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A Dynamic Wheel-Rail Impact Analysis of Railway Track under Wheel Flat by Finite Element Analysis

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A Dynamic Wheel-Rail Impact Analysis of Railway Track under Wheel Flat by Finite Element Analysis

Abstract Wheel–rail interaction is one of the most important research topics in railway engineering. It involves track impact response, track vibration and track safety. Track structure failures caused by wheel–rail impact forces can lead to significant economic loss for track owners through damage to rails and to the sleepers beneath. Wheel–rail impact forces occur because of imperfections in the wheels or rails such as wheel flats, irregular wheel profiles, rail corrugations and differences in the heights of rails connected at a welded joint. A wheel flat can cause a large dynamic impact force as well as a forced vibration with a high frequency, which can cause damage to the track structure. In the present work, a three-dimensional (3-D) finite element (FE) model for the impact analysis induced by the wheel flat is developed by use of the finite element analysis (FEA) software package ANSYS and validated by another validated simulation. The effect of wheel flats on impact forces is thoroughly investigated. It is found that the presence of a wheel flat will significantly increase the dynamic impact force on both rail and sleeper. The impact force will monotonically increase with the size of wheel flats. The relationships between the impact force and the wheel flat size are explored from this finite element analysis and they are important for track engineers to improve their understanding of the design and maintenance of the track system.

Keywords: wheel-rail force, impact analysis, finite element, wheel flat, sleeper and vibration frequency.

1: Introduction

Wheel–rail interaction is one of the most important research topics in railway engineering. It relates to safety of the track. The impact forces can be extremely large and can cause serious failures in the track structure, which can lead to significant economic loss for track owners through damage to rails and to the sleepers beneath. The wheel–rail impact forces occur by reason of imperfections in the wheels or rails such as wheel defects, rail defects and differences in the height of rails connected at a welded joint. These situations can cause the wheel-sets impact onto the track, thereby causing an impact force and a forced vibration with high frequencies. These defects are normally caused by wheel–rail friction during braking and the rolling of wheels over long periods of time. They can also be caused by problems in rail joints. It is quite hard to avoid wheel–rail impacts because wheel and rail defects normally cannot be detected in the beginning stage when the flat occurs, although eventually the flat can be detected by the equipments [24]. Therefore, the consideration of wheel–rail impact forces in track design and maintenance is critically important.

In recent years, many researchers have focused on the exploration of dynamic impact forces induced by wheel–rail defects. A number of models have been developed [1] [2] based on theoretical and traditional methods such as Beam on Elastic Foundation (BOEF) in the analysis [3] [4]. However, investigation of the impact forces on sleepers due to wheel–rail impact forces is a complex non-linear dynamic problem because of the non-linearity of materials and the dynamic contact. Consequently, previous theoretical analyses have had to employ significant simplifications in their assumptions about the behaviour of various track and vehicle components. For example, traditional impact analysis always assumes some bodies are rigid or under small deformation, or they use Hertzian contact theory based on a half-space assumption and a linear material model.

An alternative method for solving such dynamic impact problems is numerical modelling and simulation, for which the finite element method (FEM) is the most widely used. Researchers have developed a number of finite element models for railway track force analyses, such as a wheel–rail system for different heights at a rail joint [5]; a model for the static analysis of entire track structures [6] [7]; static-state and steadystate analysis of wheel–rail interaction [9] and the analysis for predicting wear on wheel profiles [8] [10] [11] similar to rolling simulation [12]. Furthermore, a two-dimensional FE track model based on BOEF analysis to investigate wheel flat impact has also been developed by [13]. The simulations mentioned above have achieved useful results, thus they demonstrate the advantages of FEM for solving non-linear problems in the track structure. Previous research has focused on the magnitudes of forces in track structures and on force variations in time or distance domains. Some parametric analyses have been conducted but these have been done using traditional methods. Most importantly, there is no comprehensive study of the effect on the dynamic impact force by the wheel-flat based on real situation. The present work is focused on: the relationship between wheel flat size and wheel–rail impact force; the relationship between wheel flat size and impact force on sleepers; and the relationships between wheel flat size and frequencies of induced vibration on both rail and sleepers. In reality, train speed and brake force vary and cause different wheel flat sizes. Thus, the wheel–rail impact force varies. Track engineers evaluate the track capacity, life span and categorize the track uses often through the impact force on the track. Therefore, these relationships are important and helpful for improving track design and methods of maintenance.

