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NONLINEAR PHENOMENA OF SHORT AND LONG WAVELENGTH RAIL DEFECTS ON VEHICLE-TRACK INTERACTION

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Key words: Dynamic interaction, Coupling vehicle-track modeling, short wavelength defect, long wavelength defect, coupled rail surface defects.

Abstract. Every day, over 100 million of train traffics are being operated globally. The first priority in train operation management is ‘*safety*’. During each operation, the dynamic interactions between vehicle and track usually impose vibrations and acoustic radiations and become a moving source of vibroacoustics along the railway network. Especially when there is imperfection of either wheel or rail, the dynamic amplification of loading conditions and reflected vibration effects on infrastructure and rolling stocks is significantly higher. In practice, imperfection of rail tracks is inevitable and can be classified into short wave length and long wave length defects. The imperfections can statistically induce high-intensity impact loadings up to 25% of *annual* track loading conditions. The short wavelength defects include high-frequency related rail surface defects such as dipped joint rails, rail squats, rolling contact fatigues (RCFs), rail gabs and crossing noses. The long wavelength defects are those associated with low frequency vibrations such as differential track settlement, mud pumping, bridge ends, stiffness transition zone, etc. Most previous studies into vehicle-track interactions were concerned only to a single one of the defect separately. This study is the world first to evaluate the coupling dynamic vehicle-track interactions over coupled short and long wavelength rail defects. The vehicle model has adopted multi degrees of freedom coupling with a discrete supported track model using Herizian contact theory. This paper highlights the nonlinear dynamic impact load factors experienced by railway track components due to wheel/rail contacts. The insight into the nonlinear phenomena will enable predictive track maintenance and risk-based track inspection planning adaptation to enhance public safety and reduce unplanned maintenance costs.

1 INTRODUCTION

There are two types of modern railway tracks suitable for various purposes of railway operations. A selection of trackform (ballasted vs ballastedless track) is commonly based on capital cost, lifecycle cost, asset management and operations (e.g. train speeds, axle load, and ride quality). For over 200 years, railway tracks have evolved. Considering the whole life cycle values, modern ballasted railway tracks have become the most efficient and effective infrastructure for the railway industry operating *below* 250 km/h of train speed for over centuries. The ballasted tracks have been tailored and optimised over and over; and they are often used in light rail tracks, metro networks, suburban rail network and intercity rail lines since they are relatively inexpensive and quite superior in terms of maintainability and constructability [1-4]. In contrast, ballastless tracks or concrete slab tracks are often utilised for highspeed rail lines (with train speed over 250 km/h) to reduce maintenance costs due to highspeed related swift degradation of traditional track components (e.g. accelerated densification and dilation of ballast, poor ride comfort due to differential track settlement/stiffness, ground-borne noise and vibration problems, etc.) [1, 5-9]. In practice, railway maintainers and operators are suffering from many of rail surface defects that lead to increased maintenance (either planned or unplanned), operational downtime and delay, more frequent monitoring and track patrol, and possibly the broken rails leading to train derailments [10-12].

Commonly, urban rail operators experience social problems of ground-borne, structure-borne noise and vibration induced by short and long wavelength rail defects. For instance, extra level of noise excited by rail squats (short wavelength type) was observed by residents near Woolloomooloo viaduct in Sydney, Australia [13-14]. Reportedly, the cost of rail replacement due to rail squats and studs has become a major part of the whole track maintenance cost in European countries [15]. Note that the rail squats and studs are typically classified as the growth of any cracks that has grown longitudinally through the subsurface. The subsurface lamination crack later results in a depression of rail surface sometimes called 'dark spot' [16]. There are two initiation types of the rail surface defects. Such rail defects are commonly referred to as 'squats' when they were initiated from rolling contact fatigue (RCF) cracks, and as 'studs' when they were associated with white etching layer due to wheel slips or excessive tractive effort [16].

