Nonlinear 3D finite element modeling for structural failure analysis of concrete sleepers/bearers at an urban turnout diamond

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CHAPTER

Nonlinear 3D finite-element modeling for structural failure analysis of concrete sleepers/bearers at an urban turnout diamond

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INTRODUCTION

Railway urban turnout is a special track system used to divert a train from a particular direction or a particular track onto other directions or other tracks. It is a structural grillage system that consists of steel rails, points (or called “switches”), crossings, steel plates, rubber pads, insulators, fasteners, screw spikes, beam bearers (either timber, polymer, steel, or concrete), ballast and formation, as shown in Figure 6.1 [1]. There are two types of turnouts, a conventional turnout and a tangential turnout.

![Components of a turnout.](image)
Standard conventional turnouts are designed typically for straight main line track. The combination of switch length, heel angle, and cross rate defines the turnout type, and they all typically have the same components. Tangential turnouts are defined by the radius of the turnout. Components in a tangential turnout vary as manufacturers place their own designs over the standard configuration. The traditional turnout structure generally imparts high-impact forces on to its structural members because of its blunt geometry and mechanical connections between closure rails and switch rails (i.e., heel-block joints) [2].

A turnout is an inevitable structure in railway tracks whose crossing imparts a significant discontinuity in the rail running surface. The wheel/rail interaction on such imperfect contact transfer can cause detrimental impact loads on railway track and its components [1–9]. The transient vibration could also affect surrounding building structures. In addition, the large impact emits disturbing noises to railway neighbors [5,10–13]. The impact and ground-borne noises are additional to the normal rolling noise. Many previous studies have predicted impact forces and noise using numerical models [5]. However, only a few have implemented impact mitigation strategies in the field and even fewer field trial reports are available in the literature [5–13]. The impact mitigation strategies at an urban turnout include wheel/rail transverse profiling and longitudinal profiling of crossings, increased turnout resilience and damping, changes to rolling stocks, external noise/vibration controls, etc. [14–20].

Recently, there have been several incidental cracks in concrete bearers in a turnout within an urban railway network in Australia. The concrete bearer cracked under a rail pad, the part which supports the rail, where the train wheels shift over a diamond formation. The purpose of this chapter is to demonstrate an engineering failure analysis of a complex structural system using a case study of turnout diamond (Figure 6.2), which is often found in railway network. The finite-element model

![Figure 6.2](image_url)

**Figure 6.2**
A typical railway turnout diamond (k crossings induce double impacts on the supporting bearers).
has been developed to determine if the turnout diamond systems were sufficient for the impact loading experienced by that bearer. The emphasis of this study will be placed on the current design and analysis of turnout bearers and the application of nonlinear finite-element modeling to aid failure analysis of railway structural components.

Table 6.1 summarizes recent research on failure analysis of turnout components. It can be found that most research has paid special attention on traditional turnouts, while information on such special trackworks as diamond, single, and double slips are very limited. As a result, this study is imperative for complex turnout design, maintenance, and renewal.

2 INTERACTION OF TRAIN AND TURNOUT STRUCTURE

The wheel/rail contact over the crossing transfer zone has a dip-like shape where the wheel trajectory is not smooth. The accurate shape of the wheel trajectory (running top) and dip angle will depend on the wheel and running rail profiles. The associated dip angles, which are the acute angles between the tangents to the wheel trajectory at the point where it abruptly changes direction, can then be estimated from the wheel trajectories, as illustrated in Figure 6.3.

2.1 TURNOUT FORCES

It is generally assumed that the high-frequency impact force \( (P_1) \) that occurs either at a nose or at a wing rail has little effect on the rail foot [14]. On the other hand, the dynamic \( P_2 \) force (the second peak in the impact force history) has significant influence on the crossing components. The distance from the point of impact to the point of the peak impact force depends on a number of factors including train speed. The common damage zones to be considered are the rail foot within 0.75 m of any joints to plain rail; the base of the crossing in transfer zone (extending 0.75 m on both sides of the pick up point); and other components in vicinity of the crossings.

In a calculation of \( P_2 \) force, the track damping \( C_t \) is normally negligible. For plain tracks, it is commonly found that the track mass is relatively low in comparison with the wheel set mass and is then neglected. In contrast, for a turnout crossing, the track mass tends to be of significance and it cannot be neglected. Jenkins et al. [15] has proposed a formula for estimating a dynamic \( P_2 \) force as follows:

\[
P_2 = P_0 + 2\alpha \cdot v \cdot \left[ \frac{M_u}{M_u + M_l} \right]^{1/2} \cdot \left[ 1 - \frac{\pi \cdot C_t}{4\sqrt{K_t \cdot (M_u + M_l)}} \right] \cdot [K_t \cdot M_u]^{1/2},
\]

where \( P_2 \) is the dynamic vertical force (kN); \( P_0 \) is the vehicle static wheel load (kN); \( M_u \) is the vehicle unstrung mass per wheel (kg); \( 2\alpha \) is the total joint angle or equivalent dip angle (rad); \( v \) is the vehicle velocity (m/s); \( K_t \) is the equivalent track stiffness (MN/m); \( C_t \) is the equivalent track damping (kNs/m); and \( M_l \) is the equivalent track mass (kg).
Table 6.1 Recent Research on Failure Analysis of Special Trackworks (2013–2015)

