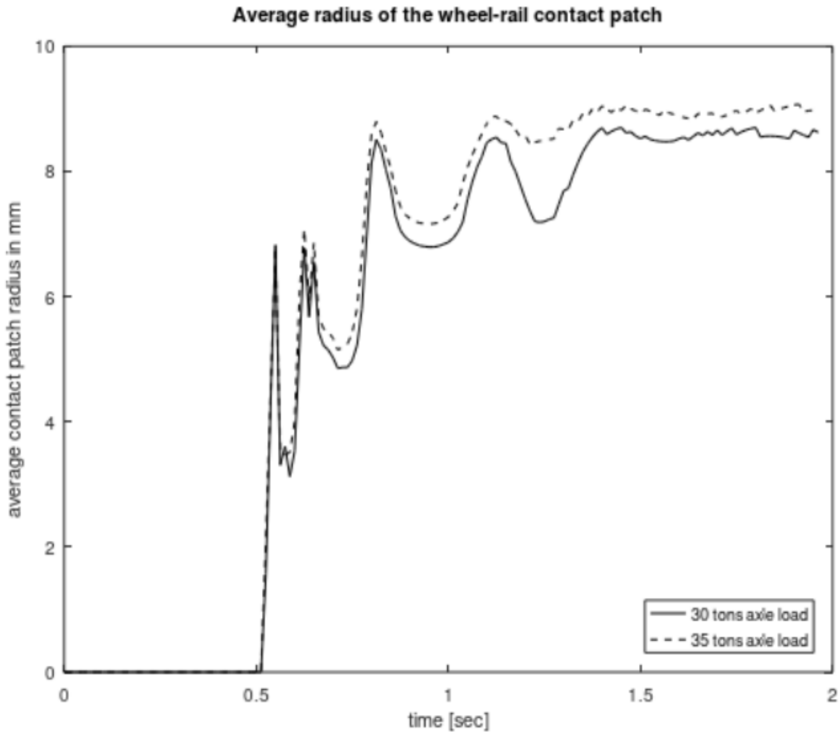



Figure 13: Difference between the contact point average radius between 30 tonnes and 35 tonnes in the diverging route and facing move

In Figure 11 and Figure 12 the graphs begin at different times. This is due to the models capacity of having multiple contact points. In Figure 11 it shows when the wheel first makes contact with the switch blade and wear is observed, which is earlier than to where Figure 12 and Figure 13 shows contact. In the latter two this is due to graphs displaying the "main" contact point between wheel and switch blade and does not include the first point of contact, as these are separated in the simulation model and the in the post processing results.

In the through route in the switch panel the wear rate is higher when load is increased which can be seen in Figure 14. This difference indicates that the wheel-rail dynamics have changed and it requires further investigation to the effects of the higher load, as well as looking at ways to reduce wear mechanisms in this part of the S&C.

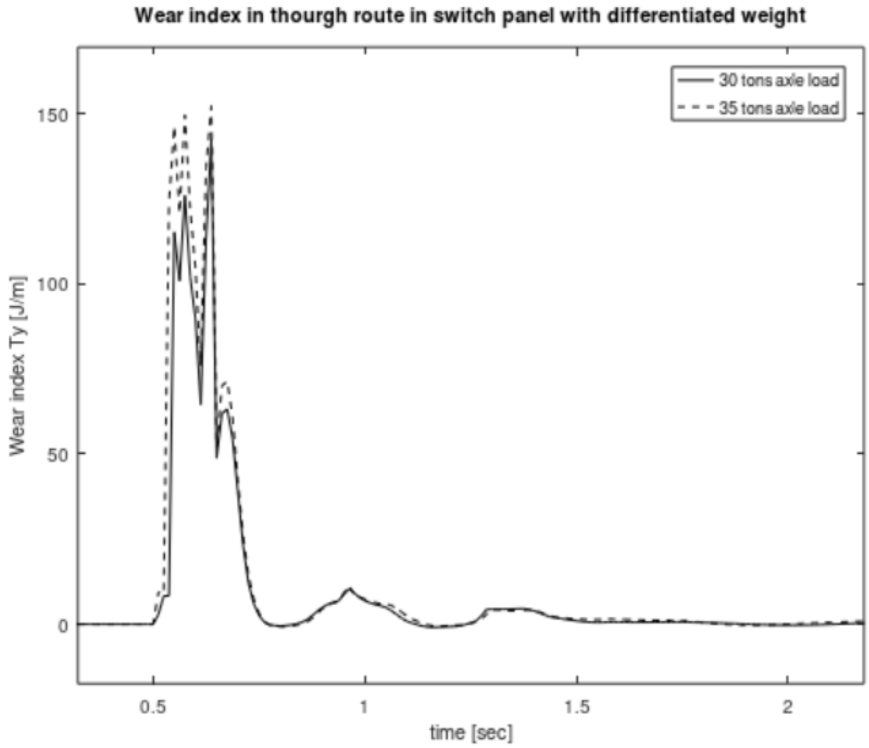



Figure 14: Difference of wear rate comparing 30 tonnes axle load and 35 tonnes axle load in the through route facing move in the switch panel

The lateral movement in the through route in the switch panel lasts for longer in the scenario with the increased weight, but for both weights the hunting motion is stabilised and the sinusoidal movement ceases before the wheelset enters the crossing panel.

In the crossing panel there was a change in wear rate in the diverging route with an increase of almost 50 % in the peak wear index with increased load, shown in Figure 15. The differences in wear for the through route are not as significant but with the intricate wheel-rail interactions in the crossing panel an investigation here was also performed.

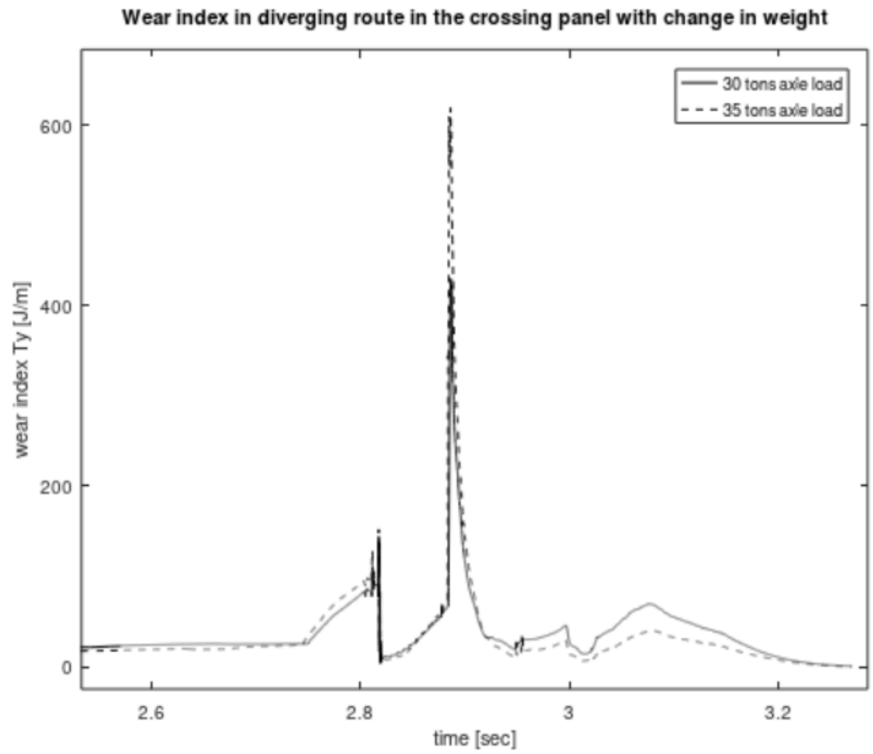



Figure 15: Difference of wear rate comparing 30 tonnes axle load and 35 tonnes axle load in the diverging route facing move in the crossing panel

All these results tell us that a change in axle load has significant impact on the damage and performance in S&C. Therefore a study was carried out to investigate the effect of cant and track gauge alteration, and the results from this follows here.

## 4.2 Cant Alterations in Diverging Route

Cant was introduced as a measure to compensate for the lateral acceleration and lateral forces experienced when the train is in the curved track in the diverging route. On open track curves will be fitted with a cant or superelevation to reduce the lateral forces, but on straight track this is not necessary. Therefore cant was only introduced in the diverging route when simulating switches and crossings in this study. When looking at results from the introduction of cant there was done a comparison between the loaded with 35 axle load with each other and the unloaded scenario with 30 tonnes (situation today). As already shown there are big changes in the contact points position, wear, dynamic forces and geometrical properties of the contact patch itself when axle load is increased. Here, the findings from introducing 37 mm and 75 mm cant in the diverging route is presented.

