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Structural optimisation of fixed railway crossing panels accounting for eccentric wheel–rail contact loads

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ABSTRACT

The potential for structural optimisation of fixed railway crossing panels is studied. Structural requirements for the crossing panel are established, and a rigorously validated simulation model of dynamic vehicle–track interaction is employed to evaluate the structural loading within the panel. A multi-objective optimisation problem is formulated to minimise the component masses within the crossing panel while ensuring that the induced loading does not exceed their respective limits. This problem is solved using a genetic algorithm and a Pareto front is generated, considering both traffic directions (moves). It is shown that the trailing move is the dimensioning case, due to increased bending stresses in the crossing rail arising from eccentric loading. Using three different weighting strategies, based on CO₂ emissions, deterioration and economic cost, the Pareto solutions are weighted to identify the optimal designs. The economic and deterioration-based weightings show similar trends, favouring a reduction in the crossing rail dimensions at the expense of a larger sleeper cross-section. In contrast, the environmental-based weighting supports a more balanced reduction in both sleeper and crossing rail dimensions.

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1. Background

Railway Switches & Crossings (S&Cs) are frequently classified as ‘hungry assets’ due to their high maintenance costs [1]. In most railway networks, the density of S&Cs is in the order of one per km of track, which corresponds to over 300,000 units within the networks of the EU27 countries [2]. The cost of maintenance of an S&C unit is estimated to be equivalent to that of about 0.3 km of plain line track [2]. In 2015, the annual reinvestment and maintenance cost for the 10 812 S&Cs installed in the Swedish network was estimated to be in the order of 1.045 billion SEK (100 million EUR) [3]. In addition to the high costs associated with installation and maintenance, the rails and sleepers in an S&C contain substantially more steel and concrete per km than plain line track, resulting in a significantly higher CO₂ footprint.

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The rail discontinuities in an S&C lead to high dynamic wheel–rail contact forces, in particular in the crossing panel [4]. This is one of the main reasons for the higher maintenance cost and why a stronger track structure is needed in crossing panels. Failures associated with S&Cs account for 25–30% of all infrastructure failures on the European railways [5].

Based on data from the Swedish network collected between 2014 and 2020, failures in the crossing panel were relatively few in number but resulted in one of the highest average traffic delays per failure, nearly 350 minutes [6]. This may be compared to the switch panel, where the average delay per failure was around 150 minutes, although failures were more common. The number of trains affected by failures in the crossing panel was also among the highest of all infrastructure-related failures considered in the study. Given these numbers, there are strong economic and environmental incentives for structural optimisation of the crossing panel. The objective of this paper is to present a methodology for such optimisation and to discuss the potential benefits and trade-offs associated with alternative crossing panel designs.

Compared to the single-aspect optimisations of S&Cs that have been carried out in recent years [7–11], a structural design optimisation requires more detailed knowledge about the loading and structural constraints of the various components in the crossing panel. Generally, the main focus of previous work has been to reduce the maximum wheel–rail contact force. For example, in [8], the running surface geometry of the crossing rail was optimised to minimise the maximum Hertzian wheel–rail contact pressure for a representative set of wheel profiles. Several studies have focussed on optimising the rail pad stiffness to decrease deterioration related to some of the most common failure modes. Examples considered in the literature include wear and rolling contact fatigue (RCF) of the crossing rails, differential ballast settlement and fatigue of the track components as studied in [9], wheel–rail contact forces and rail seat loads in [10], and loads on the ballast in [11]. While some of these studies have considered the profile of the crossing rail running surface in the optimisation, no study has included the stiffness and mass of the crossing rail as design variables.

All of these studies have assumed that reducing the maximum wheel–rail contact force directly reduces the structural loading and running surface deterioration and thereby increases the service life of the S&C. Thus, the structural requirements of the S&Cs have either been neglected or considered in a very simplistic way, such as by limiting the maximum displacement [10,11] of the crossing rail, with the exception of [9] where rail foot stress was considered using Euler-Bernoulli beam theory. For a structural optimisation of the crossing panel to be meaningful, the change in component loading due to design changes must be accounted for. This introduces additional complexity to the optimisation problem as structural requirements, such as component load limits and representative dynamic load scenarios, must be established. Defining the dynamic loading scenario is particularly challenging as it may vary substantially over the service life of the crossing panel due to changes in support conditions, changes in wheel–rail contact geometry and variations in traffic conditions. In an optimisation problem, due to the significant computational cost, it is not feasible to consider the evolving dynamic vehicle–track interaction throughout the entire service life of the crossing panel. Instead, this paper utilises a representative dynamic loading case, based on the results presented in [12], where different combinations

of axle loads, wheel profiles and train speeds were studied, together with predefined load limits for each component, to formulate the optimisation problem.

2. Simulation model

The applied simulation model of the crossing panel is a representation of the 60E1-500-1:12 demonstrator installed in the Austrian network as a part of the In2Track projects in the EU-funded Shift2Rail research programme [13]. The structural finite element model of the crossing panel, consisting of 37 sleeper spans and originally developed in [14], has been implemented in the commercial multi-body simulation software Simpack. A convergence study was performed in [15] for a similar simulation case, and it was demonstrated that this track model length is sufficient, with no significant influence from the rail boundaries at the crossing transition. It is a two-layer track model with rails and sleepers represented by beam elements. This implementation uses substructures generated by Craig-Bampton model reduction [16] from individual finite element beam models of the crossing rail, stock rails and sleepers, respectively. The superelement node spacing is 15 cm for the crossing rail, 60 cm for the stock rails and varying between 10 and 60 cm for the sleepers. The model incorporates scanned crossing rail surface geometries from in-situ crossings and scanned wheel profiles from in-service wheels. In addition, the cross-sectional properties used for the beam elements, such as the area moment of inertia, vary along the crossing rail and have been determined from a CAD model.

The track model uses linear bushings for the rail fastenings (including rail pads) and bi-linear bushings for the ballast to allow for potentially voided sleepers. The rail fastenings and under sleeper pads (USPs) have varying stiffness depending on their placement within the crossing panel. The fastenings below the crossing rail have a stiffness of 25 MN/m, while the remaining fastenings have a stiffness of 50 MN/m. The USPs have a bulk modulus of 220 or 150 MN/m per m² depending on their placement within the crossing panel. Uniform sleeper support (no hanging sleepers) is considered. Generally, the consideration of hanging sleepers is important for the component loading [12]. However, field measurements of sleeper voids are difficult to perform and the conditions between different locations may vary significantly. Due to these factors, unlike for wheel profiles, no consistent voiding pattern can be assumed and since each hanging sleeper configuration influences the track components differently, it is computationally infeasible to include all possible configurations in the optimisation. To compensate for the neglect of potentially poor sleeper support conditions, the vehicle model uses a field-scanned 4 mm hollow worn wheel profile. In [12], it was shown that this approach gives higher structural track loading than extreme voiding conditions. This approach is sometimes used in the railway industry to represent a severe dynamic load case, as sleeper support conditions are generally considered to be unknown. Full details about the crossing panel model are given in [14].

The vehicle model includes one bogie based on the Manchester benchmark vehicle [17]. Based on the conclusions from [12], the model has been adapted to an axle load of 30 tonnes, a vehicle speed of 60 km/h and a wheelbase of 1.92 m, corresponding to the locomotives on the iron ore line in Sweden. In this paper, all simulations consider the bogie running in the through route, as the generated vertical loading has been observed to be more severe in this route due to the higher speeds. The output from the simulations is sampled at a frequency of 2500 Hz. The wheel–rail contact algorithm in Simpack is the discrete

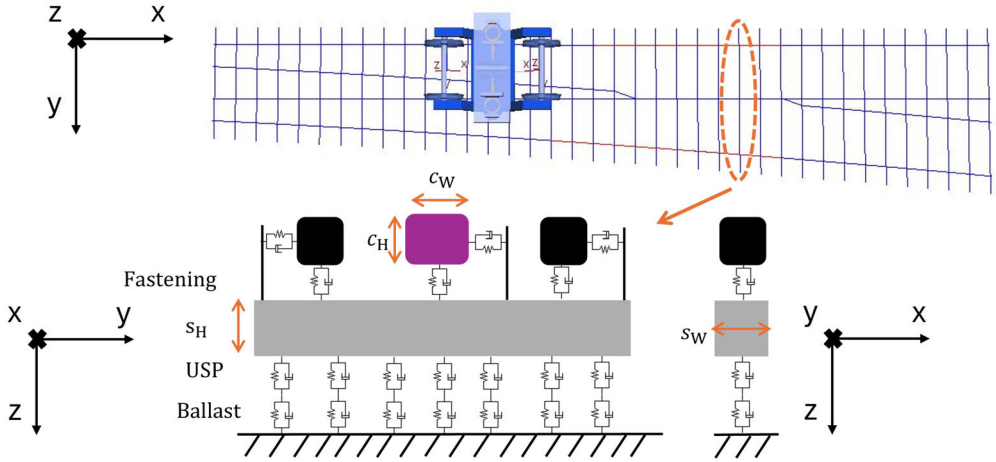


Figure 1. Vehicle and track models implemented in Simpack. Top view of full model, cross-section view and side view of one sleeper. The sleeper and crossing heights and widths are given as s_H , s_W , c_H and c_W , respectively.

elastic contact method, which is based on the semi-Hertzian STRIPES method [18,19]. A modified version of the FASTSIM [20] algorithm is employed to model the tangential contact.