<table>
<thead>
<tr>
<th>Year</th>
<th>Authors</th>
<th>Keywords</th>
<th>Type of Special Trackwork</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Turnout</td>
</tr>
<tr>
<td>2015</td>
<td>Kaewunruen [8]</td>
<td>Bearer failure</td>
<td>x</td>
</tr>
<tr>
<td>2015</td>
<td>Tan et al. [21]</td>
<td>Rail switch failure</td>
<td>x</td>
</tr>
<tr>
<td>2015</td>
<td>Pålsson and Nielsen [22]</td>
<td>Switch and crossing</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Camci et al. [23]</td>
<td>Motor switch failure</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Kaewunruen [24]</td>
<td>Crossing degradation</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Kaewunruen [25]</td>
<td>Composite bearer</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Ni et al. [26]</td>
<td>Cracked crossing</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Pålsson [27]</td>
<td>Switch and crossing</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Jönsson et al. [28]</td>
<td>Crossing</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Jeon [29]</td>
<td>Formation failure</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Wan et al. [30]</td>
<td>Crossing nose</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Griffin et al. [31]</td>
<td>Composite track slab</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Li et al. [32]</td>
<td>Turnout settlement</td>
<td>x</td>
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<tr>
<td>2014</td>
<td>Chen et al. [33]</td>
<td>Turnout defect</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Rapp et al. [34]</td>
<td>Shared corridor</td>
<td>x</td>
</tr>
<tr>
<td>2014</td>
<td>Ferdous and Manalo [35]</td>
<td>Bearer failure</td>
<td>x</td>
</tr>
<tr>
<td>2013</td>
<td>Pålsson [36]</td>
<td>Switch rail</td>
<td>x</td>
</tr>
<tr>
<td>2013</td>
<td>Eck et al. [37]</td>
<td>Switch rail</td>
<td>x</td>
</tr>
<tr>
<td>2013</td>
<td>Yang et al. [38]</td>
<td>Turnout parts</td>
<td>x</td>
</tr>
<tr>
<td>2013</td>
<td>Davis and Jimenez [39]</td>
<td>Turnouts</td>
<td>x</td>
</tr>
</tbody>
</table>
2.2 TRACK STIFFNESS

Track stiffness $K_t$ can be estimated using a beam on elastic foundation (BOEF) model. The deflection under a point load can be written as:

$$K_t = \frac{P}{\gamma} = 8\beta^3 EI,$$

(6.2)

$$\beta = \left[ \frac{k_f}{4EI} \right]^{1/4},$$

(6.3)

where $E$ is the Young’s modulus; $I$ is the moment of inertia of the rail or crossing section; and $k_f$ is the foundation modulus (track modulus).

$$k_f = 4.104P_0(\text{axle load in tons}) + 14.61 \text{ MPa}.$$  

(6.4)

2.3 EQUIVALENT TRACK MASS

Considering the bearer spacing of 600 mm, a bearer mass of 600 kg (half can be associated with the crossing), a mass per unit of the crossing is 120 kg/m, the track mass per unit rail length (m) can be estimated as 620 kg/m. The equivalent track mass (lumped) at crossing $M_t = 3m/2\beta$. Thus, $M_t$ is varied from 705 to 756 kg.
3 FAILURE MECHANISMS

In recent years, there have been several incidents of cracking in concrete bearers in turnouts within Australian railway networks. Such problem was in relation to significant degradation of turnout components due to large impact force such as that from the Clyde 700 diamond (RailCorp’s western main rail network). White ballast (ballast pulverization) could be observed near the crossings. The failure mechanism is illustrated in Figure 6.4.

3.1 STRUCTURAL INTEGRITY

Initially, the early warning sign was the deviated geometry profile of the diamond, as seen in Figure 6.4a. White ballast (ballast pulverization) could then be observed near v-crossings and k-crossing, as shown in Figure 6.4b and c. It is believed that the ballast was pulverized due to the impact vibration of crossing structures and the bearers excited by the rail dip-like irregularity. It was also found that due to the large vibration, the wheelsets tended to spread the lubricant (as the wheel collected from friction management trackside device) over the location slightly further from the nearby crossing (such as in the crossing panel as shown in Figure 6.4b).

Figure 6.4d also shows the collapse of the bearer underneath the stockrail at Country-end v-crossing. This negative (hogging) bending failure is due to a single wheel impact at v-crossing. The bearer failure had led to the discrepancy of track surface (top) between running rails and the wide track gauge irregularity. The shiny steel at wing rail could be noticed due to the contact of wheelback and wing rails. Similar mechanism can also be observed at k-crossing area.

It is noted that the concrete bearer cracked under a rail pad, the part which supports the rail, where the train wheels travel over the diamond section. The concrete bearers experienced a different type of dynamic load, compared to standard railway sleepers. This is due to their inherent stiffness nature of the diamond, which could be much higher than that of a plain track. When trains move over changing track stiffness and geometry, wheel-rail interaction induces high-magnitude impact forces. As a result, it is recommended that such forces be accounted for in the design standards to prevent further failure, as well as financial loss and possibly loss of life.