### 4.2.1 Switch Panel

Figure 16 and Figure 17 shows the contact patch average radius for two contact points between the wheel and the switch blade for three different scenarios of cant in the diverging route and facing move.

Firstly there is observed that a change of the time it takes for the wheel to make contact with the switch blade. Compared to no cant the time to contact is postponed by 0.01 seconds with 37 mm cant and by 0.05 seconds with 75 mm cant. Time difference from simulations with 75 mm cant indicate that with the speed of 60 km/h the hitting point will happen about 1 metre later. This means that when the wheel meets the switch blade, the blade has more material and is more resistant to wear, which again might effect the life time of the switch blade. The effect is not as obvious with 37 mm cant, but also here some postponing will lead to a beneficial situation. Stochastic variables are not taken into account when simulating and there are uncertainty in the results regarding the accuracy of the postponement but indications of a later hitting point are clear.

From Figure 17 it can be seen that there is a change in the average radius of the ellipse shaped contact patch between wheel and rail, starting after 0.65 seconds into the simulation. It is at this point when all load is transferred through only one contact point onto the switch blade, which is almost fully transformed into a rail with standard rail head geometry, which happens at 0.7 seconds.

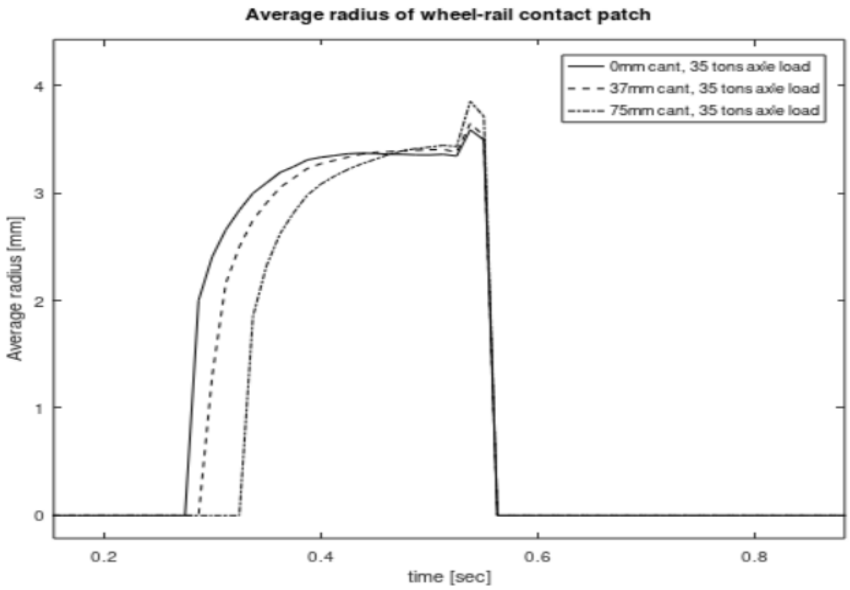



Figure 16: Average radius of the first contact point in the switch panel diverging route with different cant

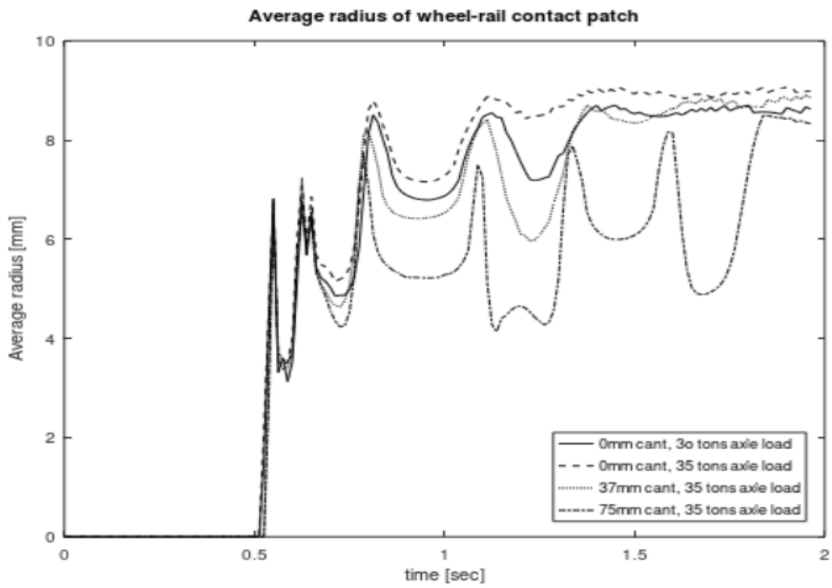



Figure 17: Average radius of the contact point with changing cant in the switch panel in the diverging route

The lateral movement and constant change of average radius in the case of 75 mm cant also effect the stresses on the switch blade through the diverging route in the switch panel. Cant has been shown to shift the contact forces from the switch blade to the stock rail in the diverging route. However, with the lateral movement in the scenario with 75 mm cant contact area is constantly changing, and when the area is small stresses between the switch blade and the wheel increases. From Figure 17 it is obvious that when average radius is low between wheel and switch blade the stress is high. Comparing the stresses to the scenario with 0 mm cant there are big differences, especially in the closure panel where the area of the contact patch is almost constant in the standard cant scenario. The wagon is pushed down by gravitational forces and the wheelset struggles to find the optimum rolling radius. This oscillating effect causes more stresses on the switch blade.

Because the wheelset will try to find its optimum rolling radius in the curve of the diverging route closure panel there will be a change of the location of the contact patch, which is shown in Figure 18 where the wheelset in the scenario with 75 mm cant moves more laterally than the wheelset with no or 37 mm cant. This is probably due to the train being pushed down towards the inner rail (stock rail) because of the weight and the wheelset tries to find the right rolling radius through the curve.

One cannot say for certain that the consequences of the greater movement with more cant is beneficial or not, but it indicates a less smooth travel through the closure panel and more oscillating movement resulting in less predictable entrance into the crossing panel.

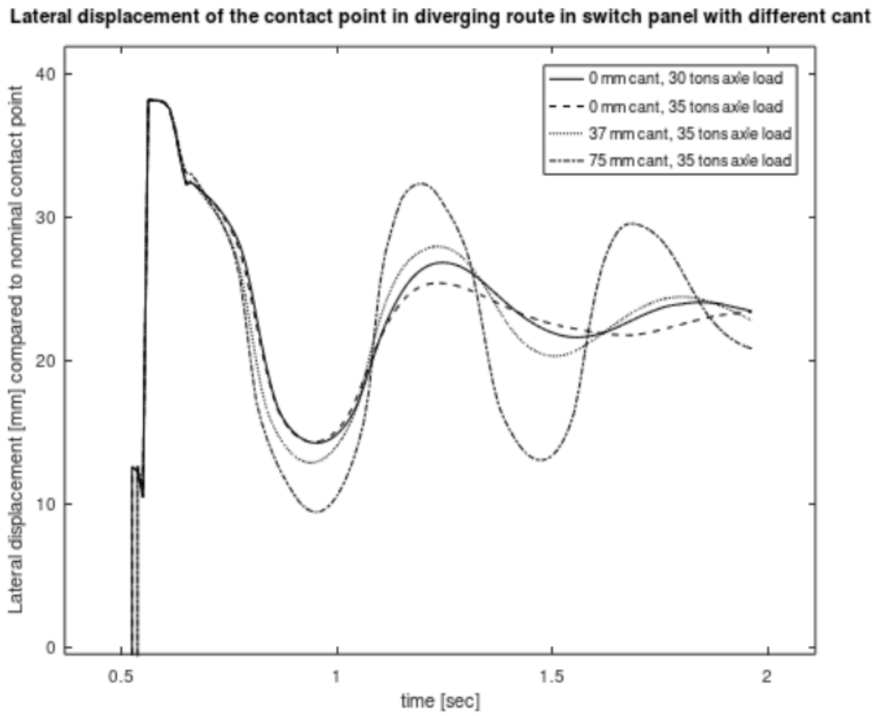



Figure 18: Position of the contact point between wheel and rail comparison with different cants and weight

From Figure 19 it is obvious that the normal contact force on the switch blade was lower when there was cant in the diverging route. As expected, the normal contact force was lower with more cant as a greater portion of the load is transferred to the stock rail as is normal in open track curves, instead of an overload on the switch blade following the cant deficiency. However, when load is transferred to the stock rail we might expect to see more rail damages and wear on this as creep and creep forces is expected to increase. Initially the contact forces on the switch rail in the diverging route was similar for all cant scenarios, but when the wheelset enters the curve and all loads from the wheel is transferred only through the switch blade and not also the straight stock rail the difference in normal contact force was evident.

