Improvement of Train-Track Interaction in Turnouts by Optimising the Shape of Crossing Nose

C.Wan1, V.L. Markine1, I.Y. Shevtsov2, R.P.B.J. Dollevoet1,2
1Section of road and Railway Engineering, Faculty of Civil Engineering and Geosciences, Delft University of Technology, Stevinweg 1, 2628 CN, Delft, The Netherlands
E-mail: c.wan@tudelft.nl, v.l.markine@tudelft.nl
2Prorail, Utrecht, The Netherlands

ABSTRACT
Proper rail geometry in the crossing part is essential for reducing damages on the nose rail. A numerical optimisation approach to minimise impact damage and wear in the crossing panel by varying the nose rail shape is presented in the paper. The optimisation formulated as a weighted multi-objective problem is solved by adapting the Multipoint Approximation Method (MAM). Dynamic vehicle-turnout interaction as a function of crossing geometry is investigated using multi-body simulation method. The optimisation problem has been solved for different sets of weight coefficients. Afterwards the robustness of the optimum designs has been analysed under different vehicle-track system conditions.

1. INTRODUCTION
Turnouts are important elements of railway infrastructure that provide flexibility of the system by enabling railway vehicles to be guided from one track to another at a railway junction (Figure 1). Due to discontinuity in the rail geometry introduced in the crossing nose, turnouts experience high impact loads from passing vehicles, which makes them sensitive to various types of rail damage such as excessive wear, plastic deformations, surface cracking and crumbling, shelling, global fracture etc. Damage of the crossing nose has become a serious problem of the Dutch Railways: currently every week two crossings must be replaced urgently.

The analysis of the dynamic vehicle-turnout interaction can be found e.g. in [1]-[3], which consists of numerical simulations and validation of numerical models through field measurements. Based on the validated model in [3], improvement of turnout performance was achieved by adjusting the vertical track elastic properties of the crossing using the numerical optimisation method [4]. In [5] the numerical optimisation of the switch rail shape was performed in order to reduce the switch rail damage. Recently a lot of experimental studies have been implemented focusing on the damage of turnout due to dynamic impact. It has been found from the field measurement [6] and [7] that the dynamic behaviour of the turnout is very sensitive to the rail geometry. The parametric study of the rail geometry at turnout crossings was carried out in [8], which shows that the nose rail shape has significant effect on the dynamic behaviour of turnout crossings. In this study performance of crossings is improved by optimising the shape of the crossing nose. The study, which follows the research presented in [6]-[8], is based on the numerical and experimental analysis of the dynamic behaviour of the train-turnout system by the means of numerical optimisation.

2. THE EFFECT OF CROSSING GEOMETRY
To investigate the effect of crossing geometry, a set of measurements has been performed on an instrumented turnout before and after its grinding maintenance. The dynamic interaction between vehicles and the turnout was analysed experimentally using the instrumented crossing (Figure 2). The measured data comprising of (among others) the 3-D accelerations of the crossing nose and locations of the maximum wheel forces on the crossing nose (Figure 2) were collected on several turnouts. The measured data in [6] and [7] confirmed the numerical
results presented in [3] that the crossing nose geometry has significant influence on the dynamic response in the
crossing area, which is also shown in Figure 3.

![Figure 2](image)

**Figure 2** Instrumented crossing nose, acceleration and fatigue area measurement data

It can be observed that the results before the grinding maintenance are considerably different after the
grinding the crossing rail, which means the geometry of the crossing has significant effect on the vehicle
dynamic behaviour. Based on such a histogram the most probable area for fatigue damage (fatigue area) on the
crossing nose can be determined, that is, 0.50-0.60 m before the grinding while 0.40-0.50 and 0.56-0.60 m after
the grinding. Moreover, the amplitudes of the dynamic impacts have been reduced significantly after grinding [7].

![Figure 3](image)

**Figure 3** Distribution of maximum wheel forces along the crossing nose: (a) before grinding, (b) after grinding

Additionally the effect of crossing geometry on the dynamic response of the crossing has been studied in [8],
wherein substantial improvement of turnout behaviour was obtained by using wider nose rail at 300 mm from
the nose point till the end of the nose rail. The above observations indicate the importance of the crossing
geometry and inspire further researches to improve the crossing geometry with the aim of reducing the impact
damage.

