Research Article

Nonlinear Finite Element Modelling of Railway Turnout System considering Bearer/Sleeper-Ballast Interaction

James Sae Siew,1,2 Olivia Mirza,2 and Sakdirat Kaewunruen3

1JKW Engineering, Penrith, NSW 2057, Australia
2School of Civil Engineering, University of Western Sydney, Penrith, NSW 2057, Australia
3Birmingham Centre for Railway Research and Education, School of Civil Engineering, The University of Birmingham, Edgbaston, Birmingham B15 2TT, UK

Correspondence should be addressed to Sakdirat Kaewunruen; sakdirat@hotmail.com

Received 11 September 2014; Accepted 17 February 2015

Academic Editor: Lucio Nobile

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Rail turnouts are built to enable flexibility in the rail network as they allow for vehicles to switch between various tracks, therefore maximizing the utilisation of existing rail infrastructure. In general, railway turnouts are a safety-critical and expensive feature to a rail system as they suffer aggressive operational loads, in comparison to a plain rail track, and thus require frequent monitoring and maintenance. In practice, great consideration is given to the dynamic interaction between the turnout components as a failed component may have adverse effects on the performance of neighbouring components. This paper presents a nonlinear 3D finite element (FE) model, taking into account the nonlinearities of materials, in order to evaluate the interaction and behaviour of turnout components. Using ABAQUS, the finite element model was developed to simulate standard concrete bearers with 60 kg/m rail and with a tangential turnout radius of 250 m. The turnout structure is supported by a ballast layer, which is represented by a nonlinearly deformable tensionless solid. The numerical studies firstly demonstrate the importance of load transfer mechanisms in the failure modes of the turnout components. The outcome will lead to a better design and maintenance of railway turnouts, improving public safety and operational reliability.

1. Introduction

Rail operators are considerably demanded by the public and other stakeholders to be more efficient than ever. As a result, the maximisation of utilisation and flexibility of rail network is one of the key strategies in rail asset management. A railway turnout is a critical part of the railway where tracks cross over one another at an angle to divert a train from the original track. It allows for train vehicles to cross over or switch between various tracks and in turn maximising the utility of tracks and assets. Its main components include rails, switches, crossings, steel plates, bearers, ballast, and subgrade (as shown in Figure 1). The rail turnouts are an essential part of a rail system but, at the same time, they are a costly feature to a rail system as they suffer adverse operational loads, in comparison to a plain rail track and require frequent maintenance. Due to the particular geometry of wheel-rail contact and sudden variation of track flexibility, severe impact loads may occur during train passage over the turnout. Turnout components are subjected to general wear, rolling contact fatigue, and accumulated irreversible (plastic) deformations [1, 2].

Railway track structures experience static, dynamic, and often impact loading conditions due to wheel/rail interactions associated with the abnormalities in either a wheel or a rail over their life cycle [3, 4]. Specially at turnouts crossing, the wheel-rail interaction at the transfer zone often causes detrimental impact forces and excessive dynamic actions [5–7]. Many studies showed that it is very likely that railroad turnout bearers or crossties could be subjected to severe impact loads, resulting in a rapid deterioration in terms of structural integrity and durability, track settlement, and ride comfort [8–10]. Traditional turnouts generally impart high impact forces on to structural members because of their blunt geometry and the gaps between mechanical connections between closure rails and switch rails (i.e., heel-block joints).
Although a new smoother method of geometrical analysis and tangential design to improve wheel/rail navigations has been adopted for turnouts, the transfer zone at a crossing nose in any complex turnout system still generates high-intensity impact forces to turnout components. Generally, the turnout bearers for supporting points and crossing structures were designed using the beam on elastic foundation analysis or 2D FE grillage method [11,12]. Kaewunruen [13,14] indicated from recent work that some additional factors were often neglected from the grillage analyses, although they must be taken into account in the design process, including:

(i) extra length of turnout bearers in comparison with standard sleepers,
(ii) centrifugal forces through curved pairs of rails,
(iii) forces and bending moments induced from points motors and other signaling equipment,
(iv) impact forces induced by wheel-rail interaction,
(v) mechanical rail joints.

On this ground, this numerical study was initiated by a recent number of reportedly broken concrete bearers on a mixed-traffic line in New South Wales (NSW), Australia. Due to the complexity of the loadings and damage modes in railway turnouts, this study aims to establish a three-dimensional (3D) finite-element (FE) model. The 3D FE model adopts an elastoplastic region of bending and shear deformation of materials. The 3D FE model was developed based upon a common tangential turnout used in Australia. This study indicates that the crossing panel is where turnout bearers experience the greatest bending moment and shear force and it provides the critical force envelopes for design improvement of turnout bearers.