Moreover, with heavier and faster trains, the dynamic load transferring on to track and its components such rails, sleepers, ballast and formations is higher and amplified by the traffic speeds and rail surface defects. These dynamic impact loading conditions often damage the track support and cause initial differential settlement and plastic deformation. The track issue does not stop here. Such the plastic deformation and initial differential settlement further form and couple with short wavelength defects (if any) to exponentially aggravate the dynamic loading condition [17-19]. Therefore, it is highly important to understand the coupled dynamic effect of rail defects on the rail infrastructure so that rail operators and maintainers can develop suitable cost-effective strategies for operations and maintenance. An example of strategies is to carry out preventative track maintenance (such as re-tamping, re-grinding and ballast cleaning when early sign of damage is inspected). In many regional railways (such as freight services), speed restrictions have been adopted to delay the maintenance regime when the rail defects exist. Note that these strategies are often called 'Base Operating Conditions (BOCs)' in railway industry practices. The BOCs have been developed from internal R&D activities and extensive empirical experience in

the rail industry over the centuries. On this ground, the established railway authorities of course do not lack of expertise and best practices. However, they might lose sight of nonlinear combined effects and might lead to inefficient and ineffective track maintenance if such effects are not being investigated or inspected timely. This current study will fill this gap by establishing a new insight into nonlinear phenomena of combined short and long wavelength rail defects so that the *safety-critical* track inspection regime can be effectively adapted.

Initially, the detailed modelling of rail track dynamic and wheel-rail interaction was studied in 1992 while D-track program for dynamic simulation also have been created by Cai for his PhD's thesis of Queen's University in Canada [20]. Afterward, Iwnick has done a benchmark, which was called Manchester Benchmarks in 1998 [21]. In 2005; Steffens [22] has adopted the parameter of Manchester Benchmarks to compare performance of vary dynamic simulation programs and also developed the user-interface of D-track. On the other hand, D-track had still an issue of lower result than others and then the owner has revised the program after this benchmark. Subsequently, Leong has done the Benchmark II with the revised version of D-Track in 2007 [23].

In this present study, the dynamic simulation concept by Cai [20] has been adopted as seen in Figure 1 since the track model has included Timoshenko beam theory for rail and sleepers, which enable a more accurate behaviours of tracks. Note that rail cross section and sleeper pre-stressing are among the key influences on shear and rotational rigidities of Timoshenko beam behaviors in numerical modelling of railway tracks [24-27]. The irregularity of wheel and rail will cause higher dynamic impact force that the design condition level or serviceability limit state. The exceeding magnitude of the force generated by wheel and rail irregularities will damage track components and impair ride quality [28]. This study thus is the first to present the wheel-rail dynamic forces over both single defects and coupled defects. The dynamic amplification factor will be highlighted to identify the effect of train speeds. The scope of this study will be focused on ballasted railway tracks. The commonly used 106t freight wagons will be modeled and coupled with the discrete supported track model. The track model will be based on a standard rail gauge (1.435m). The outcome of this study will help railway organization in improving the predictive maintenance and inspection regimes.

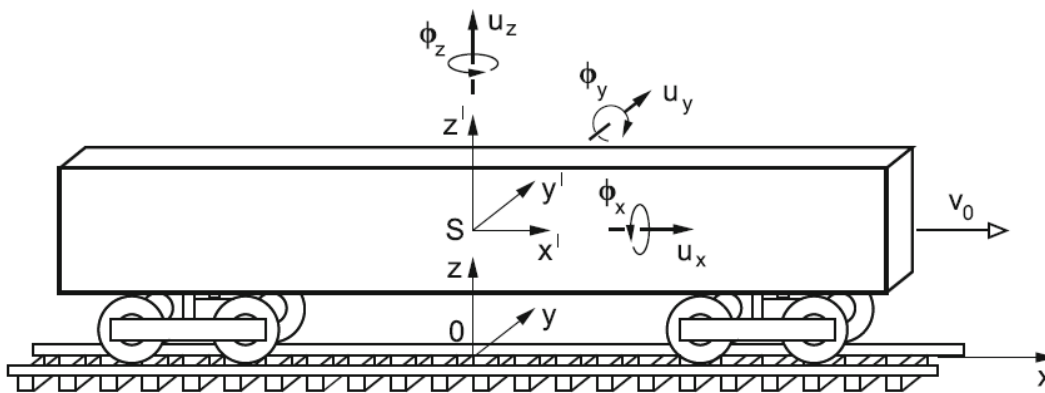


Figure 1: Coupling vehicle-track model [22]