## 3. MODELLING OF VEHICLE-TURNOUT INTERACTION

For a complex system such as the crossing with changing geometry and structure it is an ambitious task to
develop realistic numerical models. The simulations should consider all the major influences such as the
dynamic impact and the slip between the wheel and the crossing rail, while still being able to calculate the
resulting impact force and stress on the crossing nose. Since the behaviour of a crossing is sensitive to both the
track features and the vehicle suspensions, it is necessary to take into account the dynamic behaviour of both the
track and a vehicle. The commercial software VI-Rail that is developed as a specialised environment for railway
virtual prototyping based on the industry standard multi-body dynamics code MSC Adams [9] has been used to
simulate the dynamic vehicle-track interaction at the crossing. The additional computational cost of performing
full multi-body dynamics simulations is not large. This model has been validated using the measured geometry
and acceleration measurement data of the turnout before grinding as mentioned in section 2. The local contact
geometry, contact forces, energy dissipation and displacements as a function of the position along the turnout
during the passage of each wheel are outputs of the simulations.

A passenger wagon with double wheelsets in the front and the rear of the car body is modelled based on the
Manchester passenger train benchmark [10]. The primary suspension system of which was slightly modified in
order to correspond to the passenger train operating on the Dutch railway network, detail parameters of the
vehicle can be found in [8].

The studied turnout is a standard (right turn) design with nominal rail profile 54E1, curve radius 725 m and
crossing angle 1:15. The main-facing direction (straight track) of the turnout with the total length of 150 m has
been considered in the model, where the crossing part has been simplified as 2.0 m in length. At the crossing part
the guiding rail with the length of 2 m along the left rail was modelled, while the continuously changing rail
profiles have been used at the right track. In total, 74 rail profiles are used to represent the complex crossing
geometry. At the crossing rail side, an allowable deviation in the rail geometry over 10 m has been chosen,
which is equal to 8 mm.

<table>
<thead>
<tr>
<th>Track components</th>
<th>Stiffness [MN/m]</th>
<th>Damping [kNsr/m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail pad</td>
<td>1420</td>
<td>34</td>
</tr>
</tbody>
</table>

2
The turnout is modelled as the ‘moving track’ model (Figure 4). Each track model consists of alternative rigid and elastic layers representing the two rails, the rail pads, the sleeper and the ballast. The elasticity of the track could be defined by adjusting the properties of rail pads and ballast, which are represented as the flexible connection of sleepers/rail and sleeper/ground by linear spring-damper elements in the vertical, lateral and roll directions. Here relatively stiff rail pads and the ballast with good property have been used as listed in Table 1. The elastic properties of the track model are taken as independent of the position along the switch. The reason for using a simple track representation is that the previous study indicates that a linearized track model is sufficient for qualitative analysis of the dynamic impact as compared with the complex track models (FlexTrack) in which the track is defined as FE model integrated with MSC.Nastran. Moreover, the simple model significantly reduces computation cost which is important in an optimisation problem that requires a large number of simulations.

![Schematic of the flexible track structure (ballasted)](image)

Table 1 Track properties used in the simulations

<table>
<thead>
<tr>
<th></th>
<th>Lateral</th>
<th>Roll</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ballast</td>
<td>280</td>
<td>390</td>
</tr>
<tr>
<td>Vertical</td>
<td>120</td>
<td>48</td>
</tr>
<tr>
<td>Lateral</td>
<td>120</td>
<td>40</td>
</tr>
<tr>
<td>Roll</td>
<td>130</td>
<td>290</td>
</tr>
</tbody>
</table>

The turnout is modelled as the ‘moving track’ model (Figure 4). Each track model consists of alternative rigid and elastic layers representing the two rails, the rail pads, the sleeper and the ballast. The elasticity of the track could be defined by adjusting the properties of rail pads and ballast, which are represented as the flexible connection of sleepers/rail and sleeper/ground by linear spring-damper elements in the vertical, lateral and roll directions. Here relatively stiff rail pads and the ballast with good property have been used as listed in Table 1. The elastic properties of the track model are taken as independent of the position along the switch. The reason for using a simple track representation is that the previous study indicates that a linearized track model is sufficient for qualitative analysis of the dynamic impact as compared with the complex track models (FlexTrack) in which the track is defined as FE model integrated with MSC.Nastran. Moreover, the simple model significantly reduces computation cost which is important in an optimisation problem that requires a large number of simulations.