2. Finite Element (FE) Modelling

Railway turnout systems have generally been analysed using a grillage beam method (Manalo et al., 2010) [12,15]. Although the simplification is useful, such a method cannot adequately assist in the failure analyses of turnout components. In some cases, the results using the grillage beam method seem to have discrepancies with the field observations where the maximum bending and shear forces were evident within the crossing panel [16–19]. A number of researches have been conducted to locate the critical section within a turnout, and many of which conclude that the critical section is located specifically at the crossing panel [20–23].

This paper presents a nonlinear 3D FE analysis using ABAQUS considering the whole turnout, which comprises bearers, rail, guard rails, crossing nose, rail pads, baseplates, and guardrail support plates. The benefits of modelling in 3D space are to incorporate the effects of the neighbouring bearers and to take into consideration the longitudinal forces of the continual rail. The boundary conditions of the central 3D model can be simulated enabling vibrations to radiate beyond the model [24].


In general, the surface conditions of the wheel and rail play a critical role in the W-R contact force, in addition to geometric irregularities, train speeds, and type of track structure. The large contact force will accelerate the deterioration rate of...
Sun et al. [25] provided an insight into the potential sites for impact and fatigue damage as the train wheel traverses through the nose of the crossing. Firstly, the wing rail fatigue damage is caused by contact from the far side of wheel. Secondly, the transition of the wheel between the wing rail and nose causes a dipping movement. This is due to the tracking on the wing rail to an upward motion on the ramp of the nose resulting in fatigue damage. Greater contact stress can be seen due to the acute contact area in the crossing nose. The British Railways Board [26]
expressed that the permissible track forces in a low frequency range (or called $P_2$ forces) for railway vehicles negotiating a discontinuity in rail profile do not exceed 322 kN whilst operating at their maximum design speed. In Australia, the $P_2$ force is limited to 290 kN for locomotives on very good track structure or otherwise must be less than 230 kN. The $P_2$ force is calculated using the following formula:

$$P_2 = Q + (A_z \cdot V_m \cdot M \cdot C \cdot K),$$  \hspace{1cm} (1)

where

$$M = \left[ \frac{M_v}{M_v + M_z} \right]^{0.5},$$

$$C = \left[ \frac{\pi \cdot C_z}{4 [K_z (M_v + M_z)]^{0.5}} \right],$$  \hspace{1cm} (2)

$$K = (K_z \cdot M_v)^{0.5},$$

the lesser of

$$Q = 0.13D \times 10^3$$

or $$Q = 125 \times 10^3,$$  \hspace{1cm} (3)

where $D$ is the wheel diameter (mm), $Q$ is the maximum static wheel load (N), $V_m$ is the maximum normal operating speed (m/s), $M_v$ is the effective vertical unsprung mass per wheel (kg), $A_z$ is total dip angle of vertical ramp discontinuity (taken as 0.02 rad), $M_z$ is taken to be 245 kg as the effective vertical rail mass per wheel, $C_z$ is taken to be $55.4 \times 10^3$ N/m as the effective vertical rail damping rate per wheel, and $K_z$ is taken to be as $62 \times 10^6$ N/m as the effective vertical rail stiffness per wheel.

In addition, lateral resistance is usually designed to reinforce the structural integrity of the rail and turnout. Considerations are only given to lateral force values sustained for distances of 2 metres or more. Unless supported by appropriate technical justification, vehicles attempting to negotiate a lateral ramp discontinuity in track alignment, when travelling on a curve at maximum normal operating speed.
speed and at maximum cant deficiency, without exceeding a total lateral force level per axles of 71 kN, should introduce lateral action that can be calculated using the following formula [26]:

\[
Y = W \cdot A_d + A_y \cdot V_m \left( \frac{M_u}{M_u + M_y} \right)^{0.5} \cdot \left[ K_y \cdot M_u \right]^{0.5},
\]

(4)

where \( Y \) is the lateral force per axle (N), \( W \) is the static axle load (N), \( A_d \) is the maximum normal operating cant deficiency angle (rad), \( V_m \) is the maximum normal operating speed (m/s), \( M_u \) is the effective lateral unsprung mass per axle (kg), \( A_y \) is taken to be 0.0038 rad which is the angle of lateral ramp discontinuity, \( M_y \) is taken as 170 kg and is the effective lateral rail mass per wheel, and \( K_y \) is taken to be \( 25 \times 10^6 \) N/m as the effective lateral rail stiffness per wheel. Based on these formulas, the dynamic forces can be estimated for the design of turnout components.