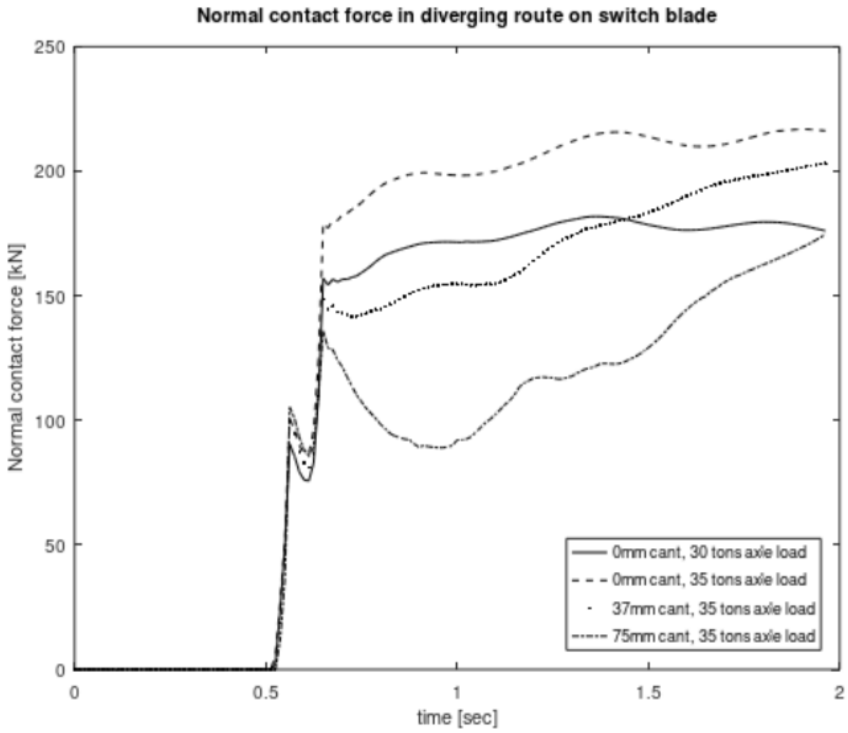



Figure 19: Normal contact force between wheel and switch blade in the switch panel diverging route with altering cant and weight

The greatest magnitude of wear index in the diverging route in the facing move is observed in the scenario with most cant (75 mm). The peak point is 6% higher than in the scenario with 0 cant and the same axle load, but the duration of the first wear rate peak is also 9% shorter, because of the delay of contact points on the switch blade. However, from the lateral movement in the case when 75 mm cant is introduced, an additional wear rate is observed in the closure panel which does not exist for the other scenarios. These wear rate peaks are not as high and will most likely not effect the life time of the rail, also because this is not observed from the second wheelset of the first bogie during simulations. Wear index comparisons in the diverging route in the switch panel is shown in Figure 20 below. With the introduction of cant there were observed differences in contact forces on the curved stock rail, which now carries more of the load. This was as expected.

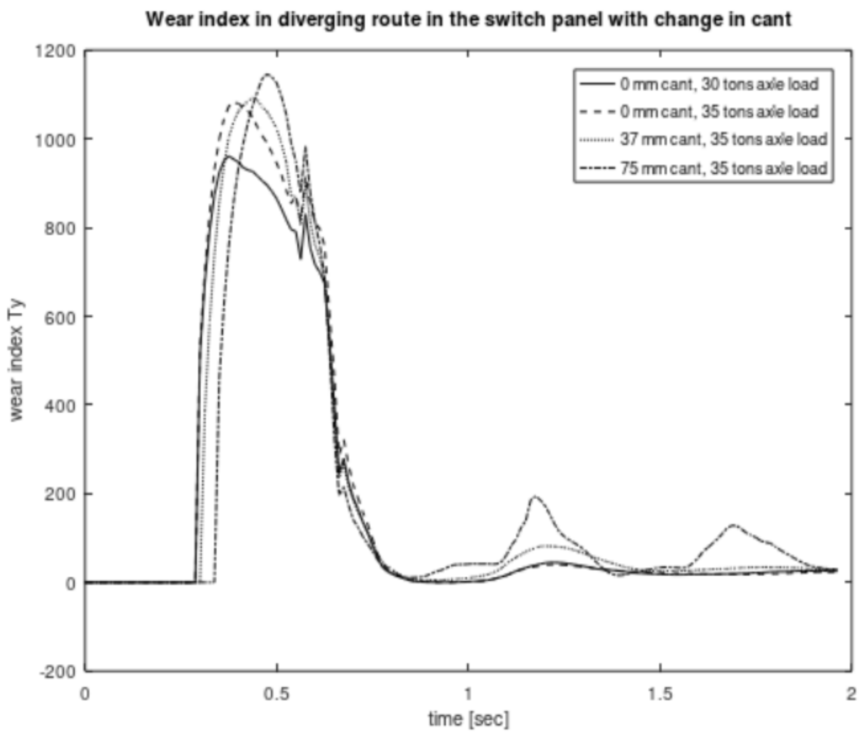



Figure 20: Comparison of wear index in the switch panel in the diverging route, comparing standard 0 mm cant to 37 and 75 mm cant

### 4.2.2 Crossing Panel

In the crossing panel the difference in wear index over a short period of time was big when comparing the scenario with 75 mm cant to the other scenarios. The simulation with unloaded (30 tonnes) gives the least amount of wear rate, and the loaded with 0 mm and 37 mm cant are similar. Compared to the other simulations with similar weight there is in the 75 mm scenario observed an increase of 36 percent increase in the wear peak.

The increased wear in the scenario with 75 mm cant is probably a coincidence and is due to unfavourable conditions and high creepage for this exact scenario. The magnitude of lateral movement of the wheelset is higher when 75mm cant is introduced. Coincidently, the wheel in contact with the crossing nose when moving from wing rail has very high relative velocity, thus generates large amount of creepage. Making the situation worse, the wheel in contact with the crossing nose is very close to the wheel throat. An attempt of 150mm cant is introduced as a check, and in this case, it is shown that the creepage is low. This indicates that if one wishes to introduce cant in S&C careful planning might be needed as the position of the wheel would have consequences towards wear on both crossing rails and wheel.

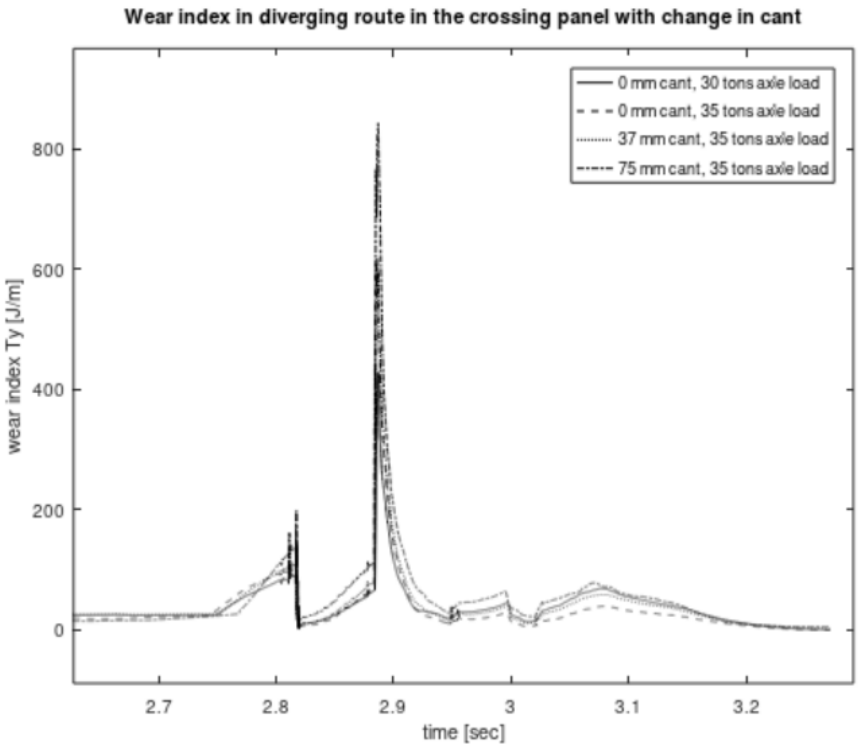



Figure 21: Comparison of wear index in the crossing panel in the diverging route, comparing standard 0 mm cant to 37 and 75 mm cant

## 4.3 Track Gauge Alterations

Different track gauges were introduced to see if performance could be improved and if wear could be mitigated. From the literature it has been shown that a widening of gauge in the trough route in the switch panel might be beneficial when it comes to guiding the wheelset past the adjacent curved switch blade. To study the effects of different track gauges simulations were run in both diverging and straight routes in both switch and crossing panel. There were nine different gauge scenarios with an interval of 2 mm between each gauge, with the narrowest being 8 mm narrower than standard and the widest 8 mm wider than the standard gauge of 1435 mm. The gauge alterations were simulated in each direction and for both weights (loaded and unloaded), which gave a total of 144 simulations. Only results from the facing move simulations were taken into account when evaluating results, which in total gave 72 simulations to evaluate.