During such time, train speed limit restrictions were put in place around the cracked bearer to prevent further damage to adjacent components and to minimize any risk of derailment. The speed limit at this track location was limited to 50 km/h due to the turnout degradation. Apart from that, it was also found that the condition of the thick SA-47 pads is still in very good condition. No severe abrasion on either bearers or pads can be observed.

3.2 GEOTECHNICAL INSPECTION

Geotechnical Services carried out an inspection and investigation by test pits at the area during a track possession. Figure 6.4e and f shows the cross profile of a test pit at the country-end v-crossing. From the geotechnical investigation, the ballast,
FIGURE 6.4
Crack of the concrete bearer under the rail pad. (a) Geometry of Clyde 700 diamond (looking toward Sydney). Poor line smoothness (lateral alignment) and vertical rail dip can be observed at the diamond. (b) Ballast pulverization at k-crossing, black greasy material was collected at fastening area. Note that there is a small gap between ballast and the ends of bearers subject to high-impact force due to some small dilation of ballast. (c) Ballast pulverization at country-end v-crossing. (d) Crack of the concrete bearer under the rail pad. (e) Cross profile of ballast and subgrade at the country-end v-crossing at rail seat area. Ballast contamination can be observed. It is noted that there is a high level of ballast densification due to high-frequency loading. (f) Cross profile of ballast and subgrade at the country-end v-crossing at middle span of bearer (1 m deep excavation).
capping layer, and formation conditions within the area were considered reasonable. The ballast layer depth at this location is about 285 mm on top of the dense capping layer. Certain level of fine contamination in ballast and its densification could be observed and the subsoil drainage was deemed not to be very good. However, there was no sign of distress and failure that could be observed from the substrata.

4 CURRENT ANALYSIS AND DESIGN METHODS

Turnout bearers undergo a lot more stress than conventional sleepers in railway tracks, as the train is moving from one track to another. Standards have been set in Australia regarding the design load on concrete sleepers, which can be found in AS 1085 Part 14, and for RailCorp in SPC 233 [16–19]. The Australian Standards 1085, part 14 (AS 1085.14) sets out the requirements for design, manufacture, testing, and installation of prestressed concrete sleepers and fastenings for railways across Australia [16]. Under its Appendix A, the standards outline the details of turnout bearers, the manufacturing details, loadings and design, fastenings, and testings. Turnouts consist of bearers of various lengths, with rails attached at predetermined locations to allow tracks to be connected to adjacent tracks. Crossovers may have discontinuous bearers along the connection, depending on how far apart the adjacent tracks are to each other. In addition to the standards set out in AS 1085.14 for standard sleepers, further considerations must be made for the extra dynamics of turnouts, including:

(i) Same method for standard bearers shall be used for distribution of axle load;
(ii) The dynamic loading on the bearer shall be designed for at least 2.5 times the static load for the impact factor;
(iii) Effects of centrifugal force should be allowed for curved rails;
(iv) Forces and moments shall be considered from point motors as well as other equipments required;
(v) Special load distributions specified by the designers;
(vi) Generally, bearers extend further than standard sleepers, and this must be accounted for in the ballast layout. In cases where they extend beyond the ballast shoulders, the moments and shears should be calculated assuming the ballast and subgrade behave as elastic foundation;
(vii) In the case of negative bending due to the extended bearer lengths, there must be adequate bending capacity built into the bearers to deal with the dynamic shear forces and bending moments. The bearer should be designed to deal with negative bending that is at least two-thirds of the maximum positive bending moments;
(viii) Transverse reinforcement should be provided at the bearer ends should there be substantial bending moments at the ends.

Based upon the AS 1085.14, the standard specifies the exact conditions and dimensions for concrete bearers. It has specified shear force and moment envelopes set out for the design of the bearers, as outlined in Figures 6.5–6.8 [17].
FIGURE 6.5
Shear force and bending moment envelopes for point motor bearer.

FIGURE 6.6
Shear force and bending moment envelopes for bearer lengths 2.6-2.8 m.
4.1 METHODS OF ANALYSIS

Actual contact pressure distribution in sleepers and bearers varies over time, making it virtually impossible to predict the exact conditions in actual conditions. Various hypothetical contact pressure distributions are shown in Figure 6.9. In Figure 6.10, the effect of the modulus \( K_s \) of ballast shows that the stiffer the sleeper support (the ballast), the more localized the contact stresses will be. Further, in Figure 6.11, the more flexible the sleeper is, again the more localized the contact stress will be.

4.1.1 Determination of rail seat load

The magnitude of the load applied to the sleepers from the rail depends on these known and unknown figures:

(i) the rail weight;
(ii) the sleeper spacing;
(iii) the sleeper stiffness characteristics;
(iv) the track modulus per rail;
(v) pad stiffness;

FIGURE 6.7
Shear force and bending moment envelopes for bearer lengths 2.8-5.2 m.
(vi) the amount of lateral movement between the rail and the sleeper;  
(vii) the amount of lateral movement between the sleeper and the ballast.

Various methods have been developed to analyze the rail seat load, and we will look at those used most frequently below.