The General Contact Element (WRGEN)[9] using actual wheel and rail profile to calculate the actual contact kinematics at each simulation step is used in the simulations. WRGEN evaluates the local contact stiffness based on geometry and materials properties. Moreover, it considers the effect of nonlinear contact patches and evaluates the contact line taking into account the effect of the wheel/rail angle of attack (pseudo 3D contact). There is no restriction of the number of contact patches in one interconnection.

The normal force in the contact area is computed using the Hertz theory for a given ellipse with undeformed penetration. While in the tangential directions the computation is based on the modification of Kalker's FASTSIM algorithm developed at TU-Berlin by Dr. Knothe[11]. The friction coefficient between the rail and wheel surface is assumed to be 0.35. Simulations are for the train travelling with speed of 140 km/h at the facing direction of the through route. The output from the simulations of vehicle-track dynamics was recorded with a sampling frequency of 2 kHz. All results were then filtered with a low pass filter with cut-off frequency 250 Hz before calculating damage criteria.

4. PARAMETERISATION OF THE CROSSING GEOMETRY
To optimise the shape of the crossing nose its geometry must be parameterised first. In the previous work[7] the effect of rail geometry at the crossing panel has been studied, wherein the geometry of the crossing nose has been parameterised by defining several key cross-sections located at certain distances from the beginning of the crossing nose.

The geometry of the cross-section could be parameterised either using the process in [8] (during manufacturing) or the grinding process (during maintenance).

According to the first approach the geometry is defined by four cross-sections (A-D) located on certain distances from the beginning of the crossing nose as shown in Figure 5. The cross-sections A (nose point) and D (1.05 m from the nose point) remain the same as the reference, which are considered as the beginning and the end of the nose rail respectively. Cross-section D has the same width (measured at 14 mm below the rail top) and the same height as the normal rail outside the crossing. Cross-sections B and C locate at 150 mm and 300 mm from the nose point, respectively, where the rail head from its top to 14 mm below the top of normal rail are assumed to be semi-ellipse with semi-axis a and b, (i=1,2). The rail profiles are generated by longitudinally interpolation of the controlling cross-sections as shown in Figure 6(a). The nominal track gauge is assumed to be constant (1435 mm).
The second approach is used for optimisation of the existing turnouts as guidance of grinding maintenance of turnout crossings. In the grinding maintenance the shape of the crossing nose after grinding mainly depends on the welder’s experience. The location and the shape of the cross-sections controlled by the welder during the grinding process differ from the ones used during the manufacturing process of the crossing nose. An additional cross-section E (figure 6 (b)) could be introduced depending on the main damage position at the crossings. For this approach, the controlling cross-sections A and B are the measured profiles of the crossing and they will be fixed during the optimisation. Cross-sections C and D could keep their shape if not damaged or using the reference profiles as in the manufacturing. That is, only the controlling cross-section E needs to be optimised, which is represented as a spline with a series of flexible controlling points.

The rail profiles of the wing rails are fixed, that is to say, neither the shape nor the location of the wing rails with respect to the track centre line would change during the optimisation. This indicates that by introducing the variation of the controlling cross-sections of the nose rail, the track gauge (T) as well as the flangeway (F) will be varied along the crossing as a function of the location (Figure 7). The variation of T and F is equal at both the through route and the divergent route. For the controlling sections B and C the track gauge and flangeway are:

\[ T_i = T_{ref} + a_{ref} - a_i, \quad i = 1, 2 \]

\[ F_i = F_{ref} + a_{ref} - a_i, \quad i = 1, 2 \]

Here \( T_{ref} \) and \( F_{ref} \) are the reference (standard) track gauge and flangeway, which are taken as 1435 mm and 43 mm, respectively. \( a_{ref} \) is the reference half-axle of the controlling cross-sections (Figure 5). With the defined

5. OPTIMISATION PROBLEM

5.1 Design variables

The present paper will focus on optimisation of new designs of crossings using the first approach. In order to investigate the possibility of improving the nose rail shape a relatively wide range of the design variable variation is considered with the requirement of the flangeway larger than the wheel flange width (32 mm).