### 4.3.1 Switch Panel

The results show significant differences in performance parameters when track gauge was altered. Looking at the diverging route in the facing direction in the switch panel there was found that the contact point between wheel and switch blade happened at a position further into the S&C. Here, a general widening makes the contact happen later because of the contact points in a wider track make room for the switch blade to "build itself up" before it hits the wheel, rather than hitting the wheel flange. When there is a narrowing the switch blade will make contact with the wheel at an earlier stage at the flange, where the blade is thinner and the flange contact induces more creepage and unwanted events.

Narrowing of the gauge is found to be not favourable in the switch panel, even though there is little lateral movement in the closure panel in the diverging route. This is due to flange contact on both wheels, meaning that in the case of 8 mm narrower track there is no room for the wheelset to move laterally. Flange contact is not beneficial in any parts of the track, and is also why the guidelines for maintenance suggests that when track is narrower than 1432 mm, from Table 1, maintenance measures are needed. Comparisons between lateral movement of the contact point between switch blade and wheel is shown in Figure 22 where the scenario with 8 mm narrowing seems to be the most stable.

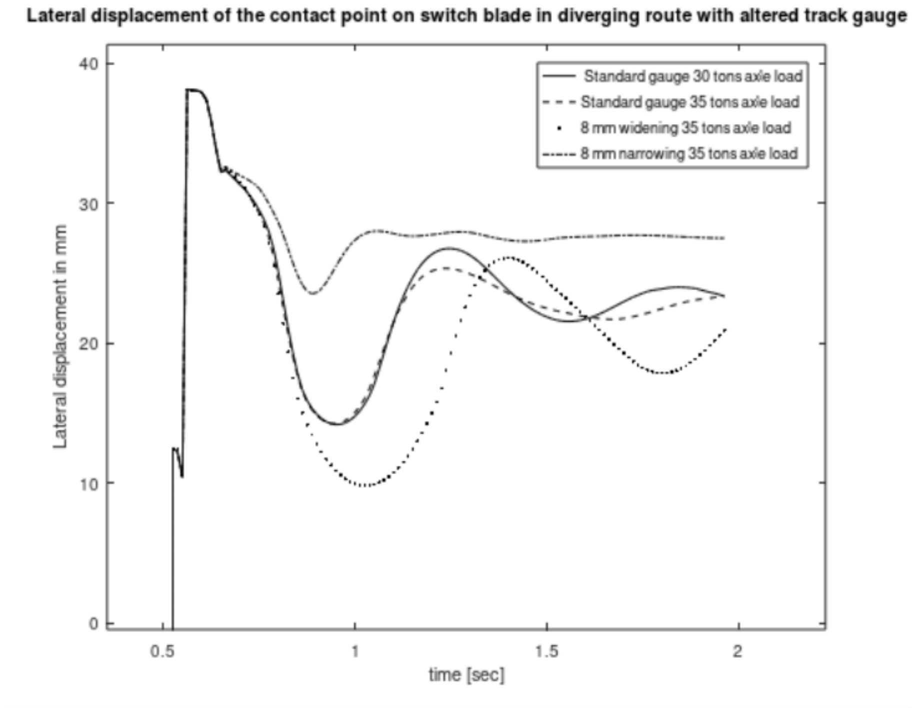



Figure 22: Position of the contact point compared to nominal contact point with four different track gauge scenarios. Positive offset from 0 is lateral displacement towards the tread edge compared to nominal contact point

It is almost never beneficial with a narrower track gauge when cornering, as the wheelset often prefers a wider gauge steering through curves and finding the optimal rolling radius difference. From Figure 22 it could at first glance be a nice consequence of narrowing the gauge in the switch panel that there is very little lateral movement of the wheelset through the curve in the closure panel. This is not favourable here, as the flanges comes into contact and locks the wheel into place, creating more creepage and wear.

When evaluating the wear index with altered track gauges there are significant differences to be found in the switch panel diverging route. It changes gradually as expected with all different track gauge scenarios, with shorter duration of wear rate with wider gauge, but also the peak wear rate increased with wider gauge. The narrower the gauge became compared to standard, the longer wear was present and the peak was approximately on the same level as the standard gauge. Duration of contact plays a big role in wear, and the longer wear index is observed the longer there is wear present. As the area under the graphs represent the energy dissipated a longer duration and similar high peak indicates that with narrower gauge more wear will be experienced. In Figure 23 the wear rate is presented for four different scenarios displaying the differences in terms of wear rate on the switch blade in the diverging route dependent on track gauge.

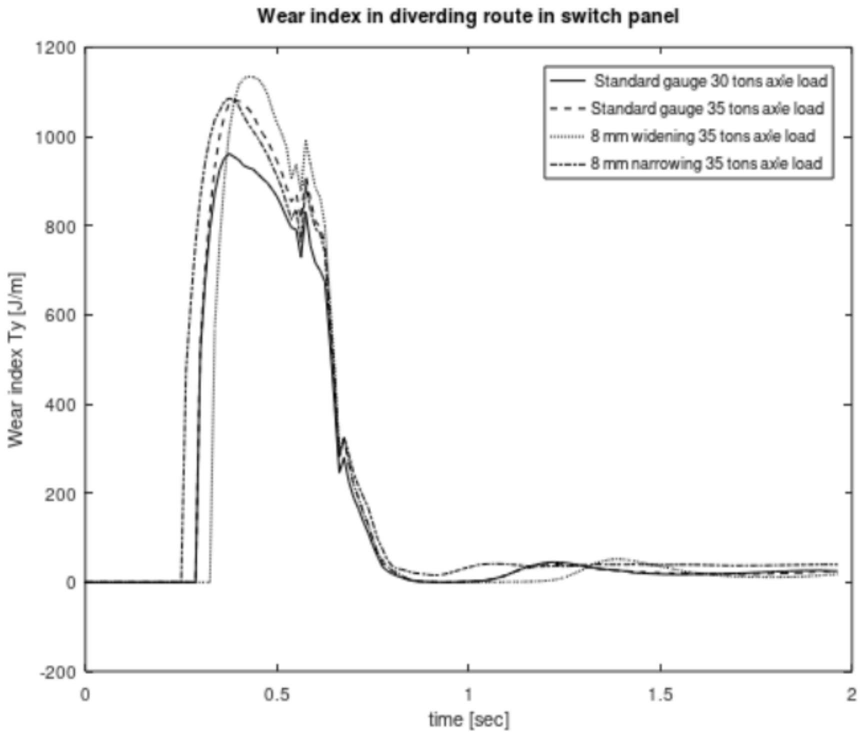
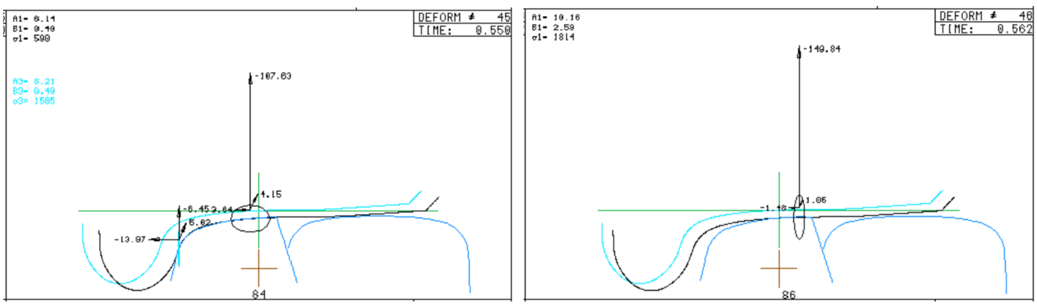



Figure 23: Wear index in the switch panel in the diverging route, comparing weights in standard gauge to 8 mm wider and 8 mm narrower with high load

In the switch panel through route there are many significant differences when track gauge is widened by 8 mm. The biggest difference from the standard track gauge is that there is only one contact point between the straight switch blade and the wheel. This leads to fewer issues and less complicated steering mechanisms, discussed in 2.3.1. This is achieved by simply moving the contact points to the stock rail further out on the tread and giving room for the switch blade to enter underneath the wheel. A similar effect to the FAKOP and CAFTERSAN design is then accomplished with less complicated contact conditions in the through route as well as delaying the time to for the contact point to occur. As already mentioned it also reduces the number of intricate contact points, as shown in Figure 24.