### 4.1.2 The BOEF model

Under this model, suggested by Talbot and Clarke [20], one can determine the maximum rail seat load by:

\[
q_r = S \cdot k \cdot y_m \cdot F_1,
\]

where \(q_r\) is the predicted rail seat load; \(S\) is the sleeper spacing; \(k\) is the track modulus; \(y_m\) is the maximum rail deflection caused by the interaction of a number of axle loads about a given reference position (Figure 6.12); and \(F_1\) is the factor of safety to account for the variations in the track support caused by variations in the standard track maintenance (Clarke adopts \(F_1\) is the 1).
<table>
<thead>
<tr>
<th>Distribution of bearing pressure</th>
<th>Moment diagram</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform pressure</td>
<td></td>
</tr>
<tr>
<td>Flexure of sleeper produces variations from (a)</td>
<td></td>
</tr>
<tr>
<td>Tamped either side of rail</td>
<td></td>
</tr>
<tr>
<td>Laboratory test</td>
<td></td>
</tr>
<tr>
<td>Principal bearing on rails</td>
<td></td>
</tr>
<tr>
<td>Maximum intensity at ends</td>
<td></td>
</tr>
<tr>
<td>Maximum intensity at middle</td>
<td></td>
</tr>
<tr>
<td>Centre-bound</td>
<td></td>
</tr>
<tr>
<td>End-bound</td>
<td></td>
</tr>
<tr>
<td>Improper tamping at sleeper mid-span</td>
<td></td>
</tr>
<tr>
<td>Depression at rail seat</td>
<td></td>
</tr>
<tr>
<td>Mud pumping</td>
<td></td>
</tr>
<tr>
<td>Over-tamping at sleeper ends</td>
<td></td>
</tr>
</tbody>
</table>

**FIGURE 6.9**

Hypothetical distribution of bearing pressure and bending moment.
Sleeper/ballast pressure distribution

Bending moment diagram

$K_s = 20 \text{ MPa}$

$K_s = 50 \text{ MPa}$

$K_s = 100 \text{ MPa}$

$K_s = 200 \text{ MPa}$

**FIGURE 6.10**

Sleeper support modulus on ballast pressure and sleeper bending moment.
Sleeper/ballast pressure distribution

Bending moment diagram

"S6" Steel sleeper
(6 mm strip section)

"W14" Steel sleeper
(14 mm rolled section)

Mainline timber sleeper

Heavy haul concrete sleeper

FIGURE 6.11
Sleeper support stiffness on ballast pressure and sleeper bending moment.
4.1.3 The American Railroad Engineering Association method

American Railroad Engineering Association (AREA) developed a formula based implicitly on the elastic foundation model [20]. All experimental work used a constant spacing of 510 mm between the sleepers, with an assumed track modulus of 13.8 MPa. Therefore, the equation for the track spring constant for all bearers was defined as:

\[ D = \frac{k \cdot S}{C_1} \cdot \frac{A_S}{C_2} \times \frac{S}{10} \approx 13.8 \times 510 \times 10^3 \times \frac{A_S}{228 \times 10^3} \approx 3 \times 10^{-5} A_S, \]

where \( D \) is the track spring constant; \( k \) is the track modulus; \( S \) = sleeper spacing; and \( A_S \) is the effective sleeper support area beneath the rail seat (mm\(^2\)).

Based upon further experiments, it can be determined that the product of the track modulus and maximum deflection, \( k_y m \), to be nearly constant, and hence the effective area has little effect on the rail seat load. AREA then developed a simplified diagram (Figure 6.13), and showed that the AREA method could be expressed by:

\[ q_r = D_f \times P, \]

where \( q_r \) = maximum rail seat load (kN); \( P \) = design wheel load (kN); \( D_f \) = distribution factor, expressed as a proportion of the wheel load.

4.1.4 The ORE method

The Office of Research and Experiments of the International Union of Railways (ORE) developed a statistical method to calculate the maximum rail seat load [20]. Through experimental analysis, a formula was derived determining the maximum rail seat load:

\[ q_r = \bar{\varepsilon} \cdot c_1 \cdot P, \]

where \( P \) is the design wheel load, based upon the ORE formula for the impact factor; \( \bar{\varepsilon} \) is the dynamic mean value of the ratio \( q_r/P_S \) where \( q_r \) and \( P_S \) are mean values of the rail seat load and the static axle load, respectively; \( c_1 = \frac{\varepsilon}{\bar{\varepsilon}} \), where \( \varepsilon \) is the the maximum value of the ratio \( q_r/P_S \) and \( c_1 \) is \( \approx 1.35 \).
4.1.5 The three adjacent sleepers method

Based upon a simple assumption [20], the maximum possible sleeper loading occurs when the wheel load is distributed between three adjacent sleepers. The central sleeper will have a load of $x$ percent, while the two adjacent ones will have
a value of $x/2$ percent. Therefore, the value of $x$ will be 0.5, and hence the maximum rail seat load is:

$$q_r = 0.5P,$$

where $P$ is the design wheel load; $q_r$ is the predicted rail seat load.

### 4.2 SLEEPER BENDING MOMENTS

The material chosen for the analysis of maximum sleeper bending at the rail seat and center is irrelevant, but rather the flexural limits, usually expressed in the form of positive or negative bending moment capacity in concrete, imposed on the sleeper varies accordingly.