The design variables are:

\[ X = [a_1, b_1, a_2, b_2], \]

With the following side limits:

\[ 2 \leq a_1 \leq 13 \text{ (mm)}; \]
\[ 2 \leq b_1 \leq 14 \text{ (mm)}; \]
\[ 2 \leq a_2 \leq 18 \text{ (mm)}; \]
\[ 2 \leq b_2 \leq 14 \text{ (mm)}. \]
range of the design variables the flangeway larger than 35 mm is guaranteed.

![Figure 7 Variation of track geometry at a certain location of the crossing panel (example of the through route).](image)

According to the given space of design variables the track gauge is allowed to vary from the nose point (cross-section A) to the end of the nose rail (cross-section D) with the amplitude between -8 and 8 mm at each travelling route.

A turnout design with \( a_1 = b_1 = 4 \text{ mm} \), \( a_2 = b_2 = 9 \text{ mm} \), which is from the current manufacturing process in the Dutch Railway, is used during the optimisation as the reference design.

### 5.2 Objective function

The optimisation problem focuses on reducing the most common damages on the crossings such as RCF due to dynamic impact and wear. Therefore the normal contact pressure and energy dissipation during the vehicle passage through the crossing are to be minimised. The contact pressure \( (S) \) is formulated in the equation (4) while the energy dissipation \( (W) \) is estimated using the wear index \( W \) calculated according to Kalker[13].

\[
S(X,t) = \frac{3}{2} \cdot F_s(X,t) \cdot A(X,t),
\]

\[
W(X,t) = F_s(X,t) \cdot \xi + F_s(X,t) \cdot \eta.
\]

The objective function is formulated as:

\[
F_n(X) = w_1 \frac{\bar{S}(X)}{S^*} + w_2 \frac{\bar{W}(X)}{W^*} \rightarrow \min
\]

where \( \bar{S} \) and \( \bar{W} \) are the accumulative contact pressure and energy dissipation of the designed turnout, which are expressed in the form of the Kresselmeier-Steinhauser function (KS function) [14]:

\[
\bar{S}(X) = \frac{1}{\mu} \ln \left[ \sum e^{\mu S(X,t)} \right],
\]

\[
\bar{W}(X) = \frac{1}{\mu} \ln \left[ \sum e^{\mu W(X,t)} \right].
\]

Here the parameter \( \mu \) determines the discrepancy between \( \bar{S} \) (or \( \bar{W} \)) and the most critical value of \( S^\text{max} \) (or \( W^\text{max} \)), details about the KS function could be found in[14]. \( S^* \) and \( W^* \) are corresponding values from the reference turnout used as the normalised factors. \( w_1 \) and \( w_2 \) are the weight coefficients \((w_1 = w_2 = 1)\). The reason of using accumulative value instead of maximum contact pressure and energy dissipation in the objective function is that, the location of the maximum amplitude of dynamic response (dynamic impact) changes with the variation of the design parameters, which is difficult for the optimisation. While the accumulative value accounts for response of every time step and gives a penalty of the maximum value during the whole simulation, which will smooth the objective function and makes the optimisation much easier.

During the optimisation \( S \) and \( W \) are checked only for the crossing rail due to the fact that the dominant impact occurs at the crossing when the wheel-rail contact transfers from the wing rail to the nose rail.

### 5.3 Constraints

Several constraints are used in the optimization. Design parameters should follow the restrictions \( a_1 \leq a_2 \) and \( b_1 \leq b_2 \) to avoid dimples both at the lateral and vertical directions along the crossing nose. Also a limit for the ratio between lateral and vertical forces \( Y/Q \) is imposed in the constraints to take into account the risk of derailment due to the allowed big amplitude of track gauge variation as discussed in section 3. The \( Y/Q \) ratios on both sides of the track were calculated after low-pass filtering the \( Y \) and \( Q \) results with cut-off frequency 20 Hz \((f = v/l)\), with \( v = 140 \text{ km/h} \) and \( l = 2 \text{ m} \). In general, large steady-state lateral forces are accompanied by large vertical loads tending to keep the \( Y/Q \) ratios below critical levels. However, large \( Y/Q \) ratios can occur transiently as a result of sudden dynamic reductions in vertical loading. Derailment can occur if \( Y/Q \) ratio over a distance of more than 2 m is greater than the critical value.