The model as implemented in Simpack is shown in Figure 1. One cross-section of the crossing panel is illustrated with rail fastenings between sleeper and rail, as well as the support stiffness between sleeper and ground modelled with two springs connected in series, one representing the USP and the other the ballast and subgrade (check rails are omitted in the picture but are included in the model).

Large variations of in-situ track bed modulus (the combined stiffness of the ballast and subgrade) can be expected. Recent experimentally determined values obtained using either track recording cars or dedicated track instrumentation are reported in [21,22], with measured track bed modulus ranging from about 50 MN/m³ to 140 MN/m³. As found in the literature on the subject, values used in simulation models generally vary within similar ranges. For example, in the S&C benchmark [1], the value 60 MN/m³ was used. Based on the extensive field measurements in the Shift2Rail crossing demonstrator in Austria, the track bed modulus was calibrated to 120 MN/m³ [14,23].

Track displacements are directly available as output from each simulation, while sleeper–ballast contact pressures and normal stresses due to bending of crossing rails and sleepers are computed in a postprocessing step as described in [14]. The model has been extended in this paper to include rail torsion, and the resulting contribution to bending stress arising from Vlasov torsion. Due to the shape of the open cross-section in most crossing rails, an example of which is shown in Figure 2, the contribution from Vlasov torsion can be significant, especially in the trailing move where eccentric impact loading generates a torque. The Vlasov term is added to the original Euler–Bernoulli bending stress in Equation (1),

$$\sigma_{xx}(x, y, z) = -\frac{M(x)}{I(x)}z(x) + \frac{B(x)}{K_w(x)}\Psi(x, y, z) \quad (1)$$

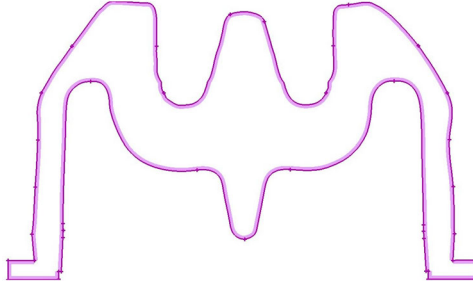


Figure 2. A typical open rail cross-section near the location of wheel–rail impact along the crossing rail.

where $\Psi(x, y, z)$ is the warping function along the beam, which is simplified to only include one point per cross-section, namely the point where the maximum total bending stress is expected, $B(x)$ is the bi-moment, and K_w is the warping constant, which is dependent on the cross-sectional geometry. The torsional and warping properties of the cross-sections are determined from a CAD model of the crossing rail.

The bi-moment along the crossing rail is defined as

$$B(x) = -EK_w(x) \frac{d^2\phi(x)}{dx^2} \quad (2)$$

where $\phi(x)$ is the torsion of the crossing rail.

The model has been validated against an extensive set of measurement data from the demonstrator S&C, including displacements and stresses in sleepers and crossing rail, and sleeper–ballast contact pressure [14]. The bending strain with the added Vlasov term is validated against the measurement data from strain gauges in Figure 3, with the measurement positions located on the bottom of the crossing rail. A significant improvement is seen for the trailing move, where the torsion is largest due to the eccentric impact loading. It was found that the effect of the Vlasov contribution on the maximum stress varies from nearly 0% in the solid cross-sections of the crossing rail to as much as 15% in the open cross-sections near the impact location when traversed in the trailing direction. A smaller improvement is also seen for the facing move.

However, even with this refinement, beam theory does not fully capture the increase in stress arising from the complex geometry present at some longitudinal positions along the crossing rail. To account for this, a scaling factor of 1.4 has been applied to all calculated values of crossing rail bending stress. This factor varies along the crossing rail, but the selected value corresponds to the maximum discrepancy identified when comparing the beam finite element model with a 3D solid finite element model under clamped-clamped boundary conditions and a force applied at the approximate shear centre (closest node) of different cross-sections within the potential wheel transfer zone. The position of the maximum stress predicted by the 3D model corresponds to the position used for extracting the warping function $\Psi(x)$.

The beam model used in this paper, but without the torsional degree-of-freedom and the Vlasov torsion term, has previously been compared with a model incorporating a full 3D representation of the crossing rail, see [23]. The comparison showed that the main benefit of the more detailed model is its ability to capture the variation in stresses due to the

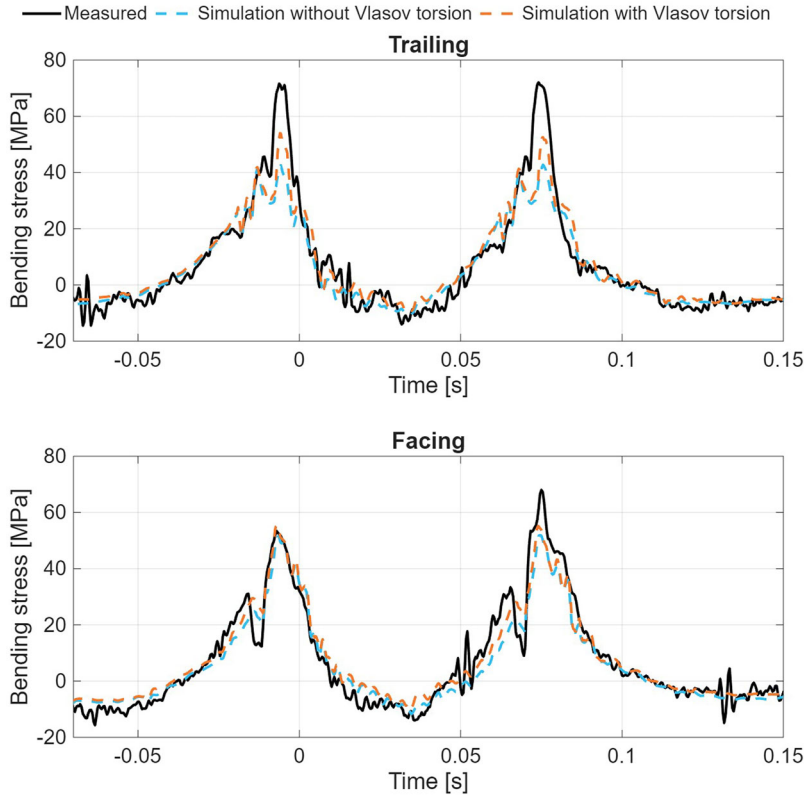


Figure 3. Measured stress beneath the crossing nose compared with simulated stress with or without the inclusion of Vlasov torsion.

complex geometry at the bottom of the crossing rail, but at the cost of a higher computational effort. However, due to the intricate 3D mesh required to represent this geometry, the cross-sectional dimensions are considerably more challenging to parameterise within an optimisation framework.

3. Optimisation problem

The studied optimisation problem for the crossing panel is formulated in this section. In total, six scaling (non-dimensional) design variables are used. Two of these are related to the linear track stiffness of the simulation model:

- h_{kf} - Rail fastening stiffness scaling
- h_{kb} - Ballast stiffness scaling

To vary the size of the crossing panel components, four design variables are introduced to linearly and uniformly scale the heights and widths of the beam elements representing the crossing rail and the sleepers:

- h_{cW} - Crossing rail width scaling

- h_{cH} - Crossing rail height scaling
- h_{sW} - Sleeper width scaling
- h_{sH} - Sleeper height scaling

For each cross-section of the crossing rail, the scaling of the vertical and lateral area moments of inertia and the torsional constant used to model the beam elements is assumed to be proportional to a rectangular cross-section shape. Thus, this scaling is defined as

$$I_{s,vert}(x) = I_{n,vert}(x)h_{cW}h_{cH}^3 \quad (3)$$

$$I_{s,lat}(x) = I_{n,lat}(x)h_{cW}^3h_{cH} \quad (4)$$

$$K_{s,v}(x) = K_{n,v}(x)h_{cW}^3h_{cH} \quad (5)$$

where $I_{n,v}$, $I_{n,l}$ and $K_{n,v}$ are the nominal vertical and lateral area moments of inertia and the torsional constant, respectively. The same procedure is adopted for the scaling of the sleeper properties. Since the warping constant and warping function of a closed rectangular cross-section very close to 0, meaning that close to no Vlasov torsion occurs, this simplification must be adjusted when determining the warping properties. In this work, the formulas for an I-beam are used to scale the warping constant and the warping function

$$\Psi_s(x) = \Psi_n(x)h_{cW}h_{cH} \quad (6)$$

$$K_{s,w}(x) = K_{n,w}(x)h_{cH}^3h_{cW}^3 \quad (7)$$

where Ψ_n is the nominal warping function and $K_{n,w}$ is the nominal warping constant. This scaling approach has been verified against a uniformly scaled version of the crossing rail cross-section, resulting in a maximum deviation of less than two percent.