2 DYNAMIC BEHAVIOURS

To design a ballasted railway track, it is important to assure safety and satisfy all the requirements of various stakeholders. The variety of dynamic loads from vehicle-track interaction and different environments make the rail infrastructure design highly complicated.

2.1 Dynamic wheel load

The dynamic loads induced by the vehicle moving on track structure will be considered as quasi-static force (equivalent to a static load times a factor) when the loads contain typically less than 10 Hz and is used to study behaviour of vehicle and design of the track foundation. In many cases, the perfect rail and wheel surfaces will be used to calculate of contact force and a factor (dynamic factor) to apply to the static wheel load and give a result for quasi static wheel load. The low-frequency load from a vehicle will apply to track geometry at a few cycles per second (up to 20 Hz).

In Australia, the most common method for calculation is presented by the Railway of Australia (ROA) manual or called “A Review of Track Design Procedures” or the “Blue Book”. The Dynamic Impact Factor (DIF) from this method ignores vertical track elasticity. The dynamic vertical wheel load (PD) is expressed empirically as a function of the static wheel load (P_s) where \emptyset is the Dynamic impact factor (always ≥ 1). For example, $PD = \emptyset P_s$.

2.2 Eisenmann’ Classical Formula

The Eisenmann formula is the most common method used for calculation the dynamic impact factor, used in a general track design. At the same time, the Eisenmann formula is modified by ROA and is used internationally. The Eisenmann formula (Eq. 1) and modified Eisenmann (Eq. 2) are shown respectively.

$$P_D = (1 + \delta\eta t) \cdot P_s \quad (1)$$

$$P_D = (1 + \delta\beta\eta t) \cdot P_s \quad (2)$$

where δ = Track condition factor; η = Speed factor, where $\eta = 1$ for speed (v) < 60 km/h and $\eta = 1 + [(v-60)/140]$ for $v > 60$ km/h; t = Upper confidence level (UCL) factor; $\beta = 1$ for loaded vehicles; and 2 for unloaded vehicles.

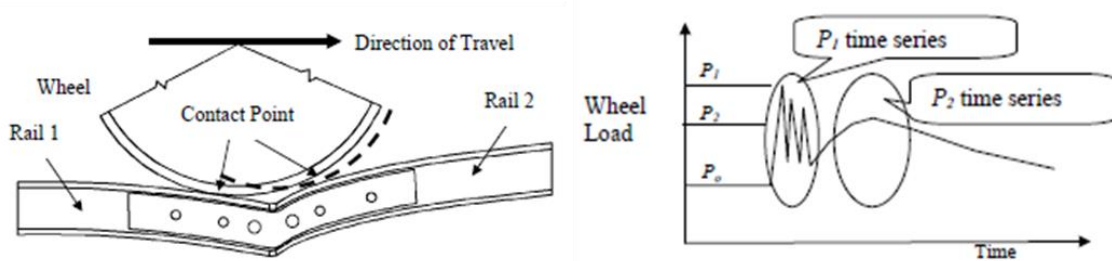


Figure 2: Impact force of wheel/rail contact at dipped rail joint [13]

2.3 Wheel trajectory over short wavelength rail defect

Dipped rail joint is termed to define the sum of an angle of dipped trajectory between each rail and the horizontal (in milli-radians). The two components of this angle consist of permanent deformation of the rail ends and the deflection of the joint under load. Jenkins et al. [29] described that the wheel travelling across a dipped rail joint creates the force peak as P1 and P2. The shape of the irregularity and characteristics of the vehicle create impact loading which the force at dipped joint increases almost linearly with the speed and angle of the dip. When the trains travelling at high speed approach a rail joint, the wheel lose will lose contact with the railhead of rail and land on the connected rail that generate the high dynamic impact force as illustrated in Figure 2.

