EFFECT OF VERTICAL MISALIGNMENT OF ADJACENT SLEEPERS ON THE INCREASE IN DYNAMIC LOADS AROUND RAIL JOINTS

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SUMMARY

In spite of the extensive usage of continuous welded rails, a number of rail joints still exist in the track. Although a number of them exist as part of turnouts in the yards where the speed is not of concern, the Insulated Rail Joints (IRJs) that exist in ballasted tracks remain a source of significant impact loading. A portion of the dynamic load generated at the rail joints due to wheel passage is transmitted to the support system which leads to permanent settlements of the ballast layer with subsequent vertical misalignment of the sleepers around the rail joints. This vertical misalignment of the adjacent sleepers forms a source of high frequency dynamic load raisers causing significant maintenance work including localised grinding of railhead around the joint, re-alignment of the sleepers and/ or ballast tamping or track component renewals/ repairs. These localised maintenance activities often require manual inspections and disruptions to the train traffic leading to significant costs to the rail industry. Whilst a number of studies have modelled the effect of joints as dips, none have specifically attended to the effect of vertical misalignment of the sleepers on the dynamic response of rail joints. This paper presents a coupled finite element track model and a rigid body track–vehicle interaction model through which the effects of vertical misalignment of sleepers on the increase in dynamic loads around the IRJ are studied. The finite element track model is employed to determine the generated dip from elastic deformations as well as the vertical displacement of sleepers around the joint. These data (dip and vertical misalignments) are then imported into the rigid body vehicle-track interaction model to calculate the dynamic loads.

1. INTRODUCTION

Rail joints are used commonly in rail track structures. In the conventional rail joints, two joint-bars are used to connect the ends of the adjoining rails. Joint-bars, bolted to the web of the rails, are used to align the rail ends to create a smooth running surface. Insulated rail joints are a group of rail joints which electrically isolate the rails as part of the automatic signalling system. The low bending stiffness of the joint-bars at rail joints compared to the rail section results in high vertical deflection and dynamic forces under passing wheels, which subsequently leads to deterioration of the rail head, joint-bars and bolts as well as degradation of ballast and sub-grade in the vicinity of the joints. The performance of the IRJs has become a significant economic problem due to lower service life and higher variability relative to continuous welded rails (CWR).

For the analysis of dynamic loads over rail joints, multi-body dynamics and finite element method have been used in the literature. Jenkins et al. [1] modelled the dipped rail joint as a dipped continuous beam supported by sets of springs and dashpots at the location of sleepers. They reported that there existed two contact force peaks (generally known as P1 and P2); the first is characterised with high amplitude and high frequency, and the second is characterised with low amplitude and low frequency. Wu and Thompson [2] developed a dynamic rail wheel contact model for the determination of rail joint impact noise. They studied the influence of gap size, vertical misalignment of rail ends and rail joint dip on the vertical dynamic loads over the joint. Koro et al. [3] established a dynamic finite element model to investigate the edge effects of rail joint. Timoshenko beam elements were used to model the joint structures including the joint bars. Tie
springs were employed to connect the joint bars to the rail supported on a discrete elastic foundation. They investigated the influence of gap size and train speed on the impact force over the joint. They reported that the values of dynamic loads were more sensitive to the gap size than the train speed for velocities lower than 150 Km/h. Pang and Dhanasekar [4] presented a FE model for contact-impact prediction in the vicinity of end-post on the railhead. The model consists of a single wheel carrying a specified share of wagon load rolling at a steady state under the prescribed longitudinal velocity along the railhead crossing the joint. The model was used to predict impact force time series as the wheel crosses the end-post for various configurations of the insulated rail joint. In this model, a maximum impact factor of 1.16 was generated between the new wheel and new IRJ at a train speed of 80 km/hr. They also investigated the influence of different properties of IRJ on dynamic loads. They proposed the best design parameters for IRJ include a glued fibreglass end post with 5mm gap size. Dhanasekar et al. [5] presented a field test carried out by the CRE with support from Queensland Rail (QR) which were used to validate the FE model.