To keep the dimensionality of the problem as low as possible, the design variable vector $\mathbf{u} = [h_{cA} \ h_{sA}]$ is used in the following structural optimisation problem, where h_{cA} is the crossing rail area scaling factor and h_{sA} is the sleeper area scaling factor. The mass is scaled linearly with the area scaling (height scaling multiplied by width scaling). To uniformly scale the dimensions of each cross-section, $h_{iW} = h_{iH} = \sqrt{h_{iA}}$, where $i \in [c, s]$, are adopted. These area-scaling variables are used here as a simplified metric, recognising that any practical modifications to rail and sleeper cross-sections in the field may need to be more complex to ensure safe operation and manufacturability. Furthermore, the crossing rail running surface is left unchanged by the geometric modifications, i.e. all geometric changes apply solely to the beam properties representing the crossing rail. Thus, the optimisation results in target values for the stiffness and mass properties of the components. While the exact design is beyond the scope of this paper, the design variable limits were chosen such that they could feasibly be achieved even when additional manufacturing and service constraints are taken into account.

Table 1. Component load limits used in the optimisation.

Component loading	Limit value	Limit category
Crossing rail bending stress (σ_c)	$b_c = 168$ [MPa]	Fatigue limit [12]
Sleeper bending stress (σ_s)	$b_s = 10.6$ [MPa]	Crack initiation limit [12]
Sleeper–ballast contact pressure (σ_b)	$b_b = 0.3$ [MPa]	Load bearing limit [12]

The optimisation problem P_1 to minimise the mass of the crossing panel is formulated as

$$\begin{aligned}
 P_1 : \quad & \min_{\mathbf{u}} \quad (m_c(\mathbf{u}), m_s(\mathbf{u})) \\
 \text{s.t.} \quad & \Omega = \{\mathbf{u} \mid 0.5 \leq h_{cA} \leq 1.5, 0.5 \leq h_{sA} \leq 1.5\} \\
 & g_1(\mathbf{u}) = \sigma_c(\mathbf{u}) - b_c < 0 \\
 & g_2(\mathbf{u}) = \sigma_s(\mathbf{u}) - b_s < 0 \\
 & g_3(\mathbf{u}) = \sigma_b(\mathbf{u}) - b_b < 0
 \end{aligned} \tag{8}$$

where m_s is the total mass of the sleepers and m_c is the mass of the crossing rail. Furthermore b_c , b_s and b_b are the specified load limits for crossing rail, sleeper and ballast, respectively. Based on the pre-study presented in [12], all specified constraint values are presented in Table 1.

Due to the difficulty of reliably controlling track stiffness in the field, and in order to reduce the computational effort of the optimisation problem, the track stiffness variables have not been included as design variables. As they would ideally be included to achieve an optimal design, several cases of the optimisation problem are instead solved, each with different constant values of the track stiffness parameters, to assess their influence on the optimisation outcome. In the field, the support conditions will be different for each S&C individual. The approach adopted in this paper is therefore to consider extreme support conditions, namely very stiff and very soft, to ensure that the selected design values are suitable across the full range of conditions. Two stiffness-scaling levels for ballast (h_{kb}) have been specified in combination with three scalings for the rail fastening stiffness (h_{kf}), resulting in the six optimisation cases specified in Table 2. These cases are evaluated for both the facing and trailing moves, yielding a total of twelve optimisation cases. This approach provides insight into how variations in track stiffness relate to the balance of the structural constraints in both running directions. Based on the field observations discussed earlier, the ballast bed modulus is varied between 60 MN/m³ ($h_{kb} = 0.9$) and 140 MN/m³ ($h_{kb} = 2.1$). The rail fastening stiffness under the crossing rail is varied between 25 MN/m ($h_{kf} = 1$) and 100 MN/m ($h_{kf} = 4$), and between 50 MN/m and 200 MN/m in the remainder of the crossing panel. The lowest scaling factor ($h_{kf} = 1$) corresponds to the fastening stiffness used in the S2R demonstrator.

The multi-objective optimisation problem was solved using the genetic algorithm *gamultiobj* in MATLAB, which is based on the algorithm published in [24]. This method was selected primarily for its efficiency in parallelisation, which is essential given the substantial computational effort required, where the optimisation demands an extensive simulation of dynamic vehicle-track interaction for each function evaluation. The parameters controlling the genetic algorithm, such as population size and maximum number of generations, were kept as low as possible. Since the aim of the optimisation is to identify

Table 2. Different combinations of scale factors for rail fastening stiffness h_{kf} and ballast stiffness h_{kb} used in the optimisation. Cases 1–6 are in the facing move, while 7–12 are the same six cases but in the trailing move. The rail fastening stiffness varies within the crossing panel. Here, the nominal lower and upper limits are given.

Case	1	2	3	4	5	6	Nominal value
h_{kf}	1	1	2	2	4	4	25–50 [MN/m]
h_{kb}	0.9	2.1	0.9	2.1	0.9	2.1	66.6 [MN/m ³]

trends and provide guidance for further development, strict convergence was not required. For all optimisation cases, the population size was set to 36 and the maximum number of generations to 27, resulting in 972 simulations per optimisation case in Table 2. Larger values of the population size and maximum number of generations however evaluated for one optimisation case and resulted in only minor changes to the final solution, while the computational cost increased significantly. From each optimisation, the Pareto front was determined, representing the set of optimal solutions for which no objective can be improved without worsening at least one other objective. For a multi-objective optimisation problem with objective function vector $\mathbf{f} = [f_1 \ f_2]$, a point $\mathbf{x}$ is said to dominate a point $\mathbf{y}$ if:

$$\begin{cases} f_k(\mathbf{x}) \leq f_k(\mathbf{y}) & \text{for all } k. \\ f_k(\mathbf{x}) < f_k(\mathbf{y}) & \text{for some } k. \end{cases} \quad (9)$$

The Pareto front was determined for each optimisation case to assess how the ballast and rail fastening stiffnesses affect the optimisation results. The Pareto front alone cannot identify a single optimal solution, as the preferred design depends on the weighting applied to the objective functions.

It is proposed that relevant weighting strategies focus on minimising track deterioration, economic cost, or environmental impact. Applying these perspectives to weight the objective functions enables a more precise optimum design solution for each strategy. These weighting approaches are examined in more detail in Section 5.

4. Results from multi-objective optimisation

The Pareto front for each optimisation case is presented for the facing and trailing moves in Figures 4 and 5, respectively. As the results for all cases showed a convex solution, the Pareto fronts with insufficient resolution were manually refined by adding a small number of additional simulations to allow clearer comparison between cases. For some cases, no horizontal tail of the Pareto front could be identified, i.e. increasing the sleeper dimensions did not permit a corresponding reduction in crossing rail dimensions. This occurred, for example, in optimisation cases 2 and 4. The cause for this is that for these cases the increase in sleeper mass increases the maximum wheel–rail contact force more than it relieves the crossing rail due to the increase in stiffness provided, leading to an increase in rail stress with increased sleeper dimensions.