For this analysis, experimental data is hard to obtain and is usually only available for particular conditions of the track being tested. The FEM is the best choice for such a complicated non-linear analysis. In previous studies, the only finite element model which applied to wheel flat impact analysis was two-dimensional and was still based on BOEF. The wheel flat impact effect was assumed as a mass dropped onto the track, and it was assumed that the displacement of the wheel-set complied with the Harvesine function [14]. Recently, Some of 3-D FE models have been developed for wheel-rail impact analysis, such as a 3-D FE model of wheel-track system for wheel-rail impact analysis under rail defects [22] [25], and these models have been validated by comparing the simulation results with results by theoretical study and measurements [22] [23] [24]. However, in previous studies, there was no 3-D FE model of wheel-track systems based on real situations developed for wheel-rail impact analysis and rail-sleeper analysis under wheel defects. Therefore, if results which are closer to actual situations are to be achieved, rather than base the entire analysis on the BOEF principle, researchers must develop a 3-D FE model for wheel-track system impact analysis considering wheel flat. In this model, the wheel–rail interaction should be considered, the proper configurations of important track structure components should be applied, and the material properties should be non-linear.

In this paper, based on ANSYS, a 3-D finite element model for the study of wheel flats is developed. This model includes a wheel, a rail, three rail pads, three sleepers, a ballast layer, a capping layer and a formation layer. The model considers wheel–rail dynamic contact, the non-linearity of materials and treats all track components as deformable bodies. Thus, the simulation in this paper is more realistic than the simulations in previous models. This FEA model is firstly validated both theoretically and numerically. The effects of the wheel flat on wheel–rail impact force and on vibration frequency are then investigated. Based on the model, the relationships between the impact force and the wheel flat size are also thoroughly explored.

2: Relevant theoretical study

This study analyses the wheel-rail impact force induced by a wheel flat, and the principle of rolling impact is presented in Figure 1. The wheel-set drops for a distance from point O1 to O3 and to induce an impact on top of rail at point B.

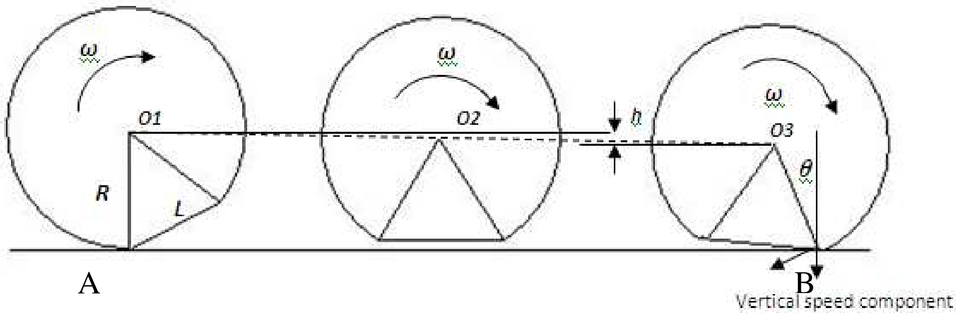

Figure 1. The principle of wheel–rail impact induced by wheel flat

If assuming the wheel flat length *L*, and wheel radius *R* are given as constants then the equation of motion can be derived as expressed by:

$$\omega\sqrt{\frac{2h}{g}} = 2arcsin\frac{L}{2R} - arccos\left(1 - \frac{h}{R}\right) \qquad (1)$$