#### 4.2.1 Maximum sleeper bending moment at the rail seat

For determining the maximum bending, a uniform effective contact pressure distribution between the sleeper and the ballast is assumed [20] and the calculations based upon this simple assumption would yield an upper-bound solution sufficient for sleeper design purposes.

In the upper-bound solution, it is assumed that the support from the ballast is a point load at the end of the sleeper, equal to the rail seat load (Figure 6.15a), giving the equation:

$$M_r = q_r \left( \frac{l - g}{2} \right),$$

where $M_r$ is the maximum sleeper bending moment at rail seat; $q_r$ is the maximum rail seat load; $l$ is the overall sleeper length; and $g$ is the distance between the rail centers.

A more realistic method assumes half the rail seat load is distributed over the sleeper overhand length (Figure 6.15b), giving:
Pressure distributions for the calculation of design bending moment at rail seat [40–41].
(a) Upper-bound solution. (b) Less conservative solution. (c) Solution for sleepers fitted with bearing plates.
For sleepers with bearing plates fitted (Figure 6.15c), then the maximum bending moment at the rail seat for this condition is given by:

\[ M_r = q_i \left( \frac{l - g}{8} \right). \]

where \( j \) is the length of sleeper bearing plate.

### 4.2.2 Maximum sleeper bending moment at the center of the sleeper

Maximum bending at the center of the sleeper generally occurs when it is said to be centerbound (or when load is greater in the center). The two common solutions for this problem are the Battelle Solution and the Raymond Solution. Battelle assumes that the sleeper is resisted by a single-point load at the center of the sleeper, and can be calculated from:

\[ M_c = q_i \frac{g}{2}, \]

where \( M_c \) is the maximum sleeper bending at sleeper center.

This solution is believed to be extremely unlikely to occur in real-world situations, but can still be considered an upper-bound solution for the maximum sleeper bending moment at the center of the sleeper.

Raymond assumes a uniform bearing pressure over the total length of the sleeper, based upon experimental work carried out by Talbot [20]. Therefore, the maximum bending moment can be calculated from:

\[ M_c = q_i \left[ \frac{g}{2} - \frac{l}{4} \right] = q_i \left[ \frac{2g - l}{4} \right]. \]

### 4.2.3 Flexural requirements of sleepers

As noted earlier, the type of flexural limit imposed upon sleepers varies according to the sleeper type. For prestressed concrete sleepers, the design bending moment capacity of the sleeper is what must be considered. There are various methods of analysis, as outlined below.

#### 4.2.3.1 The AREA design method

The AREA method assumes that, due to a gradual redistribution over time, the sleepers will settle on the ballast to a uniform contact distribution over the entire length of the sleeper [20]. This produces positive flexure at the rail seat and negative flexure at the center of the sleeper. Based on its own method of determining the rail seat load, the maximum positive bending moment at the rail seat is given by:

\[ M_r = \frac{W(l - g)^2}{8}, \]
where $M_r$ is the maximum positive sleeper bending moment at rail seat; $l$ is the total sleeper length; $g$ is the distance between rail centers; $W$ is the assumed uniformly distributed load.

The maximum negative bending moment at the center is given by:

$$M_c = \frac{Wg^2}{8} - M_r,$$

where $M_c$ is the maximum negative bending moment at the center of the sleeper.

By combining these two equations, the solution is

$$M_c = q_r \left(2g - l\right).$$

This moment calculated is considered adequate under normal service conditions, but sleeper cracking will still occur due to abnormally high loads.

4.2.3.2 The ORE design method

The ORE design method is based upon experimental data from both laboratory and actual track conditions. It also assumes the maximum rail seat load to be that calculated with the ORE method [20].

The ORE experiments found that the actual stresses are higher than the theoretical stresses, and were also location dependent. A parameter was chosen to adjust for the difference, being the *lever arm*. Also, the load spread is assumed to be 45° through the sleeper from the rail seat (Figure 6.16). Therefore, the lever arm is:
\[ \Lambda = \frac{Q - b}{2}, \]

where \( \Lambda \) is the lever arm; \( Q \) is the sleeper end distance; and \( b \) is the half length of rail loading at sleeper neutral axis.

The theoretical bending moment is calculated with:

\[ M_t = \frac{q_l}{2} \Lambda, \]

where \( M_t \) is the rail seat bending moment.

To account for the theoretical and experimental moments measured, the lever arm is factorized:

\[ \Lambda_m = \phi \cdot \Lambda, \]

where \( \Lambda_m \) is the factorized lever arm and \( \phi = \) dynamic impact factor of the sleeper bending moment at the rail seat.

The empirical coefficient \( \phi \) is found to be \( \sim 1.6 \), and combining all these equations provides the maximum sleeper bending moment at the rail seat:

\[ M_t = \tau \cdot c_1 \cdot \phi \cdot \Lambda \cdot \frac{A}{2} \cdot P. \]

Similarly, the bending moment at the center can be determined to be:

\[ M_c = M_t \cdot \zeta \cdot \frac{EI(\text{centre})}{EI(\text{railseat})}, \]

where \( M_c \) is the bending moment at sleeper center; \( \zeta \) is the scale factor relating bending stress at rail seat to sleeper center; and \( EI \) is the sleeper stiffness.