The main difference between the facing and trailing moves is the higher loading of the crossing rail in the trailing move. This is caused by the impact load on the wing rail, which induces torsion in the crossing rail and consequently results in a larger required crossing rail section for the trailing move. Therefore, the trailing move is the dimensioning running

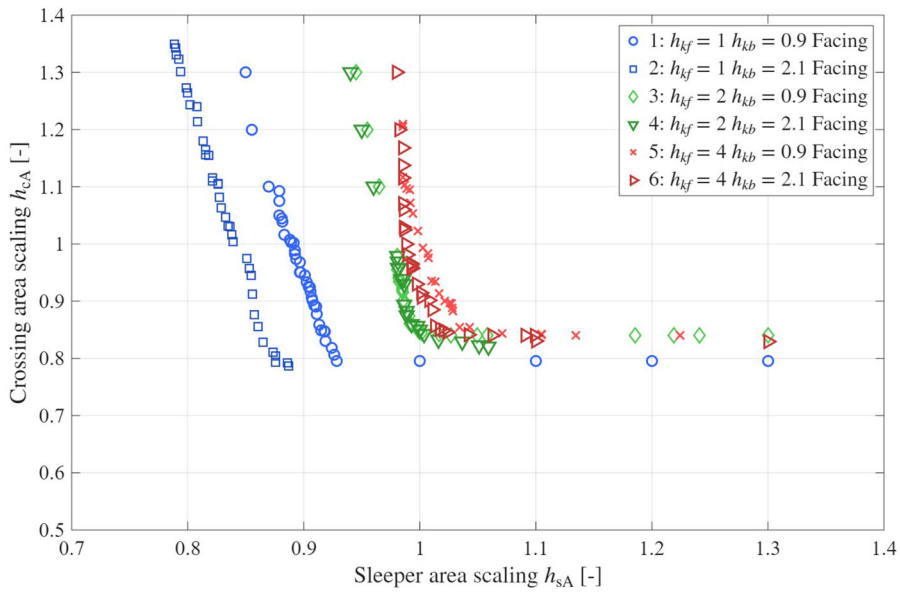


Figure 4. Pareto fronts for all optimisation cases in the facing move.

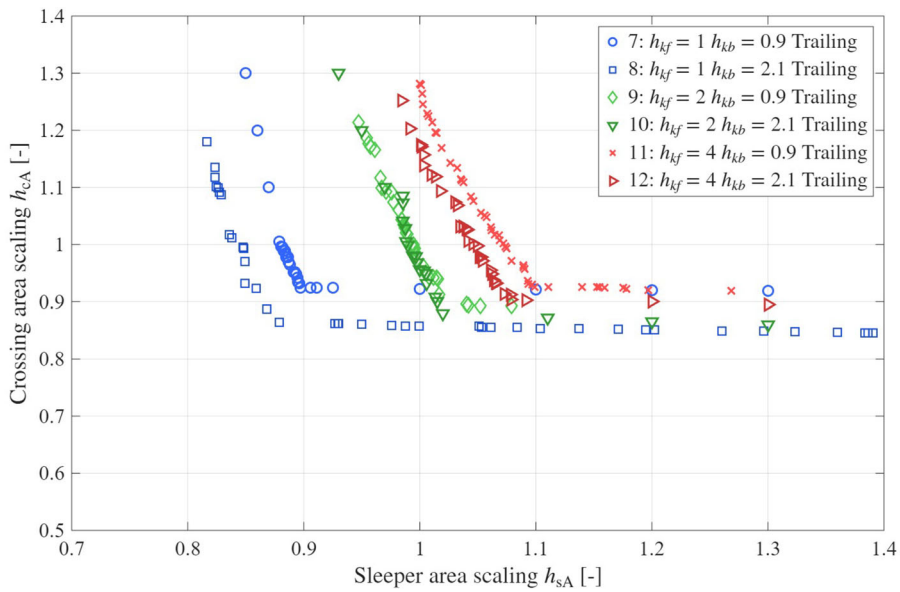


Figure 5. Pareto fronts for all optimisation cases in the trailing move.

direction. Overall, a trend emerges in favour of softer rail fastenings in both directions, as they allow for significantly smaller sleeper dimensions. In addition, soft fastenings also yield the smallest possible crossing rail dimension for both trailing and facing moves. As specified above, the load case used in the optimisation was based on a very hollow worn

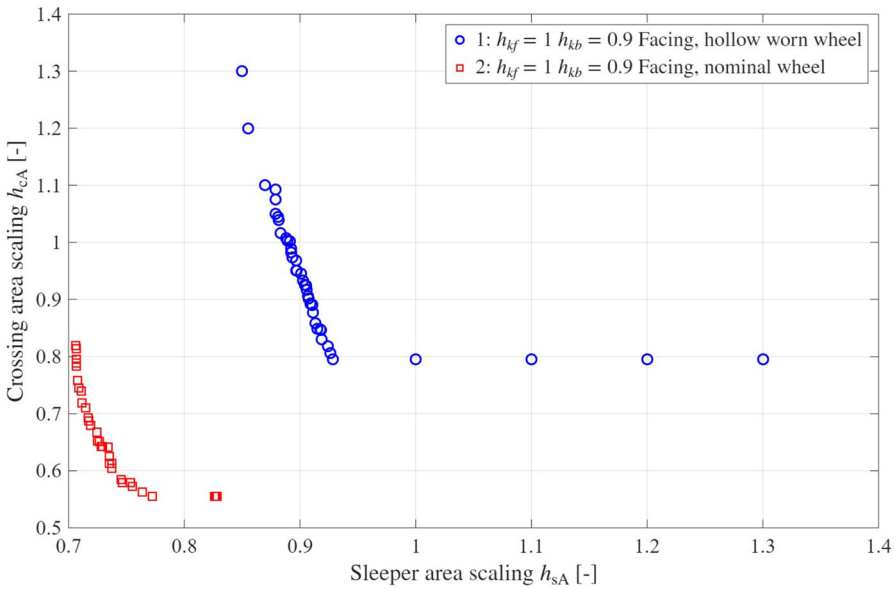


Figure 6. Comparison of Pareto fronts for Case 1 using a hollow worn wheel and an identical case but with a nominal S1002 wheel profile.

wheel. If instead a nominal wheel is assumed, the reduction possible increases significantly. One such optimisation was performed using a nominal wheel while keeping the same ballast and fastening stiffnesses as in case 1. The comparison is shown in Figure 6.

From Figures 4 and 5 it is evident that a higher rail fastening stiffness increases the structural loading of the sleeper. In Sweden, the most recently implemented type of S&C uses rail fastenings with a static stiffness of around 60 MN/m, which can be compared to cases 3, 4, 9 and 10. Older generations of S&C still present in the railway network are significantly stiffer, with a rail fastening stiffness of up to 1200 MN/m [25]. According to the Pareto fronts presented here, such a high fastening stiffness would be expected to result in even higher sleeper loading, leading to sleeper fracture. When this is considered together with the fact that sleepers are underrepresented in failure statistics and generally exhibit long service lives, it suggests that the dimensioning load case used here may be overly severe. Alternatively, it may indicate that the design load prescribed in sleeper standards is conservative relative to the actual failure limit of sleepers, or a combination of both.

While the rail fastening stiffness can be controlled with relatively high reliability, the ballast stiffness is far more difficult to manage and varies between locations and over time due to environmental conditions and differential settlement. The implementation of under sleeper pads could be one measure to better stabilise and control the long-term sleeper support stiffness. However, to what extent this can be achieved has not been studied here. The results in Figure 5, nevertheless, indicate that there is a benefit in specifying the under sleeper pad modulus to achieve a target support stiffness at the installation site, as softer ballast performs better, see cases 7 and 8.

To reduce the economic and environmental costs, optimal material use for each component is desirable. This implies that, in an optimally designed structure, each component is subjected to a structural loading that is close to the specified structural constraint,

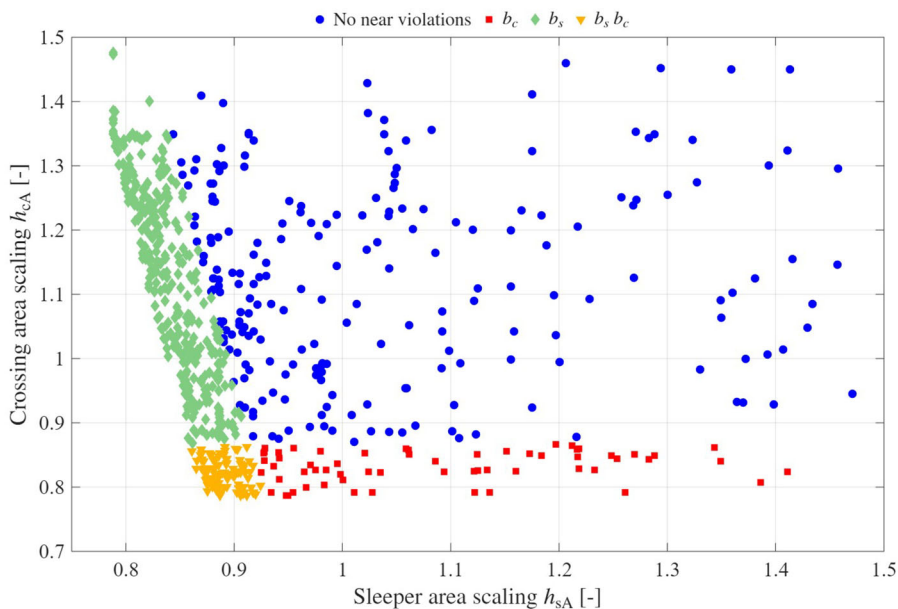


Figure 7. All feasible points (combinations of crossing rail area and sleeper area that fulfil all constraints) plotted with indicators of active constraints (within 10% of the limit) for optimisation case 2.

assuming that the worst-case loading scenario has been considered. Among the constraints considered in the optimisation studied in this paper, the sleeper–ballast contact pressure does not inherently benefit from operating near its limit. In contrast, the other two constraints (crossing rail stress and sleeper stress) do exhibit such benefits, since proximity to their limits corresponds to efficient use of the component material. For this reason, an optimally designed crossing panel in terms of optimal material use, under the worst-case dynamic load scenario, would be expected to operate close to both of these stress-related constraints. However, other considerations, such as CO₂ emissions, deterioration mechanisms or maintenance strategies may shift this balance and influence what constitutes an optimal design in practice.