2.2. Turnout Components. The FE model comprises entirely 3D deformable solids, straight and curved rail, bearers of varying length, and a ballast layer as the track support. This study focuses on the behaviour of the bearer and ballast; therefore, a suitably accurate rail seat load within a tangential configuration is required for the analysis. Steel rails were modelled in 3D space to account for their cross-sectional properties, the width of the contact patch between the wheel and rail, the width of the rail web, and the width of the rail footing. The rail and switch rail profiles were validated against rail authority’s specifications [27–29]. Table 1 shows the engineering properties of turnout structural components. Concrete bearers have been modelled as rectangular blocks with dimensions nominated according to the specifications varying lengths between 2.5 m to 7.5 m according to the turnout design.

The elastic modulus of steel rails and crossing is defined by the initial slope of the stress-strain relationship to the extent of the upper yield threshold, as illustrated in Figure 2. For concrete material, it is assumed that its initial compressive stress behaviour is to be linear given that it does not exceed \( 0.4 f'_c \) [30]. Beyond the linear threshold, nonlinear stress is expressed as a function of strain according to (5). A graphical representation of the stress-strain relationship of concrete is depicted in Figure 3. Consider

\[
f'_{c}.
\]

(5)
Table 1: Design properties of materials.

<table>
<thead>
<tr>
<th>Materials</th>
<th>Elastic modulus (MPa)</th>
<th>Compressive strength (MPa)</th>
<th>Tensile strength (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Concrete</td>
<td>38,000</td>
<td>36–55</td>
<td>4.0–6.30</td>
</tr>
<tr>
<td>Prestressing tendon</td>
<td>200,000</td>
<td>—</td>
<td>1,700</td>
</tr>
<tr>
<td>Steel rails</td>
<td>205,000</td>
<td>—</td>
<td>—</td>
</tr>
</tbody>
</table>

2.4. Load Conditions. Loading configuration is in accordance with Standards Australia [33]. The design loads are depicted in Figure 4(a), which simulate the worst case loading configuration that can be exerted onto a rail track. The FE model predicts the behaviour of the turnout by considering multiwheel impacts, which would simulate in-service and cyclic loading and have been adapted as a set of concentrated loads negotiating the turnout to represent a moving coupled locomotive (in order to form the worst case scenario for the design). The coupled locomotive is then simulated with four 300 kN axle loads and a single 360 kN axle load 2 meters ahead of the group.

The above load set is applied to the model at 600 mm increment initially to coincide with turnout bearer spacing, or referred to hereafter as load sets. A total of 48 load steps (including model initiation) are modelled to generate the overall movement of the locomotive negotiating the turnout. When approaching the crossings, the load step is later set at 50 mm increment. Figure 4(b) illustrates loading configurations for particular steps.

2.5. Validation. The numerical model was previously calibrated using the field measurement data [13, 14, 34, 35]. Along with an accurate resultant deflection, the time required to compute the analysis is also significant in selecting an optimum mesh size. The typical aggregate size of ballast in practice is anywhere between 13 mm and 65 mm. An initial analysis is carried out to determine the maximum
3. Results and Discussion

In practice, frequent maintenance of supporting bearers and fastening systems can often be observed in the field even though those components have been designed in accordance with engineering standards. This study has therefore investigated such an important issue. It is found that the bearers, which undergo the greatest deflection of a coupled locomotive pass, are the bearers underneath the crossing nose (maximum at bearer number 47). The sensitivity analysis illustrates the maximum deflection in all bearers with the passing of a moving couple train load, 300LA [33]. From the sensitivity analysis, it can be seen from Figure 7 that the bearer directly underneath the crossing (number 47) experiences the greatest deflection, with a resultant of 2.54 mm. The sharp spike in deflection clearly defines the moment at which each wheel axle impacts the above rail, in this case the crossing nose. Any lateral sliding due to such vertical load action cannot be observed.