(a) Standard gauge (b) 8 mm widening of gauge

Figure 24: Multiple (three) contact points with the switch blade with standard gauge (a) and clean transition to switch blade in the through route with only one contact point with 8 mm widening of the gauge (b)

The benefit of fewer contact points is seen in all scenarios with gauge widening, from 2 mm to 8 mm, but the closer to 0 mm widening the more lateral movement is observed immediately after the point of contact. This is due to the rolling radius differences and the lateral offset, as the wheelset tries to find the optimum rolling radius for each wheel. The critical speed with the load combined with suspension and wheel shape is most probably higher than 60 km/h which all simulations are run with. This makes the hunting motion decrease to zero after only a few seconds in all scenarios even though the suspension is relatively soft.

When the track is narrower than standard, the contact point happens initially on the wheel flange, where the effective rolling radius is very big, in addition to this the lateral forces pushing the contact point away from the flange creates bigger movement in the through route in the switch panel. It is clear that the interaction is worsened by narrower track gauge, and rightfully the standards for maintenance is set at a very small margin of error when it comes to narrowing in the switch panel. The simulation results corresponds with the limits of the standards.

In Figure 25 the difference in wear rate in the through route of the switch panel is presented. From the simulations results the wear rate in scenario with 8 mm widening is negligible, where it is less than 10% of the wear rate in the standard gauge. The wear rate increases in duration and magnitude when the track gauge is narrower. In the figure wear rate on the straight switch blade with standard gauge, 2 mm narrower gauge and

8 mm wider gauge is displayed, all scenarios with 35 tonnes axle load. Wear rate was not comparable between the extreme scenarios of 8 mm narrowing and widening, so 2 mm narrower track is used to show the difference track gauge makes to wear rate in the through route of the switch panel. When narrowing the gauge by 8 mm the contact position on the wheel flange accompanied with the higher induced lateral force made the wear index magnitude more than 5 times greater compared to standard gauge. For visual representation the narrowest scenario was therefore not included as it would be hard to see nuances between the other scenarios.

The low values of wear rate in the wider gauge scenario comes from the clean contact point with less creepage compared to multiple contact points in standard and narrower track gauge scenarios. Benefits of widening the track is observed already from a 2 mm widening, but the negligible wear rate is only observed in the scenarios with 6 mm widening and 8 mm widening. Likewise, the disadvantages of narrower track gauge is observed already when gauge is narrowed by 2 mm, where wear rate almost doubles in magnitude compared to standard gauge. The duration of the wear index is also prolonged in scenarios with narrower track gauge, so the effects is worsened by greater absolute magnitude and longer duration.

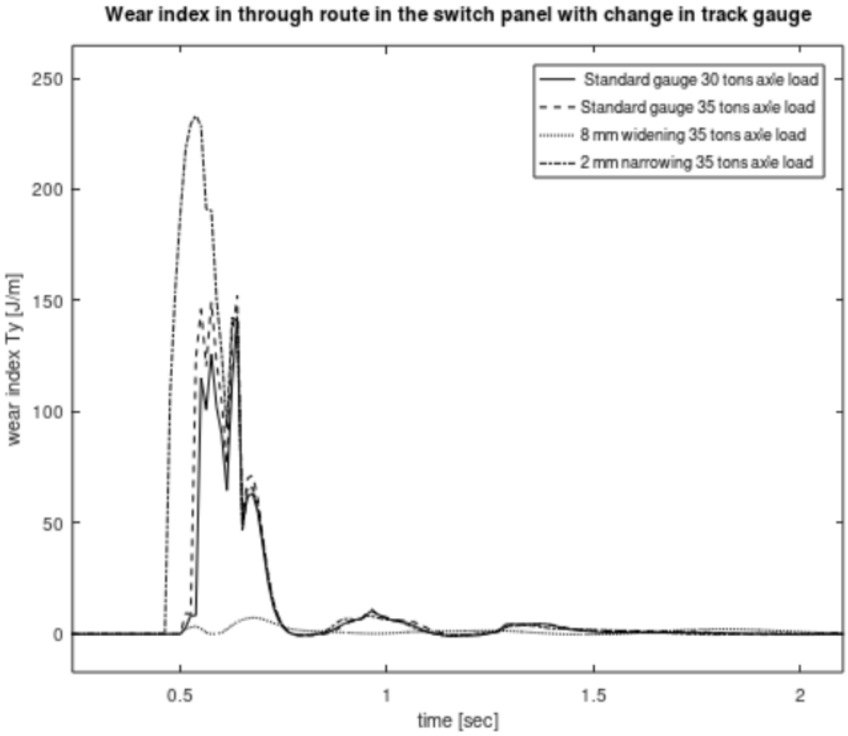
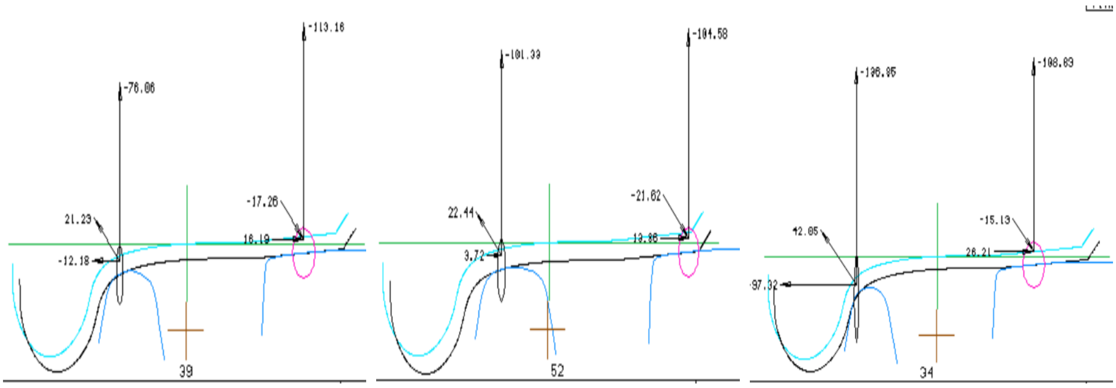



Figure 25: Wear index in the switch panel in the through route, comparing weights in standard gauge to 8 mm wider and 2 mm narrower with high load

### 4.3.2 Crossing Panel

In the diverging route in the crossing panel it was found that narrower gauge is not favourable. The wear index in simulations from 2 mm narrower to 8 mm narrower gauge is between 2 to 5 times higher in absolute magnitude over a short time period. This indicate that non favourable contact point conditions are present at the time of nose contact which also shows from the results. The crossing nose meets the wheel close to the wheel flange, as can be seen from Figure 26, and the difference in rolling radius of the contact points disturb the steering of the wagon more in the scenario with narrower gauge than normal. This induces more creepage and creep forces which in turn increases the wear index on the crossing nose and degrade it faster compared to a crossing panel with normal or wider gauge. Including the results from narrower gauge scenarios in a figure does not work as the magnitude of the peak is that many times higher than for normal and wider scenarios, making it very difficult for humans to distinguish differences. The negative effects from flange contact and significant rolling radius differences on the wheel in the crossing panel with narrower gauge is the same in both the diverging and through route and are similar to the other scenarios with narrower gauge.



(a) Standard gauge (b) 8 mm wider gauge (c) 8 mm narrower of gauge

Figure 26: Difference in contact point position with different track gauge in the diverging route in the crossing panel. Flange contact is severe in the case of the narrowest gauge

In Figure 27 wear index with track widening of 8 mm is compared to standard track gauge with both 30 and 35 tonnes axle load in the diverging route in the crossing panel. It shows that wider track gauge reduces the wear index compared to the two scenarios with standard gauge.

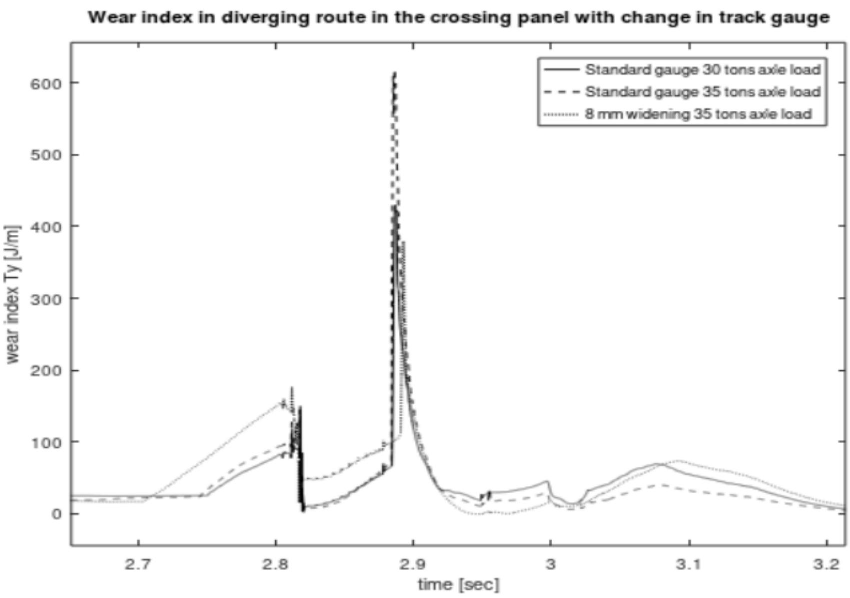



Figure 27: Wear index in the crossing panel in diverging route, comparing different gauges and loads scenarios

In the through route in the crossing panel differences in wear rates are observed. Where a widening of 2 mm makes the wear index drop down to wear values similar to scenario with 30 tonnes axle load and standard gauge, and where the scenario with 35 tonnes and standard gauge has slightly higher (10%) compared to 30 tonnes and standard gauge. With a widening of 8 mm the reduction in peak wear is bigger, about two thirds of the standard gauge. With a narrower gauge however, the increase in peak wear index is so high that it may cause issues at the crossing nose when the flange makes contact. Once again, the results show that the strict requirements limits for narrow track gauge in switches and crossings are good.