To examine the constraint proximity, all feasible points from the optimisation of case 2 (soft rail fastening and stiff ballast) have been plotted in Figure 7 as an example. If a metric is within 10% of its constraint value, it is considered to be nearly violated and therefore denoted with a specific marker and colour, depending on the combination of nearly violated constraints. For example, the green diamond markers indicate situations where the constraint for the sleeper bending stress is nearly reached, while the red square markers indicate situations where the crossing rail bending stress is close to its constraint value. Optimal material use is achieved when both b_c and b_s are simultaneously near their respective constraints, which is represented by the orange triangle markers in Figure 7. Generally, similar behaviour regarding constraint proximity is observed across all optimisation cases, with the main difference being that some cases are closer to convergence than others.

The component stresses that were calculated for case 2 are presented in Figure 8, where the colour of each marker indicates the magnitude of the loading in relation to the constraint value. The upper limit of each colour scale is set equal to the constraint value. In

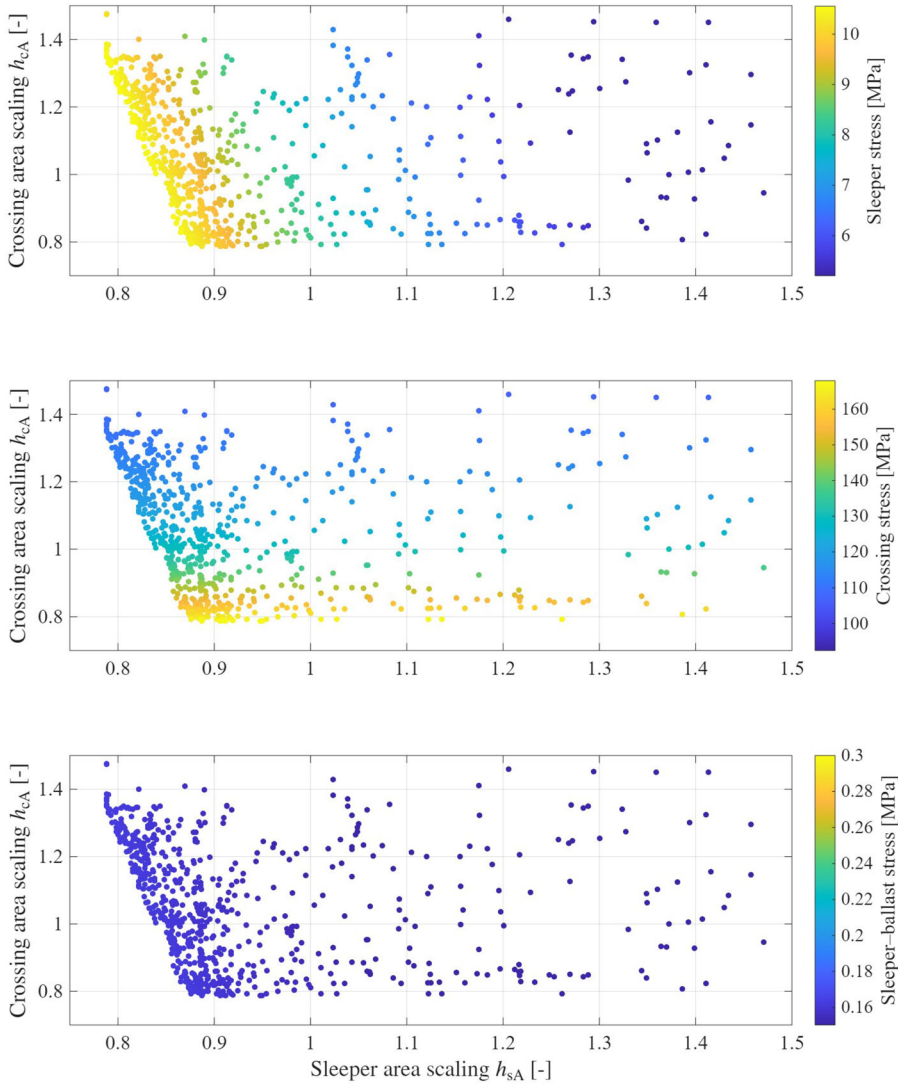


Figure 8. The calculated structural loading for all feasible points (combinations of crossing rail area and sleeper area that fulfil all constraints) for each constraint in optimisation case 2.

the two upper subfigures, it can be seen that the calculated bending stress for each component is strongly correlated with the corresponding dimensions. This highlights that a balanced solution exists in which both components can be simultaneously reduced in size while maintaining acceptable stress levels.

It was observed that none of the combinations of crossing rail area and sleeper area, in any of the optimisation cases, resulted in a solution where the sleeper-ballast contact pressure constraint was violated. In the optimisation framework used in this study, any detrimental effects associated with increasing sleeper-ballast contact pressure, provided it remains below the constraint value, have been neglected. However, as an increasing sleeper-ballast contact pressure is expected to increase track settlement, such effects could

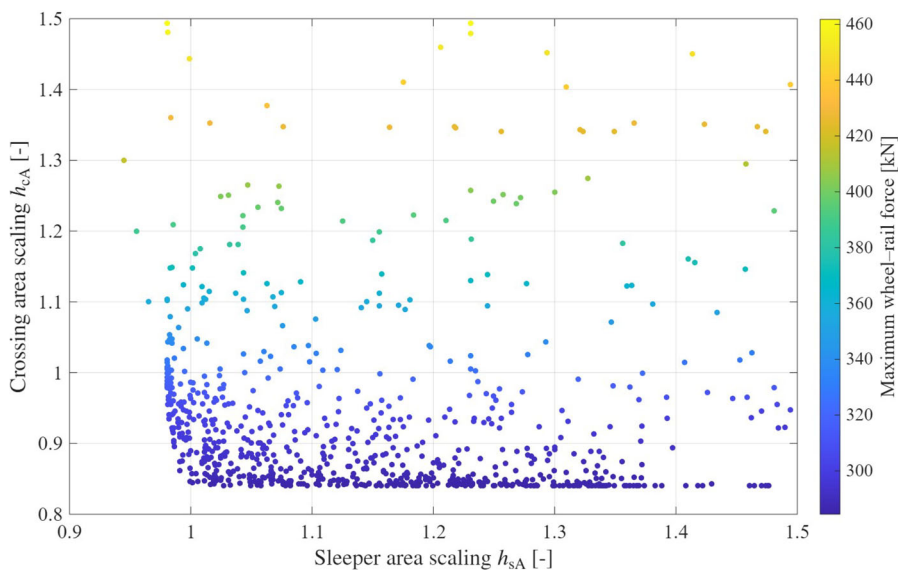


Figure 9. The calculated maximum wheel-rail contact force for all feasible points for optimisation case 3.

be incorporated in future optimisation studies, for example through the implementation of an appropriate penalty function based on a settlement model.

One example of how variations in the crossing rail and sleeper cross-sectional areas influence the maximum wheel-rail contact force is shown in Figure 9. The example is taken from case 3, but similar trends were observed across all optimisation cases. While the low-frequency contribution (up to 250 Hz) of the wheel-rail contact force was obviously considered in the optimisation, structural displacement and loading are predominantly influenced by the applied load within this frequency range. Consequently, it may be argued that the high-frequency (above 250 Hz) content of the contact force may not have been fully accounted for within the optimisation framework used in this study. High-frequency contact forces are nevertheless important, as they can contribute to surface damage of the crossing rail and should therefore be considered in some manner. The major trend seen in Figure 9 is that the maximum wheel-rail contact force decreases as the cross-sectional area of the crossing rail is reduced.