### 4.2.3.3 Australian design code

The Australian design code is in many ways similar to the AREA method, but is more rigorous [20]. The positive moment, with load distributed as shown in Figure 6.17a, at the rail seat is given by:

\[ M_{r+} = \frac{q_r(l - g)}{8}, \]

where \( M_{r+} \) is the maximum positive bending moment at rail seat; \( q_r \) is the design seat load; \( l \) is the sleeper length; and \( g \) is the distance between rail centers.

The maximum negative bending moment is given by either 67\% of the maximum positive moment, or 14 kN m, whichever is greater.

For the center of the sleeper, the maximum positive moment, with load distributed as shown in Figure 6.17b, is given by:

\[ M_{c+} = 0.05q_r(l - g), \] for standard gauge,

\[ M_{c+} = 0.10q_r(l - g), \] for narrow gauge,

where \( M_{c+} \) is the maximum positive bending moment at the sleeper center.
The maximum negative bending moment, with load distributed as shown in Figure 6.17c, for the center is given by:

\[ M_{c-} = 0.5 \cdot \left[ q_1 \cdot g - W \cdot g (l - g) - W \frac{(2g - l)^2}{8} \right], \]

where \( M_{c-} \) is the maximum negative bending moment at sleeper center; \( W = \frac{4q_r}{3l - 2g} \) (maximum load per unit length of sleeper).

4.3 CURRENT DESIGN METHOD

4.3.1 Essential loading determination

To begin the analysis, basic loading diagrams must be analyzed. Determining the load experienced by the rail seat requires the usage of the methods mentioned earlier, being the BOEF model, AREA method, ORE method, and the three adjacent sleepers method, and comparing the results [20]. Using the discussed analysis methods, the load
for a sleeper spacing of 621 mm can be approximated by adjusting for the sleeper spacing, which is proportional in the earlier equations, with the results shown in Table 6.2.

Determining the contribution of the design and track layout to the failure of the bearer requires an assumption of the worst-case scenario. As seen in Figure 6.18, the bearer which failed in the diamond has been designed so that the rail wheel passes over a gap upon the track. Assuming worst-case scenario, the whole axle load would be taken upon the remaining wheel on the other rail, exerting a total $P$ equivalent to the axle load. This is unlikely in real life, as the axle is normally apart of a bogey system comprising two axles. So the most likely scenario in this situation is that the load will be shared among the remaining wheels and not taken upon by the single wheel.

The Standards used within New South Wales, Australia, specify a maximum axle load of 295 kN for locomotives and 230 kN for rolling stock, depending on the track class (i.e., track class one is for freight trains, while track class two is for passenger trains). Again, assuming the worst-case scenario, the load of 295 kN will be used in the analysis [18]. The distance of the load from the bearer end can be found using the concrete tie layout detail drawing, with the specifications outlined below in Figure 6.19. The load is assumed to be approximately in the middle of the plate (numbered 9-10-11-12).

### Table 6.2 Adjusted Rail Seat Loading for Bearer Spacing 621 mm

<table>
<thead>
<tr>
<th>Method</th>
<th>Maximum Rail Seat Load (kN, 760 mm)</th>
<th>Maximum Rail Seat Load (kN, 621 mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Three adjacent method</td>
<td>$q_r = 0.50P$</td>
<td>$q_r = 0.50P^a$</td>
</tr>
<tr>
<td>BOEF model</td>
<td>$q_r = 0.43P$</td>
<td>$q_r = 0.35P$</td>
</tr>
<tr>
<td>APEA method</td>
<td>$q_r = 0.60P$</td>
<td>$q_r = 0.49P$</td>
</tr>
<tr>
<td>ORE method</td>
<td>$q_r = 0.65P$</td>
<td>$q_r = 0.53P$</td>
</tr>
</tbody>
</table>

$^a$Three adjacent method is a constant 0.50P regardless of spacing.

---

FIGURE 6.18
Track layout and point of failure (red), and gap in rail (green).
FIGURE 6.19
Concrete tie layout detail.
This leads to the basic loading diagram of the bearer in question, shown in Figure 6.20. There is a track stiffness which provides support underneath the bearer along its length. The distance from the bearer end is an approximation, as the rail seat also accommodates the check rail, which is designed to keep the wheel on the track during the transition (note that all figures are in millimeters).

5 FINITE ELEMENT MODELING OF TURNOUT DIAMOND

5.1 COMPLETE DIAMOND ANALYSIS

A finite-element model of turnout diamond has been developed using a finite-element package, STRAND7. Analysis of the complete diamond requires modeling each bearer in the whole crossover, on which rail is applied, as shown in Figure 6.21.

Determining the load points on the rail requires a review of rolling stock data to discover the train with the closest loading points (assuming the closer the axles are, the more the forces will interact). Figure 6.22 shows the typical train bogey and axle setup. Assuming the axle load is equal throughout, the bogeys are closest at the carriage joints, as opposed to the singular carriage itself (i.e., $2 \times$ body overhang $<$ body centers). The actual data from each rolling stock, shown in Table 6.3, shows that Train 1 has the shortest distance between the two furthest axles compared to the three other rolling stock carriages.

Figure 6.23 shows the half model with the point loads applied. The track stiffness on all the bearers is calibrated using the field data, which is the same as on the singular bearer analysis, 0.1689 MPa/mm. Again, it is assumed that the axle load above the failed bearer is supported by the single rail (noticeably with the larger load shown in the figure).