The P1 force is a very high frequency ($\cong 200$ Hz -1000Hz) that is less than 0.5 millisecond in length (0.25 - 0.5 millisecond after crossing the joint). The compression of contact zone between wheel and rail create the inertia of rail and sleepers which does not directly transform to ballast or subgrade settlement. However, it has a significant effect on wheel/rail contact force. The P2 is occurred at a lower frequency range ($\cong 50$ Hz -200Hz) than P1 occurring much later at typically 6 – 8 milliseconds. The unsprung mass and the rail/sleeper mass are moving down together influencing the compression of the ballast below the sleeper. P2 forces therefore increase the contact stresses and also induce the loads on sleepers and ballast. P2 force will be considered mostly by the track design engineer. Jenkins et al. [29] provided a method of calculation as follows:

$$P_1 = P_0 + (2\alpha v) \cdot \sqrt{\frac{k_H m_e}{1 + \frac{m_e}{m_u}}} \quad (3)$$

$$P_2 = P_0 + (2\alpha v) \cdot \sqrt{\frac{M_u}{M_u + M_t}} \times \left[1 - \frac{\pi C_t}{4\sqrt{K_t(M_u + M_t)}} \right] \times \sqrt{K_t M_u} \quad (4)$$

where: P_1 and P_2 = dynamic rail force (kN); P_0 = vehicle static single wheel load (kN); k_H = a chord stiffness to the Hertzian contact stiffness; m_e = the effective track mass (kg); m_u = the vehicle unsprung mass (kg); 2α = Total joint angle (rad); v = speed of vehicle (m/s); K_t = equivalent track stiffness (MN/m); M_t = equivalent track mass (kg); and C_t = Equivalent track damping (kNs/m).

2.4 Track settlement

Track settlement can cause the train passing the track with higher dynamic load increasing high-frequency variations to the ballast and subgrade. It will then cause non-elastic or plastic deformations with permanent setting. In normal situation, the track will generally not return to the same position but to the very close point (accumulated deformation). As time passes, all of non-elastic deformations will create a new track position and this phenomenon becomes

differential track settlement. The track alignment and surface level of track also change due to the accumulated non-elastic deformations. The irregularity of track will increase low-frequency oscillation of vehicle. However, the track settlement often takes place at the transition area to a bridge. In addition, the quality of ballast, sub-ballast and the subgrade are also the factor inducing permanent deformation [30-31].

Track settlements typically consist of two phases. The first phase is after tamping that the gap between ballast particles is reduced quickly and so this layer is consolidated. The second phase is slower while the densification and inelastic behaviour of the ballast and subgrade materials are concerned as the two majors parameters influencing the ballast settlement that are the deviatoric stress, vibrations, degradation and subgrade stiffness. The empirical settlement equation for the substructure is shown below. It does only consider the ballast settlement not including subgrade settlement:

$$Z_{iN} = Z_{i0} + f(\log N, \sigma_{be}, I_{dyn}, I_{dec}, I_{Esub}) \quad (5)$$

This equation describes the settlement of ballast below the sleeper I where: Z_{i0} = the given void amplitude; N = Number of load cycles; σ_{be} = the vertical equivalent stress in the ballast layer; I_{dyn} = the dynamic factor; I_{dec} = the degradation factor; and I_{Esub} = the subgrade stiffness factor.