5. Weighting strategies

As previously discussed in Section 3, weighting can be applied to the Pareto front to reduce the set of optimum solutions. In this section, three different weighting strategies are examined: environmental impact, deterioration and economic cost.

5.1. Environmental impact

To optimise the design of the crossing panel with respect to reduced environmental footprint, and to identify the optimal combination of component dimensions within the Pareto front, an objective function based on the CO₂ emissions associated with each component

in the crossing panel has been defined. The total CO₂ emission of the crossing panel is evaluated as

$$f_{\text{CO}_2, \text{total}} = m_s \text{CO}_{2, \text{eq}, s} + \gamma_L m_c \text{CO}_{2, \text{eq}, c} \quad (10)$$

where m_s and m_c are the total sleeper mass and crossing rail mass (nominally $m_s = 6400$ kg, $m_c = 1040$ kg), respectively, while $\text{CO}_{2, \text{eq}, c}$ and $\text{CO}_{2, \text{eq}, s}$ are the equivalent carbon dioxide emissions ($\text{CO}_{2, \text{eq}}$) per kg of crossing rail and sleeper as specified in representative environmental product declarations (EPDs) [26,27]. For the total sleeper mass, the mass of 10 crossing panel sleepers is considered. Furthermore, γ_L is a weighting factor introduced to account for differences in the expected life between the sleepers and the crossing rail. To determine this factor, the thresholds of accumulated traffic load in million gross tonnes (MGT) that trigger replacement of different components within the S&C, according to the Swedish Transport Administration, have been considered [28]. These thresholds allow a relationship to be established between the service lives of the crossing rail and the sleepers. Typically, sleepers within the S&C have four times longer life than the crossing rail. However, according to the Swedish infrastructure manager, there are cases where isolated sleeper replacement in the crossing panel may occur earlier in an attempt to improve ballast conditions near the crossing rail. In such cases the sleepers may effectively have only twice the service life of the crossing rail. Nevertheless, as the life of sleepers is normally four times longer than the crossing rail, the weighting factor γ_L has been set to 4.0. Potential variations in service life due to changes in component dimensions, such as those that might arise from reduced wheel–rail contact forces, have been neglected here.

For each of the generated Pareto fronts in the trailing move, the dimensioning running direction, the calculated values of the objective function in Equation (10), normalised by the nominal value, are plotted in Figure 10. The optimal solution is indicated by a red circumference. This solution corresponds to a case with the lowest fastening stiffness. In addition, it also represents optimal material usage for both sleeper and crossing rail, as it shows proximity to both constraints. The best-performing solution features soft rail fastenings combined with stiff ballast and results in a CO₂ footprint reduction of 13% relative to the nominal design, even when accounting for the severe dynamic loading due to a 4 mm hollow worn wheel profile. If the ballast stiffness is unknown, the lower ballast stiffness level must be assumed, which reduces the maximum achievable emission reduction to 8% relative to the nominal design. Overall, the results show that reducing the crossing rail dimensions is the most effective measure to achieve a lower environmental footprint, whereas the sleeper dimensions have a smaller impact. This could be partially attributed to the shorter service life of the crossing rail. However, if γ_L was reduced to 1, i.e. the crossing rail and sleeper are assumed to have equal service life, the influence of the sleepers was increased, but the same optimum solution was found.

5.2. Deterioration

Similar objective functions can be established for metrics related to crossing panel deterioration, such as surface- and subsurface-initiated RCF, plastic deformation, wear, differential settlement and rail pad fatigue.

The potential initiation of surface-initiated RCF cracks have a relationship with the magnitude of rail wear, which can be approximated using the simplified estimate of railway wear

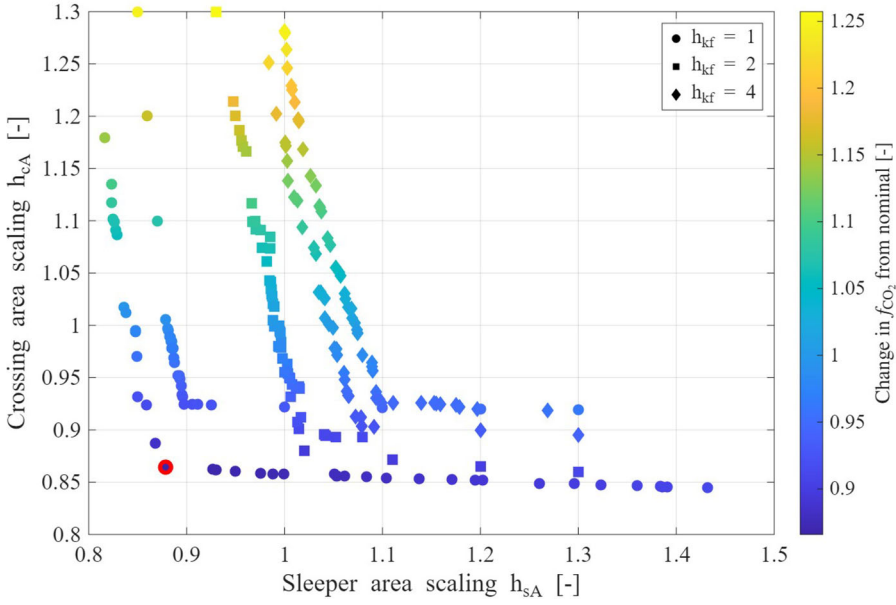


Figure 10. Calculated f_{CO_2} emissions for each of the Pareto points from the optimisation cases in the trailing move with $\gamma_L = 4$. The best-performing solution is indicated by a red circumference. The results are normalised against the f_{CO_2} emissions of the crossing panel with nominal sleeper and crossing rail area scalings. Fastening stiffness scaling is indicated by marker shape.

$T\gamma$ [29]. This measure is proportional to the energy dissipated in the wheel–rail contact patch and is calculated as the product of creep forces/torques (T_x, T_y, M_z) and creepages (v_x, v_y, φ_z), expressed as

$$T\gamma = |T_x v_x| + |T_y v_y| + |M_z \varphi_z| \quad (11)$$

According to [29], the risk of surface-initiated RCF increases with increasing $T\gamma$, up to a certain magnitude, where the wear rate is high enough to wear away the material containing the developing RCF cracks and the fatigue life due to RCF is increased. This implies that certain regimes of $T\gamma$ are more critical for RCF initiation and propagation than others. These critical regimes are material-dependent and have not yet been determined for the cast manganese steel used in modern crossing rails. Another approach to quantify the risk of surface-initiated RCF, based on a Hertzian wheel–rail contact model, is the fatigue index FI_{surf} [30], which predicts that fatigue will occur if $FI_{surf} > 0$.

$$FI_{surf} = \mu - \frac{k}{p_0} \quad (12)$$

Here, k is the material yield stress in pure shear, p_0 is the maximum Hertzian contact pressure and μ is the utilised friction coefficient in the wheel–rail contact, defined as the quotient between the tangential creep forces and vertical loads as

$$\mu = \frac{F_{\text{tangential}}}{F_z} = \frac{\sqrt{T_x^2 + T_y^2}}{F_z} \quad (13)$$

As the contact model used in this paper is semi-Hertzian, the local maximum contact pressure obtained from the simulation may be conservative if inserted directly into the fatigue index. To mitigate this, the maximum contact pressure has been recalculated under the assumption of fully Hertzian contact, expressed as

$$p_0 = \frac{3F_z}{2A} \quad (14)$$

where F_z is the vertical wheel–rail contact force and A is the contact area for the given contact point, as calculated by the contact algorithm. This yields the following expression for the surface fatigue index

$$FI_{\text{surf}} = \mu - \frac{2kA}{3F_z} \quad (15)$$

For the subsurface-initiated RCF, the fatigue index FI_{sub} is used, see [30],

$$FI_{\text{sub}} = \frac{p_0}{6}(1 + \mu^2) + a_{\text{DV}}\sigma_{\text{h,res}} > \sigma_{\text{EQ},e} \quad (16)$$

where a_{DV} is the Dang-Van material parameter and $\sigma_{\text{EQ},e}$ is the equivalent fatigue limit. Based on the study conducted in [31], a_{DV} was set to 0. With this assumption, the equivalent fatigue limit $\sigma_{\text{EQ},e}$ can be simplified as the shear fatigue limit τ_f , which according to the experiments in [31] was determined as 278 MPa for cast manganese steel. This results in that subsurface-initiated RCF can be expected when the normalised subsurface fatigue index $FI_{\text{sub,n}}$ exceeds unity

$$FI_{\text{sub,n}} = \frac{\frac{p_0}{6}(1 + \mu^2)}{\tau_f} > 1 \quad (17)$$

The simulations in Simpack indicated that there is significant risk for RCF in the impact region on the crossing nose in the trailing move (wing rail for the facing move) during the transition (not shown here), but also that the influence of component dimensions was negligible. For this reason, the RCF indices are here only calculated for one rail per move: wing rail for the trailing move, and crossing nose for the facing move.