According to the available literature, although many attempts have been made to improve the structural design of IRJs, very limited works have been carried out to look into the effects of the response of the track and support system to the dynamic loads generated at the wheel - IRJ interface. This paper is an attempt to examine this important aspect.

The IRJs exhibit high variability in-service life and failure modes. The IRJs used in Australia are assembled in factory with good quality control procedure, but their service life, even under similar traffic condition, significantly varies, which causes significant concern. A research hypothesis is, therefore, put forward that the misalignment in the IRJ support system may be the reason for the variability in service life and failure mode. Variability of track vertical profile is one of the factors that leads to misalignment of joints. In particular, due to the absence of guidelines for the installation and maintenance procedure of IRJs, these irregularities might be much higher around rail joints than those in a ballasted track. In addition, the high dynamic loads and vibrations that excite the rail joints, after a prolonged period of time, may cause degradation of the ballast and subgrade layers leading to growth of track vertical irregularities around the joint.

Suzuki et al. [6] carried out some limited studies on the influence of track vertical irregularities on the dynamic loads over the rail joint. They presented a mechanistic track dynamic model around the joint. They investigated the influence of vertical track irregularity around a joint on dynamic wheel loads and rail seat forces. They showed that the differential settlement of sleepers around the rail joint has a great influence on the dynamic impact load.

Davis and Akhtar [7] modelled a typical six-bolt insulated joint of RE136 rail and analysed the stresses and strains in IRJ components. They studied the influence of differential settlement of support on IRJ. It was shown that the stress levels in IRJ components significantly increased due to irregular support system.

Three dimensional FE modelling of IRJs under wheel loads, due to high degree of sophistication and significant computational cost lacks convenient application to calculate dynamic impact loads due to sleeper misalignments. On the other hand, in multibody dynamic modelling, the complex geometry of IRJs can not be accurately modelled. Therefore, in this research a combination of a finite element model and a multi-body dynamic model is used. A global track model with IRJ components was developed to calculate the elastic dip as well as the vertical irregularity produced due to the sleeper displacements around the IRJ. The predicted vertical irregularities are then imported to the vehicle-track dynamic model to calculate the dynamic loads. In this paper, the influence of four parameters on the dynamic loads over rail joint is investigated: (i) location of misaligned sleeper relative to the rail joint, (ii) the magnitude of sleeper misalignment, (iii) the speed of the vehicle and (iv) the magnitude of the static wheel load.

2. FIELD MEASUREMENTS

In the Australian Rail Track Corporation (ARTC) New South Wales rail network, track geometry irregularities cause a lot of maintenance work such as ballast tamping and track component renewals every year. Fig. 1 and Table 1 exhibits some typical data obtained from AK track recording car: the data are so chosen that a joint exists in the range. In addition, the ballast degradation around the IRJs causes high vertical deflection of the IRJs under the passing wheel resulting in fracture of joint-bar and, in turn failure of the IRJs. The performance of IRJs has become a significant economic problem as their service life can be as short as 15% to 35% of the continuous welded rails.

The ARTC rail network in New South Wales is carrying more than 150 million gross tons (MGT) of coal per annum. This rail network consists of concrete sleepers, 60 kg rails and the e-clip fastening system. In this line the allowable axle load and train speed are 30 ton and 120 km/h respectively. The IRJs used in this line are produced in Thermit Australia factory in Brisbane with six-bolted jointbars.
3. SIMULATION METHODOLOGY

In multi-body dynamic methods the rail joint is mostly simulated in the literature as a dip with a cusp shape described by an assumed function based on a specific depth, length and angle between the rail end and horizontal [1,8,9,10,11,12]. However, there are variations in the methods of modelling the dip in mathematical representation for the dynamic models. For instance, Jenkins considered the dip with the fixed ramp angle (0.02 rad). Esveld [8] and AAR [9] used a cosine function and an exponential function respectively to describe the dipped joint. In this paper, the dip shape is not calculated by a predefined mathematical function. It is simulated by summation of elastic and inelastic dip formed from a FE model in which the elastic dip is generated due to the stiffness discontinuity in the running surface and the inelastic dip is determined from the vertical settlement of support system around the joint. In other words, the dip shape is the dipped rail profile around IRJ generated due to the stiffness discontinuity and vertical settlement of adjacent sleepers. Therefore, the depth and angle of the dip changes based on the severity and location of the settled sleeper and in each case the dip ramp angle changes as a function of the dip length. The calculated irregularities are imported into a vehicle-track dynamic model to calculate the dynamic loads.