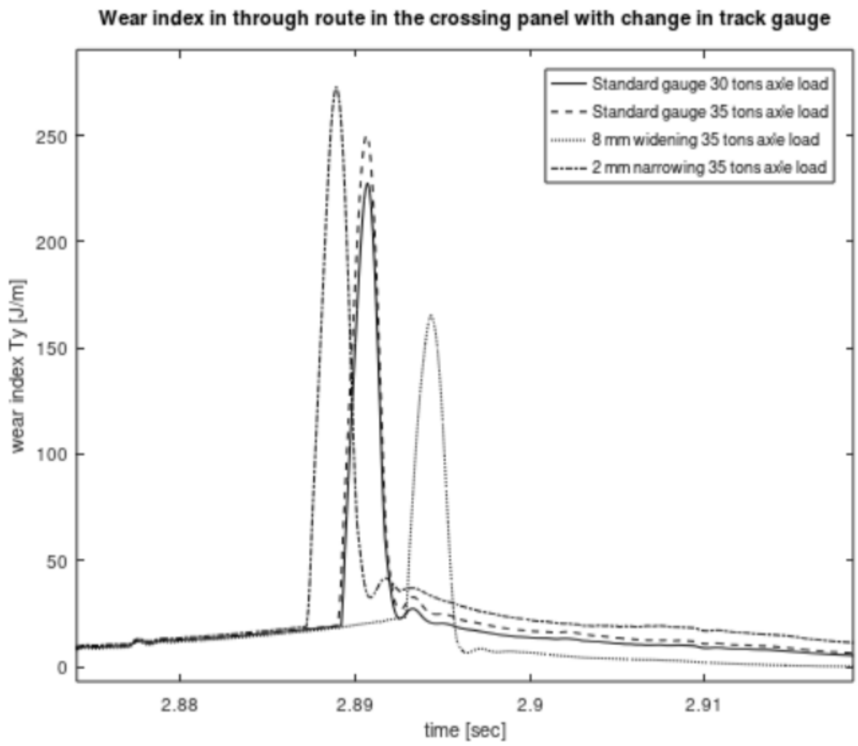



Figure 28: Wear index in the crossing panel in through route, comparing standard gauge to minor narrowing and major widening of gauge

So far, only the crossing nose and the wing rails have been discussed, but from the results it can be observed that the change of gauge also has an impact on the wheel flange contact with the check/guard rails. With standard gauge and unloaded wagon there is in the guard rail to flange contact observed a lateral contact force of 50 kN, and the increase of weight did not affect this number significantly. However, when the gauge was altered the contact forces in the flange way changed. For the wider gauge of 8 mm lateral contact forces increased by 34% induced by more lateral movement forcing the wheel to hit the guard rail with higher impact than in other scenarios. For the narrower gauge it is shown that the lateral force on the wheel flange is reduced by about 30%. The results of lateral contact force is shown in Figure 29.

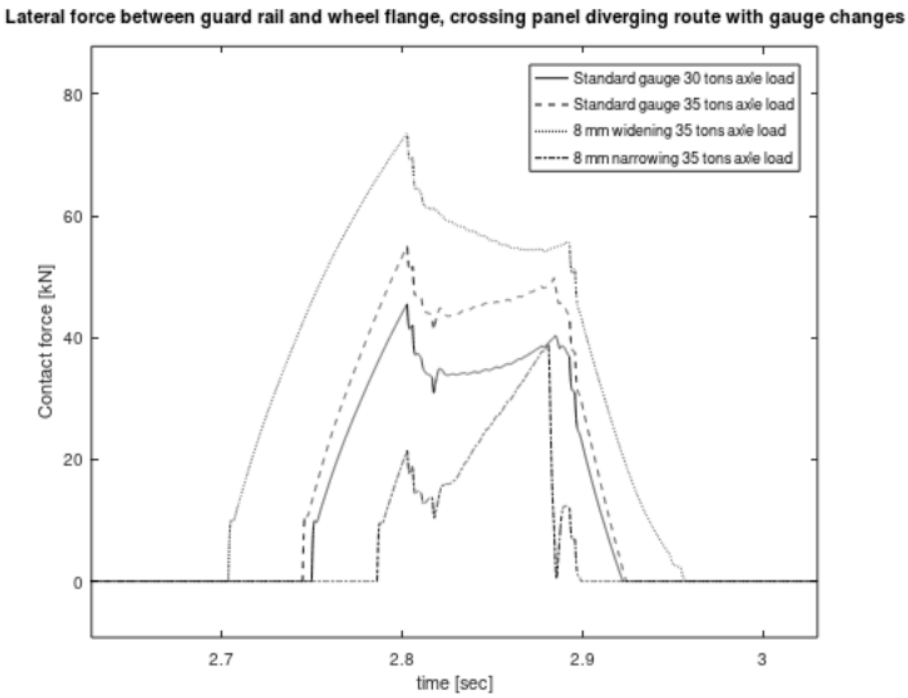



Figure 29: Lateral forces between guard rail and wheel flange in diverging route in the crossing panel

The change of flange contact with guard rails suggests that for this particular case, with the set speed of 60 km/h, wheel and rail type it is easier for the wheelset to find the optimum rolling radius in narrower gauge in the crossing panel diverging route. This corresponds to a higher equivalent conicity and makes sense when looking at the wheel geometry as the wheel conicity increases closer to the flange, and less lateral movement is needed to find optimum rolling radii.

Looking at the position of the contact point between wheel and rail in the crossing panel diverging route an interesting aspect showed up, called contact jump. It was observed when the gauge was widened by 8 mm but is also happening for all scenarios with gauge widening. The jump, shown in Figure 30 comes from lateral movement of the wheel to find the optimum rolling radius, and when the contact point is close to the middle of the tread more movement is needed to alter the effective rolling radius, hence a jump in position occurred.

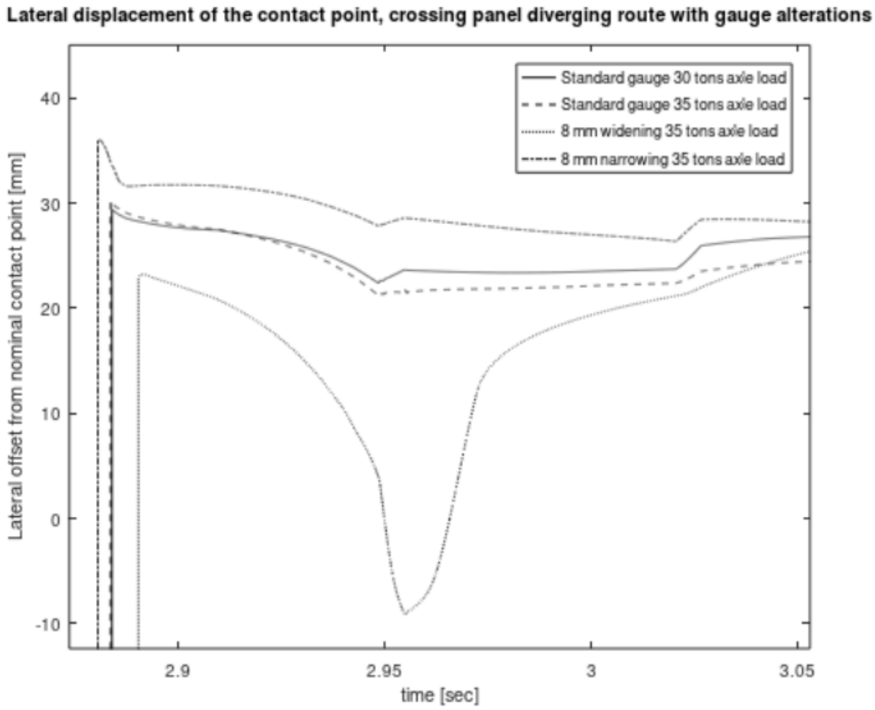



Figure 30: Contact position of the contact point in the crossing panel diverging route, with a contact jump with 8 mm widening.