3 COUPLING VEHICLE-TRACK MODELLING

The track model (D-Track) is simulated on Winkler foundation principal which only cross-section of track dynamic responses is considered symmetrically. Rail and sleeper were represented on elastic beam of either the Timoshenko type. The sleepers also support the rail as discrete rigid masses. Free-body diagram of track model are shown in the Figure 3(a) where $P(t)$ is a moving wheel force at constant speed (v). Figure 3 (b) represents the force from rail to sleeper through the rail seat (i^{th}) and reaction force $k_s z_i(y, t)$ per unit length.

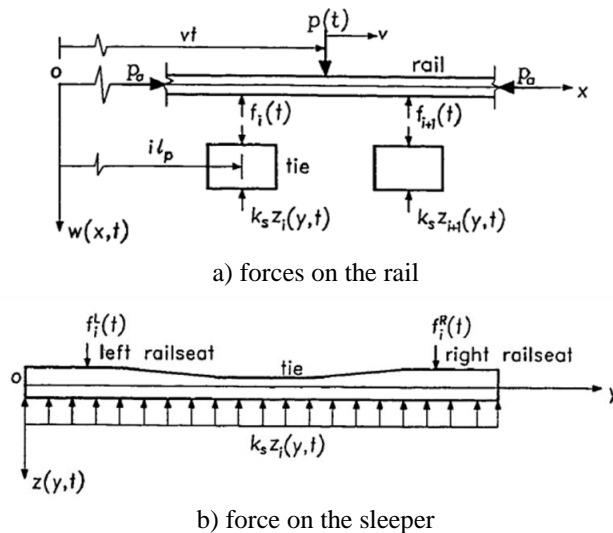


Figure 3: Free-body diagram of track model [22]

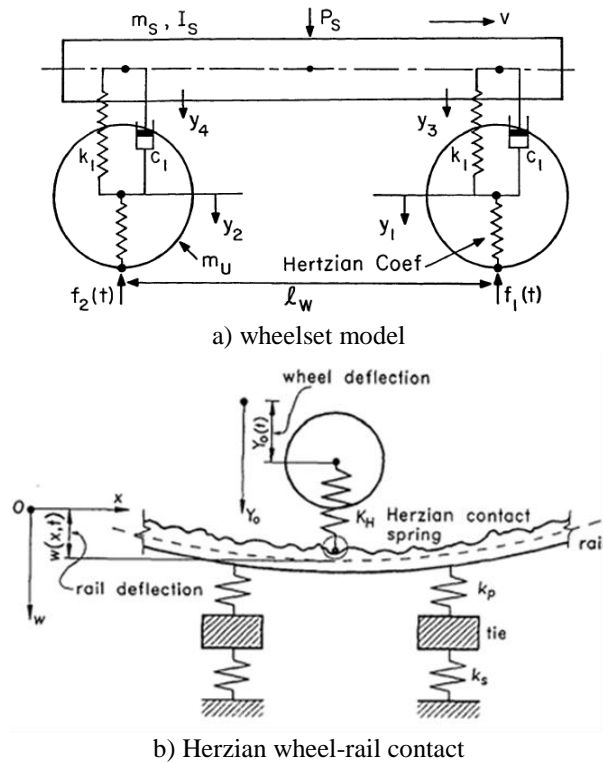


Figure 4: Free-body diagram of vehicle-track model [22]