Additionally, the impact on the nose is not allowed to occur when the nose rail is not thick enough to resist the high dynamic force. Based on the observation from the field site the minimum thickness to carry the dynamic force is taken as 20 mm, although it is preferably to use a constraint on the allowed lateral and vertical loading on
the nose rail as a function of the nominal thickness if the related information is available. Moreover, to aim for a better geometry design the accumulative normal contact force should not be larger than that of the reference turnout.

6. RESULTS
6.1 Results of the optimisations
The optimisation problem is solved using the Multipoint Approximation Method (MAM)[12] for different sets of weight coefficients \([w_1, w_2]\) (Figure 8), allowing for flexible choice of preferable behaviour of turnout designs.

![Figure 8] Optimisations with different weight coefficients: Normalised wear index \((W)\) w.r.t normalised normal contact pressure \((S)\). The weight coefficient pair is shown next to the corresponding point.

The dynamic responses of the optimisations with weight coefficients as shown in Figure 8 are presented in Figure 9. These results show that the performance of the crossing has been significantly improved, although the wear index on the stock rail has increased. Moreover, in Figure 10 other dynamic responses of the optimised turnout designs are presented. From these results it can be observed that by optimising the nose rail shape the dynamic forces on the crossing have been considerably reduced while an increase of the vertical contact force was obtained on the stock rail. Since the stock rail is relatively strong in resisting the dynamic loading as compared with the nose rail, damage is less likely to occur. As a whole, the dynamic performance of the crossing panel has been improved after optimising the nose rail shape.

![Figure 9] Dynamic responses checked during the optimisation with respect to the position along the crossing nose (results are for the passage of the leading wheelset): (a) normal contact pressure, (b) normal contact force, (c) wear index at the crossing, (d) wear index at the stock rail
The design variables of the optimum designs are shown in Figure 11, which are quite different from the reference design. The optimisation with the minimum combined-objective function among all the optimisations, which is called as **opt-min**, is found with the weight coefficients [0.3, 0.7]. The height difference between the cross-sections B and C (b₁ and b₂) are reduced to less than 1 mm (both are round 10 to 11 mm), which results in the reduction of the impact forces. The width of the nose rail (2a₁) is increased from 8 mm to around 12 mm at the cross-section B while for the cross-section C the width is reduced or increased, depends on the other sets of design variables.

**Figure 11 The design variables corresponding to the optimum designs**

### 6.2 Robustness analysis of the optimum design

Robustness analysis was performed by varying the vehicle-track system, such as introducing vertical and lateral disturbance in the wheelset movement just before the wheel entering the crossing, widening and narrowing the track gauge, and so on. Totally five different cases including the system condition used in the optimisation are checked for each optimum design by comparing the dynamic response between the optimum design and the reference turnout under the five cases. The lateral displacements of the wheelsets are all less than 5.5 mm and well damped out. Figure 12 shows the dynamic impact of the design with average geometry among the all the optimum designs, which is obtained with the weight coefficients [0.6, 0.4]. The results indicate that the optimum design is reliable under various vehicle-track system conditions.
7. CONCLUSIONS
A numerical optimisation approach for reducing the damage on the crossing nose by adjusting the nose rail shape is proposed. The optimisation problem is formulated as a weighted multi-objective problem, which has been solved with different sets of weight coefficients. The dynamic behaviour of the crossing has been significantly improved after optimisations. The height of the nose rail at the beginning is considerably increased, which ultimately resulted in the reduction of the impact forces. The width of the nose rail in the beginning is also increased.

Additionally, the robustness of the optimal design with average design variables among all the optimum designs has been analysed by checking the dynamic response under different track conditions and initial disturbance in the vehicle-track system, which indicates that the new nose rail design is reliable.

Further investigations of the optimum designs will be performed using more realistic models such as FE FlexTrack model in VI-Rail and FEM model for detailed analysis of the damages on nose rail. Moreover, the optimum shape of the crossing nose will be implemented in one of the turnouts in the Dutch railway network. Its performance will be assessed using the instrumented crossing device.

REFERENCES