5.2 INDIVIDUAL BEARER ANALYSIS

The failure analysis of the individual bearer involves modeling the single bearer, and the rail pad and steel pad on which the rail sits, is shown in Figures 6.24 and 6.25 [43]. Again, the specifications can be obtained from the concrete tie layout detail, with the bearer cross-section of $250 \text{ mm} \times 250 \text{ mm}$, as per RailCorp standard [16,17].
FIGURE 6.21
Complete crossover modeled with detailed model of rail on top. (a) Turnout diamond model. (b) More detailed rail model (left) and simpler rail model (right).

FIGURE 6.22
Typical carriage layout.
Applying the loads calculated earlier to the rail seat requires a conversion to a pressure over the area of the pad. To determine the contact area of the pad, the rail dimensions are used, with the foot width (146 mm) and rail pad width (115 mm), giving an area of 16,790 mm$^2$. This can give a pressure load for each of the methods mentioned earlier, as shown in Table 6.4.

These have to be adjusted to accommodate the model, as the pressure area is slightly smaller, producing a slightly larger pressure, as shown in Table 6.5.

Track stiffness can be applied to the base of the model to provide support for the load and it has been derived from the specifications discussed earlier. This value has been converted to MPa/mm, and as it is uniform across the whole diamond, the minimum support required (i.e., the maximum value) is used in the calculations, being 0.1689 MPa/mm. The model validation has been carried out in comparison with field measurements (displacements and track modulus) to ensure that the model can accurately provide practical results.

Table 6.3 Distance Between Bogey and Axles

<table>
<thead>
<tr>
<th>Train</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axle distance</td>
<td>2300</td>
<td>2400</td>
<td>2440</td>
<td>2440</td>
</tr>
<tr>
<td>Bogey distance</td>
<td>3998</td>
<td>3876</td>
<td>4670</td>
<td>3831</td>
</tr>
<tr>
<td>Axle distance</td>
<td>2300</td>
<td>2400</td>
<td>2440</td>
<td>2440</td>
</tr>
</tbody>
</table>

FIGURE 6.23
Half bearer with loads applied.
5.3 CONTACT SURFACES

Using the contact surface element provided by STRAND7, as for the joining of non-aligned bricks, links have to be formed. This means that the load distributed through the nodes are not transferred to the other nodes, but rather spreads out on the whole

---

**FIGURE 6.24**

Material models in STRAND7. (a) Stress-strain relationship of structural steel. (b) Stress-strain relationship of concrete.
FIGURE 6.25
Finite-element modeling of turnout bearer. (a) Bearer model. (b) Loading on rail pads.

Table 6.4 Theoretical Loading Pressure for Respective Rail Seat Loads

<table>
<thead>
<tr>
<th>Method</th>
<th>Rail Seat Load (kN)</th>
<th>Rail Seat Pressure (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Three adjacent method</td>
<td>147.5</td>
<td>8.785</td>
</tr>
<tr>
<td>BOEF model</td>
<td>103.25</td>
<td>6.149</td>
</tr>
<tr>
<td>AREA method</td>
<td>144.55</td>
<td>8.609</td>
</tr>
<tr>
<td>ORE method</td>
<td>156.35</td>
<td>9.312</td>
</tr>
</tbody>
</table>
brick face, which may have reduced the overall pressure. This contact element (as shown in Figure 6.26) enables two solid elements (bearers and pads) to contact and redistribute forces.

### 6.1 INDIVIDUAL BEARER ANALYSIS

After running the model analyses, the model results showing how the load is spread along the rail pads onto the bearer, and also how the bearer is supported by the ballast underneath. The forces experienced by the models are shown in Figure 6.27, below, with the maximum forces (FZ) shown at the top of the table.

#### Table 6.5 Actual Model Loading Pressure for Respective Rail Seat Loads

<table>
<thead>
<tr>
<th>Method</th>
<th>Model Rail Seat Area (mm$^2$)</th>
<th>Rail Seat Pressure (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Three adjacent method</td>
<td>14,820</td>
<td>9.953</td>
</tr>
<tr>
<td>BOEF model</td>
<td>14,820</td>
<td>6.967</td>
</tr>
<tr>
<td>AREA method</td>
<td>14,820</td>
<td>9.754</td>
</tr>
<tr>
<td>ORE method</td>
<td>14,820</td>
<td>10.550</td>
</tr>
</tbody>
</table>

**FIGURE 6.26**
Attaching nonaligned nodes to brick faces, such as pads onto bearer.
FIGURE 6.27
Finite-element analysis results of individual bearer. (a) Pressure distribution from loading. (b) Forces experienced from loading. (c) Forces experienced underneath bearer from loading.
6.2 COMPLETE DIAMOND ANALYSIS

The resultant model of the diamond crossover analysis can be found in Figure 6.28. They are the loading results from the analysis. As can be seen from the numbers, the results correlate well with the individual bearer forces experienced, sitting in between the minimum and maximum resultant force figures, at 378N.