To assess the risk of rail fastening fatigue and the load on the sleepers, the rail seat load metric is used. In Simpack, this is calculated as the sum of the bushing forces representing the rail fastening connecting the sleeper beneath the crossing transition to the respective rails. Furthermore, the increase in magnitude of plastic deformation due to the dynamic loading is here assumed to be proportional to the maximum dynamic wheel–rail contact force $Q_{\text{d,max}}$, which is given as the maximum wheel–rail contact force subtracted by the static wheel load.

Unlike the CO₂ emissions, these deterioration metrics are directly and significantly influenced by the specified rail fastening and ballast stiffnesses values. To isolate the effect of the design parameters, the weighting was performed for the ballast and fastening stiffnesses of case 8, since this case yielded the optimum solution in the environmental weighting. However, instead of the hollow worn wheel profile used in the optimisation, simulations for the deterioration metric were performed with a nominal S1002 wheel profile. For each metric, the maximum value has been normalised by the corresponding value from the simulation with the nominal crossing panel design, i.e. with both area scalings for sleeper

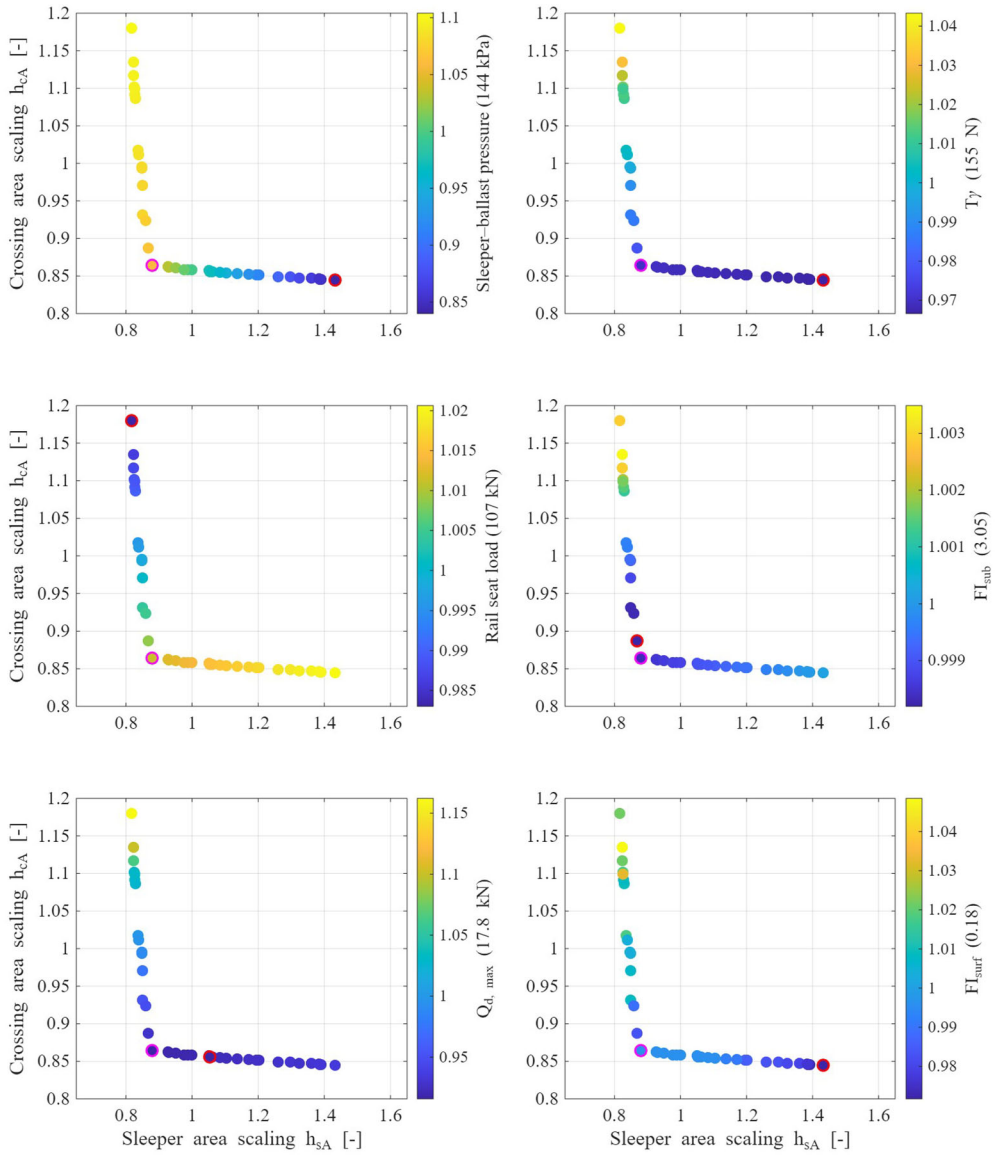


Figure 11. Effect on the normalised deterioration metrics due to changes in component dimensions for the fastening and ballast stiffnesses of optimisation case 8 with a nominal S1002 wheel profile. The nominal deterioration metric values used for normalisation are given in parentheses. The optimum for each metric is indicated with red circumference, while the optimum identified by the $CO_{2,eq,c}$ weighting is indicated with a magenta circumference.

and crossing rail set to 1, and using a nominal wheel profile. The results are presented in Figure 11.

Overall, the observed trend points towards minimising the crossing rail cross-sectional area at the expense of a larger sleeper. For the maximum dynamic wheel–rail contact force, however, the benefits of further reducing the crossing rail dimensions are eventually outweighed by the increasing sleeper size. For some deterioration metrics, such as

the subsurface-initiated RCF metric, the influence of the component dimension scaling is negligible. The change in surface-initiated RCF shows a general trend of decreasing with smaller crossing rail dimensions, with some outliers. A more compliant crossing rail does, however, lead to a higher maximum rail seat load due to the more localised transfer of forces.

The optimum found using the environmental weighting strategy is indicated by a magenta circumference in Figure 11. This solution exhibits slight increases in sleeper–ballast contact pressure and rail seat load, but achieves reductions in the remaining deterioration metrics. This demonstrates that the relatively large CO₂ emission reduction of 13% can be achieved with predominantly positive effects on the deterioration-related performance of the crossing panel.

To investigate how the fastening stiffness and ballast stiffness influence the trends of the design variables, all Pareto fronts for the facing move are plotted in Figure 12. The overall trends for the deterioration metrics remain similar, indicating that the effects of the design variables are largely independent of the fastening and ballast stiffness levels. However, while soft rail fastenings allow for smaller crossing rail and sleeper dimensions, they result in higher values for both RCF metrics, although, the effect on the surface-initiated RCF is negligible. This behaviour is attributed to the impact load occurring slightly earlier, which decreases the contact area during impact and thereby increases the contact pressure. This result is highly dependent on the wheel profile used, and should therefore not be interpreted as a general trend. In contrast, softer rail fastenings improve the rail seat load, wheel–rail contact force, and $T\gamma$.

The sleeper–ballast contact pressure is mainly influenced by the sleeper dimensions and the ballast stiffness. Apart from the direct effect on the sleeper–ballast contact pressure, the influence of ballast stiffness on the deterioration metrics is similar to that of the rail fastening stiffness but generally weaker.

5.3. Economic cost

The historical MGT thresholds for various maintenance activities in Sweden, along with the corresponding labour and material costs for each activity, are presented in [28]. Although it is difficult to quantify how changes in crossing rail or sleeper cross-sectional area influence the frequency or necessity of each maintenance activity, a general indication of which component leads to the most overall maintenance costs can be identified from Table 3, which summarises a selection of relevant maintenance activities reported in [28].

In Table 3, the costs associated with the activities ‘Manual grinding of frog (crossing rail)’, ‘Overlay welding of frog’ and ‘Replacement of frog’ have been summed to determine a cost per MGT for the crossing rail. Here the grinding and welding are related to the deterioration of the running surface of the crossing rail (wear, plastic deformation and RCF), whereas ‘Replacement of the frog’ is more vague and may reflect several different damage modes. For the sleepers, the maintenance activity considered is ‘Replacement of sleepers’. The ballast-related maintenance activities included are ‘Track geometry correction’ and ‘Ballast cleaning’.