# 5 Discussion

### Increased Load

A valid option of increasing the capacity of a freight line is to increase the weight of the transported goods. However, one must be aware of the following consequences. Through the simulations conducted, it was observed that an increase in axle load also increased the wear of S&C. A change in contact point position along the track was also observed. These consequences might induce a larger need for maintenance costs, which is already high in this specific part of the track.

### Introduction of Cant

The introduction of cant in the diverging route was tested as an option to reducing the lateral forces. This especially accounts for the switch blade, as this is more prone to damage due to the rail head geometry. In the switch panel both positive and negative effects were observed in scenarios with a large cant (75 mm), where the magnitude of the lateral movement was higher than in other cases due to gravitational forces pulling the wagon down towards the stock rail. It was, however, observed that a more shallow cant (37 mm) resulted in some similarities of the benefits of the high cant, without as many drawbacks. This could indicate that small changes in superelevation in parts of the S&C might be beneficial, and there is a possibility to include cant in the future when higher axle loads are expected. One way to introduce cant in S&C is by lowering the diverging stock rail, leaving the moving parts in a planar level.

### Track Gauge Alterations

In the results it was observed that there are benefits of widening of the track gauge when increasing load in S&C. Changing track gauge is not a newly introduced concept, as some designs are utilised on high speed lines today. With the performance benefits gained from wider gauge, especially in the switch panel, it could in the future be of interest to infrastructure managers to include a track gauge widening of heavy haul lines. By increasing the axle load, the wear also increases. To reduce the costs of maintenance, a wider track gauge will most likely lead to lower costs. In this thesis a simple general widening of the track was introduced and simulated, but designs that are in place in high speed lines could be adopted in heavy haul lines in the future. FAKOP and CAFTERSAN designs that have been presented in this thesis could be options to achieve wider track gauge. Drawbacks in the CAFTERSAN design relates to the milling of the rail head reducing the mass of the rail and might deteriorate the quality of the rail. Thus, the FAKOP design might be more preferable for heavy haul lines. Before any of these designs are introduced, a closer investigation is needed, and this thesis does only indicate that a widening of track gauge could be beneficial for performance with increased load.

**Vehicle Model**

There are some uncertainties concerning the vehicle model as updated vehicle information on the wagons used on the Ofoten Line was not available to the author. Therefore, suspension, wheel characteristics and other vehicle parameters were not altered in the model, which would have given the results better accuracy in terms of comparing to the Ofoten Line case with the objective of increasing axle load to 35 tonnes. However, the study gives good indications on what can be expected of wheel-rail interaction and wear estimations.

**Track Model Separation**

Due to the discontinuity between the two track models, there was uncertainty associated with the lateral position of the wheels coming into the crossing panel. This followed from the separation of switch panel and crossing panel that was done when modelling. However, in both diverging and through routes a comparison was made to ensure compatibility between lateral position of the stop and start so the validity of the results are better and more trustworthy. Despite this, a good way to ensure better results would be to combine track models so the train could run through the whole S&C in one go.

**Optimisation**

In the project thesis [10], which was done previous to this thesis as a pre-project, it was stated that a genetic algorithm would be used to solve the optimisation problems to way find an optimal design and layout for the S&C. Due to time constraints and the complexity of making the genetic algorithm communicate with the MBS code, it was decided that a manual investigation approach would be more rewarding and efficient in the time span of this thesis.

# 6 Conclusions

In this thesis GENSYS has been used to simulate a freight wagon similar to the Fanoo040 through 60EI-R760-1:15 S&C. An investigation of heavier axle load was performed and a study of effects on wheel-rail interaction and wear estimation when altering two track parameters was conducted.

When increasing the axle load to 35 tonnes, changes were observed that influenced performance and wheel-rail interaction for freight wagons with three-piece bogie in a railway S&C. More wear and a change in contact point position was observed, and some unfavourable events with multiple contact points was found.

From the results of the simulations, it was found that a way to improve performance for trains with 35 tonnes axle load is to widen the track gauge in the switch panel of the 60EI-R760-1:15 S&C. This will lead to less wear and more clean transitions, especially in the through route close to the switch panel. In practice, this could be done by implementing designs similar to FAKOP or CAFTERSAN systems. Narrower track gauge was found to be less favourable in most cases, and is not recommended as a measure to mitigate damage in the S&C.

The introduction of 37 mm and 75 mm of cant was shown to have performance enhancing abilities in the switch panel, by delaying the contact between wheel and switch blade. This can be included by lowering the diverging stock rail. By introducing cant, more load is distributed to the stock rail, resulting in less wear on the switch blade and crossing nose. Comparatively, simulations with 37 mm cant performed better than with 75 mm cant because of less lateral movements and bigger contact patches. In addition, simulations with 75 mm cant showed unfavourable contact jump on the crossing nose which results in a higher wear index. However, a careful investigation of the effects for the specific S&C is recommended to be certain about the effects of the cant.

# 7 Future Work

As switches and crossings are complex, they would always need further investigations and improvements, both in component design, track stiffness evaluations and many more aspects.

To continue the research from this thesis it is recommended to look into the following aspects:

- Trailing moves - As the simulations for the trailing moves were excluded from the results due to too much uncertainties, this should be included in future studies as trains run in both facing and trailing moves through S&C.

- Suspension systems - An investigation into what different suspension systems could do to affect performance through S&C is a good continuation of the work performed in this thesis. Suspension systems are important for performance through curves and on straight tracks, and should be included when investigating heavier axle loads of wagons through S&C in future studies.

  The primary and secondary suspension are different for wagons with different masses and for intended use. A softer suspension will help the wheelset run more smoothly through curves as well as in diverging route in S&C. Softer suspension systems also makes the critical speed lower, which decide how fast a train can run without experiencing uncontrollable hunting motions in case of a lateral disturbance. Stiffer suspension will not be favourable in curves and through diverging route in S&C, but will provide more stability to the wagon on straight track.

  A wagon with very stiff suspension might have issues when cornering, which again could lead to derailment. This especially accounts for long, empty tankers. When the vertical force is low and the lateral force is high the Y/Q limit will be exceeded and the risk of derailment increases. With radial bogies, this problem is not as frequent, as the angle of attack on the second wheelset is more favourable compared to standard bogies, and the lateral forces are lower. This can be seen in Figure 31

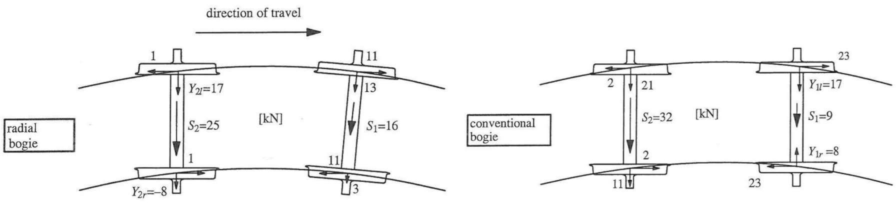



Figure 31: Radial and standard bogie through curve with different angle of attacks and lateral forces

- Genetic algorithm for optimisation - Creating a genetic algorithm that communicates with GENSYS and feeds in the next generation's rail, wheel or track parameters to find a global optimum design would also be a good continuation of this thesis work. As there has not been performed optimisations on S&C with as high axle load as 35 tonnes, this should be done to find optimum solutions.

# References


[1] A. H. Løhren and K. G. Green, "Feil i sporveksler 2015," *Jernbaneverket, Norway*, 2015.

[2] Bane Nor, "Finner feil før trafikken stopper," Bane Nor, Oslo, Norway, Des 2016. [Online]. Available: https://www.banenor.no/Om-oss/arkiv-jernbaneverket/Jernbanemagasinet-arkiv/Nyheter/desember-2016/de-oppdager-feil-for-trafikken-stopper/

[3] A. H. Løhren and A. K. Buan, "Uttrekk Vedlikeholdskostnader KV overbygning 2015," *Jernbaneverket, Norway*, 2015.

[4] A. H. Løhren and I. Olsen, "Uttrekk Vedlikeholdskostnader KV overbygning 2019," *Bane Nor, Norway*, 2021.