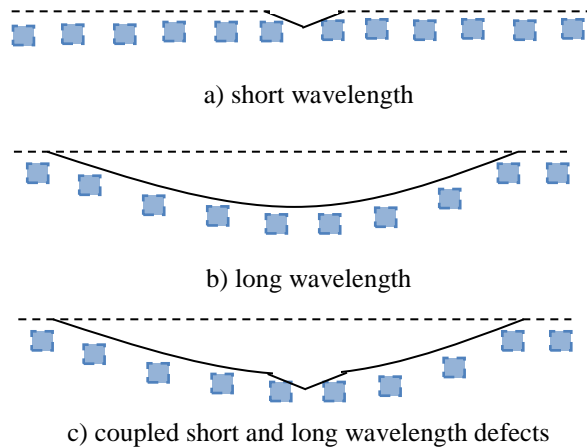


Figure 5: Rail irregularity models

The wheelset model in this modelling consists of a four-degree of freedom which include of one bogie with two-axle, rail and track. The wheelset model uses the unsprung masses (μ) and the sideframe mass (m_s, I_s) to calculate on one rail through the primary suspension (k_1, c_1) as shown in Figure 4(a). The components of vehicles are demonstrated as a spring load by using Hertzian contact model. Moreover, the equations of motion in this model used the principles of Newton's law and beam vibration to apply. Integration between wheelset and track equations can be calculated by the non-linear Hertzian wheel-rail interaction model as illustrated in Figure 4 (b). The D-Track model has been benchmarked by [23, 24] in order to assess the accuracy and

verify the precision of numerical results. D-Track is thus adopted for this study. Rail surface profile irregularities can be estimated as inverse half-sinusoidal curve as shown in Figure 5(a). For track settlement, the study focus on a long span defect (such as large mud pumping tracks) with a wavelength of 10m as shown in Figure 5(b).

4 RESULTS AND DISCUSSIONS

The numerical simulations have been carried out using 106t freight wagon with wheel radius of 0.46m and Hertzian spring constant of $0.87 \times 10^{11} \text{ N/m}^{3/2}$. The dynamic wheel/rail contact forces can be seen in Figure 6.

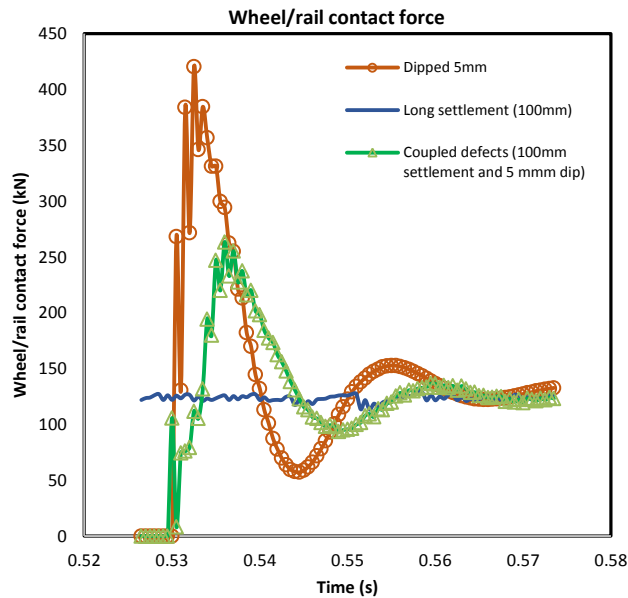


Figure 6: Wheel/rail contact force

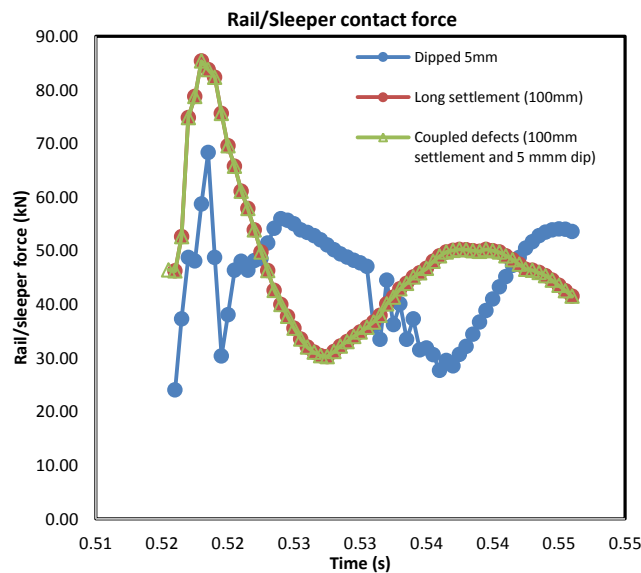


Figure7: Rail/sleeper contact force

Short wavelength defects have significant effects on the wheel/rail contact force in comparison with long wavelength defects. In contrast, surprisingly, it is apparent that long wavelength defect plays higher influence on the dynamic railseat loads (rail/sleeper contact) as shown in Figure 7.