Altogether, this results in a cost per MGT of 3800 SEK (330 EUR) for the crossing rail, 300 SEK (26 EUR) for the sleepers and 1644 (143 EUR) for the ballast. It can therefore be concluded that the majority of the cost arises from the replacement of the crossing rail and

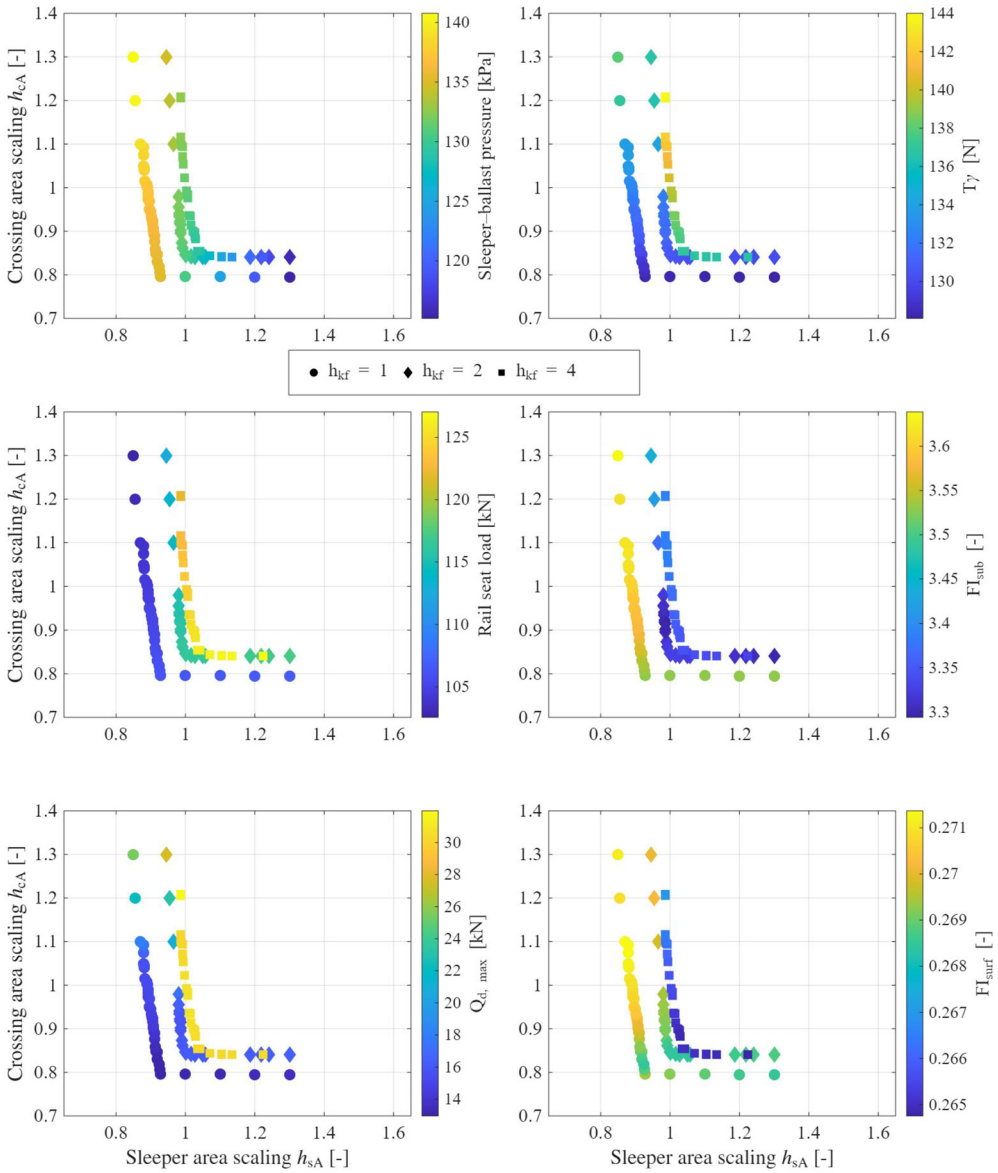


Figure 12. Effect on the deterioration metrics due to changes in component dimensions for all optimisation cases with the softer ballast stiffness level and a nominal S1002 wheel profile in the facing move. Fastening stiffness level is indicated by marker shape.

from ballast-related activities, such as ballast cleaning and track geometry correction. This provides a strong economic incentive to extend the service life of the crossing rail and to reduce ballast deterioration.

As the major cause of crossing rail replacement is material degradation following repeated welding and grinding activities, the economic weighting in this study focuses on reducing the wear, RCF and plastic deformation of the crossing rail. The metrics used for this purpose are T_{γ} , FI_{surf} , FI_{sub} and the maximum dynamic wheel–rail contact force, see

Table 3. Preventive maintenance activities and corresponding cost per accumulated traffic load. Computed based on data from [28].

Maintenance type	SEK (EUR) per MGT
Manual grinding of frog	500 (44)
Overlay welding of frog	300 (26)
Replacement of frog	3000 (260)
Total crossing rail cost	3800 (330)
Replacement of sleepers	875 (76)
Total sleeper cost	875 (76)
Track geometry correction	875 (76)
Ballast cleaning	769 (67)
Total ballast cost	1644 (143)

Figure 11. Ballast deterioration is directly linked to the magnitude of the sleeper–ballast contact pressure. As shown in Figure 11, the sleeper–ballast contact pressure follows the same trend as the deterioration metrics associated with crossing rail deterioration (except FI_{sub} , which has a negligible effect). In fact, all selected metrics, except the rail seat load, favour a design with the smallest possible crossing rail cross-sectional area.

6. Conclusions and recommendations for future design directions

In this paper, the potential of structural optimisation and the associated trade-offs between key design parameters of the railway crossing panel have been studied. A rigorously validated dynamic vehicle–track interaction model was employed to evaluate the structural loading within the panel. Based on these evaluations, the structural requirements for the crossing panel were established, and an optimisation problem aimed at minimising the component masses of the crossing panel was formulated for a dimensioning worst-case scenario. The resulting multi-objective optimisation problem was solved using a genetic algorithm, producing Pareto fronts for different combinations of fastening stiffness, ballast stiffness and running direction. Using three different weighting strategies, based on CO₂ emissions, deterioration and economic cost, the Pareto fronts were subsequently analysed to assess the design trade-offs.

The loading levels of the two running directions were examined, revealing higher loading for the trailing move, due to increased bending stress in the crossing rail caused by the eccentric loading.

To minimise the overall mass of the crossing panel, a low rail pad (fastening) stiffness is preferable. Reducing the rail pad stiffness decreases sleeper loading, thereby allowing the use of smaller sleepers. The reduced fastening stiffness also benefits the crossing rail, as it lowers the structural loading, likely due to a reduction in the maximum dynamic wheel–rail contact force. A stiff ballast support allows for the smallest component dimensions, which provides an incentive to stabilise the effective support stiffness to ensure proper support conditions.

The structural optimisation demonstrates a clear potential for reducing the environmental footprint of the crossing panel. For the configuration with soft rail pads and a stiff ballast support, the environmental weighting indicates that the CO₂ footprint can be reduced by 13% relative to the nominal design, while still achieving reductions in most deterioration-related metrics. However, this solution entails slight increases in the

sleeper–ballast contact pressure and rail seat load. If a specific ballast stiffness cannot be guaranteed, the maximum reduction while maintaining safety is 8% relative to the nominal design. Furthermore, if a certain level of wheel profile quality could be ensured, the reduction in overall crossing panel mass could be significantly increased. As a result, further reductions in deterioration, environmental footprint and economic cost could be achieved.

Structural optimisation also has the potential to reduce surface-related deterioration metrics, such as the maximum dynamic wheel–rail contact force, surface-initiated RCF, and wear. Traditionally, these forms of deterioration have been mitigated by adjusting the track stiffness through the use of rail pads or under sleeper pads. However, optimisation-driven adjustments to component dimensions can achieve reductions in maximum dynamic wheel–rail contact force and wear that are comparable to those obtained through stiffness modification of the elastic elements.

In terms of deterioration, similar trends are observed for both the facing and trailing moves. Since the trailing move constitutes the dimensioning case in the optimisation, due to the higher bending stress levels caused by the eccentric loading, it would be sufficient, at least for the studied crossing panel design, to focus solely on the trailing move for the purpose of structural optimisation.

Author contributions

CRedit: **Henrik Vilhelmson:** Conceptualization, Formal analysis, Methodology, Software, Visualization, Writing – original draft, Writing – review & editing; **Jens C. O. Nielsen:** Conceptualization, Funding acquisition, Project administration, Supervision, Writing – review & editing

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