The shear stress response of crossing bearer (number 47) at the most critical loading is illustrated Figure 8. It can be witnessed that the large spike at the midpoint is obviously the point at which the wheel impacts the crossing nose. Note that this shear response has not been considered in the component design since it is often lower than the shear capacity of prestressed concrete bearers. Figure 9 represents both the bending stresses and moment envelopes given in the design specification [28, 29] and the bending moment obtained from the FE analysis by using the methodology recommended in the specification. Four reference points are chosen according to the given bending moment envelope, which is gathered from these reference points 0 m, 0.5 m and total length minus 0.5 m and at the end span. Figure 9(a) illustrates the bending moment envelope for 2.6 to 2.8 m long turnout bearers. The black line signifies the bending moment envelope which is found in the design specification [29] and the blue line represents the shear envelope obtained from the FE analysis, specifically bearer 21 (the bearer under the closure rails). It can be seen that the computed results do not correlate well with the design allowable actions. However, this may be due to the nature of the loading of the turnout, as the train in this particular instance is often modelled to travel along the diverging, tangential path and the loading should not be expected to be linear or symmetrical, unlike what is depicted in the design specification. Figure 9(b) demonstrates the comparison of a chosen bearer with the suggested design envelope. The crossing bearer (critical bearer number 47) is chosen to validate the data sets and results. As with the former example, the black line represents the bending moment envelope specified in the design standards [28, 29]. The bending moment envelope is intended to be used for turnout bearers between the lengths 2.8 m and 5.2 m. Again,
(a) Bearer 21 (red) in relation to load step 18 (yellow)

(b) Bending stress of bearer 21 under load step 18

(c) Bending moment envelope of bearer 21 with computed values (blue) against specifications (black)

(d) Bearer 47 (red) in relation to load step 36 (yellow)

Figure 9: Continued.
Due to the unchanging response of the modelled result and the fact that the bearers under the crossings experience critical loading, it is believed that the moment envelope overlooks the adverse loading configurations of a turnout system and, instead, idealises the moment response to that of straight rail. Figures 10 and 11 display the comparative bending moment envelopes for the bearer at crossing (number 47) and the bearer at closure rail (number 21). Retaining the same convention as earlier envelopes, the blue line signifies the FE analysis. There is an obvious spike approximately at the midpoint of the bearer span. It is logical to assume that the impact of the wheel onto the crossing nose will greatly influence the shear and bending envelope. Although there may be a huge deviation between the specified value and the simulated result, it should be noted that the envelope within SPC 233 [29] shows no deviation nor increased loading within that particular location whatsoever.
It is concerning the fact that the maximum bending moment simulated, being 322 kNm, is greater than that specified by about 800%. Also that no change in loading condition or material property has been changed is noteworthy; yet, by involving a greater number of points in which to create an accurate stress diagram, the loading may be allowed to deviate so greatly. As the increase in bending moment is concentrated within the midspan, it can be deduced that the midspan could become more susceptible to permanent deformations and cracking that was not designed to not be designed to accommodate for the adverse loading.

The resultant bending diagram depicted in Figure II shows the much more comprehensive bending envelope in comparison to those given within specifications. The envelope given in the specifications depicts a much more linear approach to bending moment design, and it is speculated that the diverse geometry of a rail turnout has been overlooked altogether. The significance of the bending moment envelope obtained for the bearer under closure rail (number 21) is that not only does the larger bending moment occur at the point of wheel impact, but it also has the characteristic of uneven loading, which is precisely when the train is diverting into the tangential track.

4. Conclusions

This paper firstly presents a development of three-dimensional finite element model of a tangential turnout system for an investigation into the failure modes that were arisen from the field observations and measurements on a mixed traffic rail line whereas broken concrete bearers and loose fasteners were reported routinely. A 3D FE model has been established and validated for the analysis of a complete turnout system. The primary objective of this study is to determine the critical location, be able to realise the critical deflection, and validate shear force and bending moment envelopes of a turnout system. To address this, ABAQUS has been employed to carry out all modelling and postprocessing of a complete 3D turnout. From the detailed analysis, turnout bearers right underneath crossing panel experience the highest load actions, resulting in the largest deformations. It is also found that the turnout sleeper or bearer underneath the closure rails or where there is a change in rail curvature is subjected to a high level of vertical loading, sometimes exceeding its design load limits. These results are of significant importance to rail engineers and track designers, in order to establish a safer and more reliable turnout system. Future work will evaluate the effects of ballast voids and pockets on the dynamic behaviour and lateral sliding of turnout systems.

Acknowledgments

The authors are grateful to University of Western Sydney and RailCorp for the support throughout this study. Also, the last author wishes to thank Australian Government for his Endeavour Executive Fellowships at Massachusetts Institute of Technology, Harvard University, and Chalmers University of Technology.

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