[5] Samferdselsdepartementet (Ministry of Transport), "Jernbanereformen," Regjeringen, Oslo, Norway, January 2020. [Online]. Available: https://www.regjeringen.no/no/tema/transport-og-kommunikasjon/jernbane_og_jernbanetransport/reform-av-jernbanereformen/id2344653/

[6] The EU, "Fourth railway package of 2016," The European Union, Bruxelles, Belgium, 2021, last accessed 25.05.2021. [Online]. Available: https://ec.europa.eu/transport/modes/rail/packages/2013_en

[7] The EU, "2021 is the European Year of Rail," The European Union, Bruxelles, Belgium, 2021, last accessed 25.05.2021. [Online]. Available: https://europa.eu/year-of-rail/index_en

[8] Jernbanedirektoratet, "NTP 2022-2033: Nesten 400 milliarder til jernbane," Jernbanedirektoratet, Oslo, March 2021. [Online]. Available: https://www.jernbanedirektoratet.no/no/aktualiteter/2021/ntp-2022-2033-nesten-400-milliarder-til-jernbane/

[9] AB DEsolver, "GENSYS," AB DEsolver, Östersund, Sweden, 2021. [Online]. Available: gensys.se

[10] Ø. Brittmark, "Heavier Axle Load in Switches and Crossings - Project Assignment," NTNU, Trondheim, 2020, Project assignment.

[11] H. Ossberger and H. Bishop, "Life cycle cost-driven innovations for heavy haul turnouts," *Journal of Rail and Rapid Transit*, vol. 224, no. 5, pp. 405–412, June 2010.

[12] B. Pålsson, "Optimisation of Railway Switches and Crossings," Chalmers University of Technology, Göteborg, 2014, Doctoral thesis.

[13] C. Wan, V. Markine, and I. Shevtsov, "Improvement of vehicle–turnout interaction by optimising the shape of crossing nose," *Vehicle System Dynamics*, vol. 51, no. 10, pp. 1619–1639, 2013.

[14] B. A. Pålsson, "Design optimisation of switch rails in railway turnouts," *Vehicle System Dynamics*, vol. 51, no. 10, pp. 1619–1639, May 2013.

[15] I. Grossoni, Y. Bezin, and S. Neves, "Optimisation of support stiffness at railway crossings," *Vehicle System Dynamics*, vol. 55, no. 7, pp. 1072–1096, 2018.

[16] C. Wan, V. Markine, and I. Shevtsov, "Optimisation of the elastic track properties of turnout crossings," *Journal of Rail and Rapid Transport*, vol. 230, no. 2, pp. 360–373, July 2014.

[17] A. Lau, C. Wan, A. H. Løhren, and I. Hoff, "Numerical switch and crossing railpad vertical stiffness optimization," 2018.

[18] B. A. Pålsson and J. O. Nielsen, "Track gauge optimisation of railway switches using a genetic algorithm," *Vehicle System Dynamics*, vol. 50, no. 1, pp. 365–387, July 2012.

[19] A. Lau, "Numerical Simulation of Switch and Crossing Towards an Optimized Turnout Design," NTNU, Trondheim, 2018, Doctoral thesis.

[20] B. A. Pålsson, "Optimisation of railway crossing geometry considering a representative set of wheel profiles," *Vehicle System Dynamics*, vol. 53, no. 3, pp. 274–301, January 2015.

[21] MATLAB, The MathWorks, Inc., Massachusetts, USA, 2021. [Online]. Available: https://se.mathworks.com

[22] V. Profillidis, *Railway Management and Engineering*, 4th ed. Routledge, 2014.

[23] Bane Nor, "Overbygning/Prosjektering/Sporveksler," Bane Nor, Oslo, February 2021. [Online]. Available: https://trv.jbv.no/wiki/Overbygning/Prosjektering/Sporveksler

[24] E. Kassa, C. Andersson, and J. C. O. Nielsen, "Simulation of dynamic interaction between train and railway turnout," *Vehicle System Dynamics*, vol. 44, no. 3, pp. 247–258, 2006.

[25] E. Kassa and J. C. O. Nielsen, "Dynamic interaction between train and railway turnout: full-scale field test and validation of simulation models," *Vehicle System Dynamics*, vol. 46, no. 5, pp. 521–534, April 2008.

[26] voestalpine Turnout Technology Germany, "Kinematic Gauge Optimization FAKOP," voestalpine Turnout Technology Germany GmbH, Butzbach, Germany, 2021. [Online]. Available: https://www.voestalpine.com/turnout-technology-germany/en/products/Kinematic-Gauge-Optimization-FAKOP/

[27] Markku Nummelin, *Railway Turnouts*, 1st ed. Finnish Rail Administration, 2004.

[28] N. Svingheim, "Mot 35 tonns aksellast," Jernbanemagasinet, Oslo, February 2019. [Online]. Available: https://jernbanemagasinet.no/artikler/mot-35-tonns-aksellast/

[29] B. A. Pålsson and J. C. Nielsen, "Wheel–rail interaction and damage in switches and crossings," *Vehicle System Dynamics*, vol. 52, no. 11, pp. 1517–1540, May 2014.

[30] M. R. Bugarín and J.-M. G. D. de Villegas, "Improvements in railway switches," *Journal of Rail and Rapid Transit*, vol. 216, no. 4, pp. 275–286, July 2002.

[31] Bane Nor, "Overbygning/Vedlikehold/Sporveksler," Bane Nor, Oslo, April 2021. [Online]. Available: https://trv.banenor.no/wiki/Overbygning/Vedlikehold/Sporveksler

[32] Alan Wickens, *Fundamentals of Rail Vehicle Dynamics*. Swets and Zeitlinger, 2003.

[33] L. Xin, V. Markine, and I. Shevtsov, "Simulation of railway crossing damage due to welding defect," in *Proceedings of the Second International Conference on Railway Technology*, 2014.

[34] Voestalpine, "Crossings," Voestalpine Railway Systems Australia Pty Ltd, Sydney, Australia, 2021. [Online]. Available: http://www.voestalpine.com/railway-systems-australia/en/solutions/crossings/

[35] P. Wang, *Design of High-Speed Railway Turnouts: Theory and Applications*, 1st ed. Elsevier Inc, 2015.

[36] J. R. Oswald, "Turnout Geometry Optimization with Dynamic Simulation of Track and Vehicle," VAE Aktiengessellchaft, Zeltweg, Austria, 2000.

[37] D. Nicklisch, E. Kassa, J. Nielsen, M. Ekh, and S. Iwnicki, "Geometry and stiffness optimization for switches and crossings, and simulation of material degradation," *Journal of Rail and Rapid Transit*, vol. 224, no. 4, pp. 279–292, May 2010.

[38] A. Johansson, B. Pålsson, M. Ekh, J. O. C. Nielsen, M. Ander, J. Brounzoulis, and E. Kassa, "Simulation of wheel–rail contact and damage in switches & crossings," *Wear*, vol. 271, no. 1-2, pp. 472–481, May 2011.

[39] E. Kassa and J. Nielsen, "Stochastic analysis of dynamic interaction between train and railway turnout, vehicle system dynamics," *Vehicle System Dynamics*, vol. 46, no. 5, pp. 429–449, April 2008.

[40] P. Wang, J. Xu, K. Xie, and R. Chen, "Numerical simulation of rail profiles evolution in the switch panel of a railway turnout," *Wear*, vol. 366-367, pp. 105–115, November 2016.

[41] N. Bogojevic, P.-A. Jönsson, and S. Stichel, "Iron Ore Transportation Wagon with Three-Piece Bogies – Simulation Model and Validation," in *Heavy Machinery - HM2011*, June 2011.

[42] Bane Nor, "Rekordtunge tog på Ofotbanen," Bane Nor, Oslo, February 2019. [Online]. Available: https://www.banenor.no/Nyheter/Nyhetsarkiv/2019/rekordtunge-tog-pa-ofotbanen/

[43] Jernbaneverket, "Prøvekjøring med 32,5 tonns aksellast - Ofotbanen - Hovedrapport," 2017, provided by Helge Voldsund, Jernbanedirektoratet.

[44] A. Berhhuvud and A. Stensson, "Dynamic behaviour of ore wagons in curves at malmbanan," *Vehicle System Dynamics*, vol. 30, no. 3-4, pp. 271–284, July 2007.

[45] Greenwood Engineering. [Online]. Available: https://greenwood.dk/railway

[46] J. Pombo, J. Ambrósio, M. Pereira, R. Lewis, R. Dwyer-Joyce, C. Ariaudo, and N. Kuka, "Development of a wear prediction tool for steel railway wheels using three alternative wear functions," *Journal of Rail and Rapid Transit*, vol. 271, no. 1-2, pp. 238–245, May 2011.

[47] David Thomson, *Railway Noise and Vibration*, 1st ed. Elsevier, 2009.

[48] T. G. Pearce and N. D. Sherratt, "Prediction of wheel profile weart," *Wear*, vol. 144, no. 1-2, pp. 343–351, April 1991.

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