6.3 DISCUSSIONS

The reaction forces underneath the bearers show that as the load is increased, the bearer will definitely experience a greater force. Figure 6.29 presents the sleeper/ballast contact pressures obtained from the methods of determining the rail seat load. It is found that the data is in very good agreement with those obtained from the more complete crossover model. It is found that the maximum resultant forces from the current methods of analysis are still far below the force envelope specified in the design standards. Even if it is to be increased substantially, the load would still not cause a failure of the bearer, according to the finite-element analyses.

This puts into question whether the diamond design contributed to the failure. One factor that was not taken into account for the analysis is the possibility of high-amplitude dynamic forces experienced by the rail and bearer. The dynamic impact forces would in turn apply a greater moment force upon the bearer. It is clear from Figure 6.29 that if the rail seat loads are to increase only about 20-30%, the contact pressure between ballast and sleeper will have reached the allowable limit of 750 kPa [16,17]. Considering AS1085.14, if the specified impact factor of 2.5 is adopted, the resultant dynamic wheel load will gain 27% over the maximum allowable load (295 kN). It implies that potentially the ballast would significantly and rapidly settle under the crossings and diamonds. The large settlement can cause additional bending moment actions and their consequences, resulting in a flexural failure mode of bearer as observed in the field.

Table 6.6 illustrates an estimation of turnout renewal costs when failure of diamond component incurs. It is important to note that criticality of turnout failure is relatively high as a turnout (only about 100-150 m in track length) tends to connect two or more plain tracks together. As such, its consequences are progressive and highly interconnected to the service values other connected tracks provide. This provides the fundamental rationale for higher priority of turnout renewal and maintenance than that of a plain track. From the cost estimate list, it can be seen that material costs cover around 6-7% of the total cost. It implies that re-designing critical structural components such as concrete bearers for the capability to resist higher impact loads could potentially save the associated penalty costs of 77%.
FIGURE 6.28
Finite-element analysis results of turnout diamond. (a) Crossover model. (b) Pressure distribution from loading. (c) Forces experienced from loading. (d) Forces experienced underneath bearers from loading.
Concrete sleepers/bearers are the main components supporting steel works and redistributing dynamic forces onto ballast and formation. Because of their low cost, high durability, ease in construction, and maintenance, concrete bearers will continue to be the popular components in modern turnout systems. However, wheel-rail contact over the crossing transfer zone has a dip-like shape and can often cause detrimental

![Graph showing the relationship between rail seat load (kN) and sleeper/ballast contact pressure (kPa). The equation is $y = 3.9967x$ with $R^2 = 0.9998$.]

Table 6.6 Estimate Costs Associated with Turnout Renewal Due to Systemic Failure

<table>
<thead>
<tr>
<th>Items</th>
<th>Renewal Cost ($)</th>
<th>% Cost component</th>
</tr>
</thead>
<tbody>
<tr>
<td>Materials</td>
<td>200,000</td>
<td>6.2</td>
</tr>
<tr>
<td>Machinery</td>
<td>75,000</td>
<td>2.3</td>
</tr>
<tr>
<td>Access/possession fee</td>
<td>50,000</td>
<td>1.6</td>
</tr>
<tr>
<td>Human resources and labors</td>
<td>150,000</td>
<td>4.7</td>
</tr>
<tr>
<td>Commission and signaling</td>
<td>100,000</td>
<td>3.1</td>
</tr>
<tr>
<td>Re-design, manufacturing</td>
<td>50,000</td>
<td>1.5</td>
</tr>
<tr>
<td>Formation improvement</td>
<td>100,000</td>
<td>3.1</td>
</tr>
<tr>
<td>Others (train replacement busses, logistics, delay costs, penalty fees, opportunity costs, etc.)</td>
<td>&gt;2,500,000</td>
<td>77.5</td>
</tr>
<tr>
<td>Total</td>
<td>&gt;3,225,000</td>
<td>100</td>
</tr>
</tbody>
</table>

7 CONCLUSION

Concrete sleepers/bearers are the main components supporting steel works and redistributing dynamic forces onto ballast and formation. Because of their low cost, high durability, ease in construction, and maintenance, concrete bearers will continue to be the popular components in modern turnout systems. However, wheel-rail contact over the crossing transfer zone has a dip-like shape and can often cause detrimental
impact loads on the railway turnout and its components. The field maintenance reports have demonstrated that a number of concrete bearers failed under repeated impact loadings, especially under turnout diamonds. Such an incident has caused major downtime, high cost of turnout renewal, and reduced reliability of train operations. As a result, a nonlinear 3D finite-element model has been developed, using commercial finite-element package STRAND7, to evaluate the design methods and to analyze the failure mechanism of the concrete sleepers/bearers at an urban turnout diamond. The finite-element modeling of the bearers and the diamond crossover has shown that the contemporary design methods may underestimate the vertical quasi-static loads for design of concrete bearers. The finite-element analysis results illustrate that the dynamic load action could be at or higher than the allowable sleeper/ballast contact pressure due to the variation of dynamic loading condition by 20-30%. It is highly likely that the variation could damage the supporting ballast and result in a premature flexural failure mode of concrete bearers. As a result, the outcome of this study recommends that a higher impact factor (>2.5) for the design of turnout bearers be adopted in railway industry. This adoption can potentially save millions of dollar from prioritized maintenance and unplanned renewal and from the penalty costs associated with the turnout diamond failure.

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References