4. TRACK MODEL AROUND IRJ

A track model, with IRJ, of length 24.6 metres was developed using ABAQUS commercial software. In this model, the rails and sleepers are considered as one-dimensional beam elements in a 3-dimensional space and their stiffness is defined by material and section properties. The rail pad and ballast are represented using spring elements that have an appropriate stiffness through existing typical Australian tracks. For simulation of IRJ, a 4 mm thick insulation material (endpost) is placed in the middle of the track model. Joint-bars are modelled as two beams on both sides of the rails. They are modelled with rectangular cross sections and with material properties similar to the rail (steel). The insulation material between rail and jointbar is defined using spring elements with vertical, longitudinal and lateral stiffness. The six bolts holding the jointbars are modelled as displacement constraints between the rail and joint bar. In fact the rail web and the two jointbars are constrained to have relative movement only at bolt positions, so the IRJ assembly receives the load from passing wheels through the bolts under bending of jointbars and contact between the fishing surfaces of the rail and jointbars.
The model developed in this research is based on a typical IRJ design used in Australia with 4 mm square-cut gap, AS 60 rail and six-bolted jointbar. The track model and IRJ details are shown in Figure 2. The properties of the track and IRJ components used in this modelling are shown in Table 2.

5. VEHICLE-TRACK DYNAMIC MODEL

Non–linear Transient analyses were undertaken using VAMPIRE software to calculate the vertical wheel-rail contact forces. Using a multi–body modelling method the mathematical model of vehicle-track interaction was assembled. In this paper, the vehicle was modelled as a freight car with two axles as shown in Figure 3. The connections between the car body and two wheel-sets were modelled using the stiffness and damper elements. The car mass was characterised by its mass and inertia properties, and the geometric position of its centre of gravity. Each wheelset was characterised by its mass, inertia properties and the geometric position of its centre of gravity. The roll and yaw inertias were assumed to be the same and the nominal radius of the wheel was assumed equal to the height of the centre of gravity above the rail. The track inputs were automatically applied to all the wheelsets in the vehicle, with the timing of the input delayed according to the longitudinal position of the wheelset.

Track geometry inputs were specified in distance history form in two different input files including the track design and track irregularity files. The wheelsets were grounded by springs in the vertical, lateral, roll and yaw senses. The program then assembled a linear stiffness matrix for the vehicle, including all suspension elements with a stiffness component. The stiffness matrix was inverted, and the weights of each element were applied to the vertical degrees of freedom. The resulting displacements of each degree of freedom were used to calculate the wheel loads.

The time response calculation was then carried out for each case of speed and wheel/rail contact case. The main time–stepping loop commenced with the displacements and velocities of all the degrees of freedom of the vehicle which were calculated at the previous time step (zero for the first time step). Once the displacements of the track and the displacements and velocities of the wheel-sets were determined, the forces in all of the suspension elements were calculated easily from the given the displacements and velocities of the masses they connect.

The forces on each degree of freedom were summed, and the resulting accelerations determined by dividing the forces by the mass and the moments by the inertia. Velocities and displacements at the end of the time step were determined by integration of the acceleration.

In this model, the track was modelled as a beam on elastic foundation, in which the track vertical flexibility was defined as a series of springs and dampers. The vertical track stiffness used in the vehicle/track dynamic model was considered as the equivalent stiffness of the FE track model.