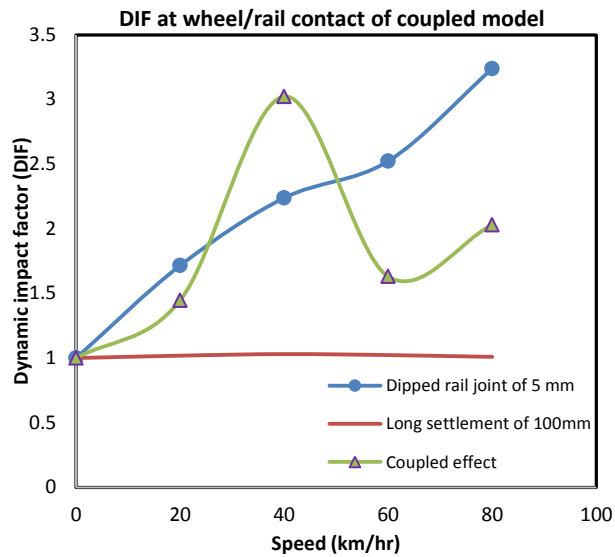


Figure 8: DIF at wheel/rail contact force

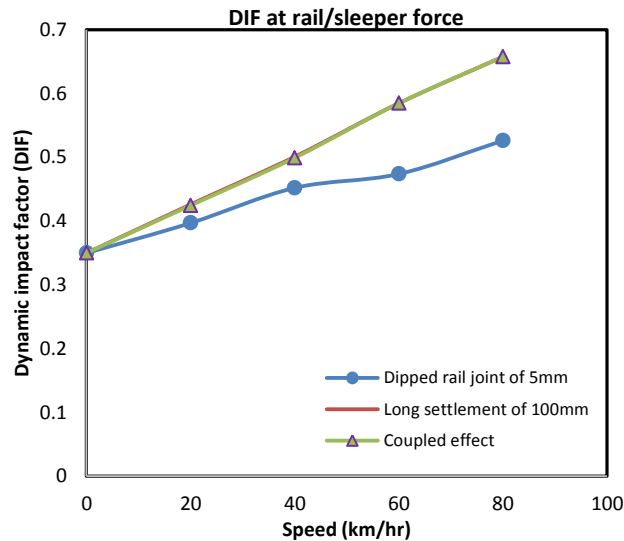


Figure 9: DIF at rail/sleeper contact force

The dynamic impact factor (DIF) can be evaluated from dynamic over static wheel force ratio. Figures 8 and 9 demonstrate the coupling dynamic vehicle-track interaction on contact forces and railseat loading conditions. It is clear that the coupling wheel/rail contact force is nonlinear (Figure 8) whilst rather linear relationships can be observed on the load sharing (Figure 9). It is clear that the individual short rail surface defects influence significantly the high frequency force at wheel/rail contact. However, the coupled short and long wavelength defects does not seem to play a strong influence on wheel/rail contact, but instead influence how the load is distributed on the sleepers.

5 CONCLUSION

This paper presents the nonlinear dynamic interactions between vehicle and track, which can cause vibrations and acoustic radiations to railway neighborhood. The nonlinear effects of coupled short and long wavelength rail defects on vehicle-track interaction have been highlighted. It is the first to evaluate the coupling dynamic vehicle-track interactions over coupled short and long wavelength rail defects. The results show that the coupled effect plays a key role on magnifying the rail/sleeper contact forces (railseat loads). The insight implies that sleepers will experience excessive dynamic loads and they can deteriorate and abrade at a relatively faster rate. More results on parametric effects of short and long wavelength as well as the effect of defect severity will appear elsewhere in the near future. This understanding into the nonlinear phenomena will help track engineers to manage and operate infrastructure assets more effectively.

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