![Figure 2. Track model with IRJ](image)
Table 2. Properties of the track and IRJ components used in the FE modelling

<table>
<thead>
<tr>
<th>Property</th>
<th>Track Component</th>
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<tr>
<td></td>
<td>Rail</td>
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<td>E (GPa)</td>
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<tr>
<td>G (GPa)</td>
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<td>I (m^4)</td>
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<tr>
<td>Stiffness (N/m)</td>
<td>_</td>
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</tbody>
</table>

Figure 3. Model used for simulation of Vehicle-track dynamic interaction

6. RESULTS AND DISCUSSION

In the simulations the mass of the car body and wheelsets were kept as 500 KN and 20 kN respectively, giving a static wheel load of 135 KN. The analyses were conducted for four different conditions as follows:
- IRJ with no sleeper misalignment.
- IRJ with vertical misalignment of the left sleeper.
- IRJ with vertical misalignment of right sleeper.
- IRJ with vertical misalignment of left and right sleepers.

The simulations with the above mentioned conditions were carried out with four different vehicle speeds and with four magnitudes of sleeper misalignments. The vertical irregularities generated around the IRJ due to elastic dip or and sleeper misalignments in either left, right or both sides of the IRJ with magnitudes of two, four, eight and sixteen millimetres were simulated (Figure 4). The irregularities were imported to the vehicle-track dynamic model to calculate the dynamic loads for vehicle speeds of 10 m/s, 20 m/s, 30 m/s and 35 m/s.

In the first stage, the simulation was conducted for the rail joint without any sleeper misalignment. In this stage, the dynamic load was generated only due to the elastic dip at rail joint. In the next simulations, the values of dynamic loads around rail joint were aggravated due to the sleeper misalignments.

The values of the dynamic loads generated around rail joint for four different speeds are shown in Figures 5 to 7. Figure 5 presents the waveform of dynamic loads excited due to misalignment of left sleeper. It can be seen that the values of dynamic loads over the joint significantly increase, as the amplitude of sleeper misalignment increases.

It is shown that for train speed of 10 m/s the values of dynamic loads increase by 35% when the vertical misalignment of left sleeper increases to 16 millimetres. As indicated in this figure, this value is increased by 38%, 53% and 106% for train speed of 20 m/s, 30 m/s and 35 m/s respectively.

Figure 6 demonstrates the effect of misalignment of the right sleeper on dynamic loads generated
around rail joint. For train speed of 10 m/s the dynamic load over the joint is increased by 15% when the vertical misalignment of the right sleeper increases to 16 millimetres. This value reaches 95% for train speed of 35 m/s.

The values of dynamic loads due to misalignment of the both, left and right, sleepers are presented in Figure 7. It can be seen that the dynamic load over the rail joint is increased by 18% when the vertical misalignment of sleepers increases to 16 millimetres for train speed of 10 m/s. The increase in values of dynamic loads reaches 67% for train speed of 35 m/s.

It can be seen that the asymmetric misalignment of sleepers around rail joint has more influence on dynamic loads than the symmetrical sleeper misalignments for train speeds more than 20 m/s. As mentioned above, the influence of vertical misalignment of left sleeper on dynamic loads is higher than those of the right sleeper. However, the location of peak dynamic load is shifted to the left hand side of the rail joint centre (joint gap) and close to the sleeper. When the left sleeper is settled, the wheel starts moving downward before reaching the joint gap causing high impact in the left side of the joint gap (close to the sleeper). Therefore, this scenario could cause high stresses in the bolt holes in the left side of the joint gap. In the case of symmetrical sleeper misalignment as well as misalignment of right sleeper, the dynamic peak load will be applied in closer distance to the joint gap which can cause high bending stresses in the joint bar as well as high stresses in the middle bolt holes.

Comparisons of the dynamic force factors generated over the rail joint are graphically illustrated in Figure 8. Dynamic force factor is the proportion of the dynamic wheel load to the static wheel load (135 KN). This figure shows the progressive increase in dynamic loads over the rail joint due to misalignment of adjacent sleepers. This also shows the considerable influence of train speed on progressive increase of dynamic loads.

**Figure 4. Track vertical profile around IRJ due to vertical misalignment of adjacent sleepers. (a) Top: misalignment of sleepers in both sides of the IRJ, (b) middle: misalignment of sleeper in the left hand side of the IRJ, (c) lower: misalignment of sleeper in the right hand side of the IRJ**
Figure 5. Wheel loads around IRJ with misalignment of left sleeper. (a) Top left: speed 10 m/s, (b) Top right: speed 20 m/s, (c) Lower left: speed 30 m/s, (d) Lower right: speed 35 m/s

Figure 6. Wheel loads around IRJ with misalignment of right sleeper. (a) Top left: speed 10 m/s, (b) Top right: speed 20 m/s, (c) Lower left: speed 30 m/s, (d) Lower right: speed 35 m/s
Figure 7. Wheel loads around IRJ with misalignment of left and right sleepers. (a) Top left: speed 10 m/s, (b) Top right: speed 20 m/s, (c) Lower left: speed 30 m/s, (d) Lower right: speed 35 m/s

Figure 8. Dynamic force factor excited over the rail joint versus sleeper misalignment. (a) Top left: left sleeper misalignment, (b) Top right: right sleeper misalignment, (c) Lower: left and right sleeper misalignments
Figure 9. Wheel loads over the rail joint, (a) Top left: wheel load 30 KN, left sleeper misalignment, (b) Top right: wheel load 80 KN, left sleeper misalignment, (c) middle left: wheel load 30 KN, right sleeper misalignment, (d) middle right: wheel load 80 KN, right sleeper misalignment, (e) Lower left: wheel load 30 KN, left and right sleeper misalignments, (f) lower right: wheel load 80 KN, left and right sleeper misalignments
To study the influence of the axle load on the dynamic impacts around the rail joint, the simulations were repeated for two other static wheel loads. Figure 9 demonstrates the values of dynamic loads around the rail joint with and without sleeper misalignments for wheel loads of 30 KN and 80 KN. These analyses were carried out only for train speed of 20 m/s.

As shown in figure 9, for 16 mm sleeper misalignment, the dynamic loads increased by 240% and 70% for static wheel loads of 30 KN and 80 KN respectively. By comparing these values with those obtained in the previous simulations (figures 5 to 7), one can see the higher rate of increase in dynamic loads for lower static wheel loads relative to high static wheel loads. This fact can be seen more clearly in figure 10 where the values of wheel loads versus static wheel loads are drawn.

When the value of static wheel load increases from 30 KN to 135 KN the maximum wheel load over IRJ increases by 200%, 112% and 54% for 2 mm, 8 mm and 16 mm sleeper misalignment respectively.

Figure 11, shows the values of wheel loads around rail joint versus vehicle speed. It shows that the influence of train speed on wheel loads around rail joint increases when the magnitudes of sleeper misalignments increase. For rail joint without sleeper misalignments, the wheel load increases by 10 percent when the vehicle speed is changed from 10 to 35 m/s. This value reaches 43% and 84% for the rail joint with eight and sixteen millimetres sleeper misalignment respectively. This shows that for high values of sleeper misalignments around rail joint, the increase in train speed has more influence on dynamic loads than increase in train axle load.

CONCLUSION

When a new insulated rail joint is perfectly installed in a well maintained rail track, the dynamic wheel loads are only excited due to the joint gap when the train passes the joint. However, in reality, deficiencies in installation and maintenance procedure of IRJs as well as the ongoing degradation of track support system stiffness cause considerable misalignments in vertical position of sleepers around rail joints. As a result, the waveforms of dynamic train-track interaction forces are adversely affected, and this speeds up the deterioration process of IRJs.

In this paper, a coupled finite element track model and a rigid body track–vehicle interaction model was employed to study the effects of vertical misalignment of sleepers on the progressive increase in dynamic loads around the IRJ.

It was found that sleeper vertical misalignment can cause the dynamic load over rail joint to increase by more than 100 percent. It was shown that for train speed more than 20 m/s asymmetric sleeper misalignment has more influence on dynamic loads than symmetrical sleeper misalignments around IRJ.

The vehicle speed has also a significant effect on dynamic loads over the joint. It was shown that the influence of train speed on dynamic loads over a rail joint with misaligned supports is much higher than that of the rail joint with no support irregularity. For a rail joint with no sleeper misalignment, the train axle load is the main factor which controls the values of dynamic impacts. However, for a rail joint with high vertical irregularity of support system, the increase in train speed has more influence on dynamic impact loads than increase in train wheel load.
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