Where *h* is the dropping distance of wheel-set; *L* is the length of wheel flat; *R* is the rolling radius of wheel; *ω* is the rotation speed and *g* is the gravity acceleration. Furthermore, according to Figure 1, the impact is not only from the wheel-set dropping on track, but also from the vertical speed component induced by wheel rotation, which was not considered in previous work. The vertical component cannot be ignored, because the rotation speed of the wheel is significant and it can induce high vertical impact forces. The vertical speed component from the wheel rotation is expressed by:

$$v_{ver.} = \omega R sin\theta \qquad (2)$$

Where *θ* is the angle which as shown in Figure 1, it is derived from the dropping distance *h*, which is expressed by:

$$\theta = arccos\,(1 - \frac{h}{R}) \qquad (3)$$

According to the theory of dynamic load [21], the wheel-rail impact force that is induced by the wheel flat at point B can be expressed by:

$$F_{im} = Q_w\left(1 + \sqrt{1 + \frac{2h + \frac{v_{ver.}{}^2}{g}}{Q_w}\frac{K_p K_b}{K_p + K_b}}\right) \qquad (4)$$

Where Q_w is the static wheel load which derived from the total mass of the vehicle, K_p and K_b are the stiffness of rail pad and railroad bed (inc. ballast, capping and formation layers), respectively.

Equation (1) is a transcendental equation, which can be solved by a numerical method. Although the wheel-rail impact force can be calculated by Equation (4), in derivation of Equation (4), significant simplifications have been used including ignoring the flexibility of wheel and rail and simply treating the railroad bed as an elastic foundation. The solution to the equation is a statics solution, which fails to reflect the dynamic properties of the impact force, but it can be employed as a reference solution to validate the FEA model which will be described in the next section.

3: Finite element model

Because wheel–rail dynamic interaction is a complicated problem and the components of the wheel–rail system have complicated shapes, this analysis will only focus on the wheel–rail dynamic impact force. The railway track structure used in this model is shown in Figure 2.

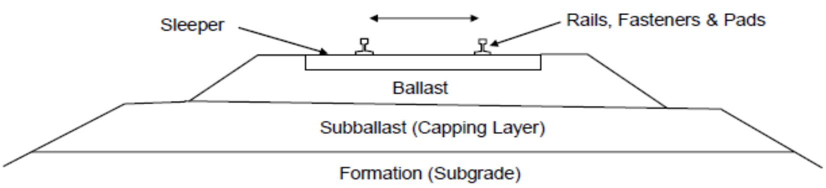

Figure 2. Diagram of railway track structure [15]

The completed model of the wheel-track system for this case includes a wheel, a rail, three rail pads, three sleepers and a ballast layer as shown in Figure 2. All component models were created in the DesignModeller application of the ANSYS Workbench platform. This system was used to simulate the general heavy-haul train that is used in

Queensland Rail (QR) coal transport. According to the data recorded in [16], the speed of the QR heavy haul vehicle was 72km/h, with a narrow track gauge of 1067mm, with a 685mm sleeper spacing. The rail is the Australian Standard 60kg/m (AS60) rail, the pads are made of High Density Polyethylene (HDPE), and the sleepers are narrow-gauge QR concrete sleepers. All relevant vehicle parameters, properties of track and sub-track components are available in [15].

The wheel was assumed to be a cylinder with the appropriate radius and thickness. The rail was created by extruding a cross section of Australian Standard (AS) 60kg/m rail. According to an analysis of wheel load distributions using the theory of a discrete supported long beam [3] as presented in Figure 3 and Table 1, installing three sleepers in the model is the most appropriate choice because it has the acceptable accuracy without too expensive computational cost. The three sleepers used in the model were created by extruding three cross sections of a standard gauge concrete sleeper in the appropriate positions. Three rail pads were created corresponding to the number of sleepers. The rail pads were created as elastic boxes with appropriate dimensions. The ballast layer was created using the shapes shown in the diagram of the track structure in Figure 2. The dimensions can be found in [16], and the main parameters are presented in Table 2. In this FE model, the fastenings are ignored because this study focuses on the global responses for the real system. The function of fastenings is to keep the rail, rail pads and sleepers connected together without relative motions. To simplify the calculation, a contact has been applied in order to replace the fastenings functions. Because the model is axially symmetrical, only a half of the entire track structure was considered.

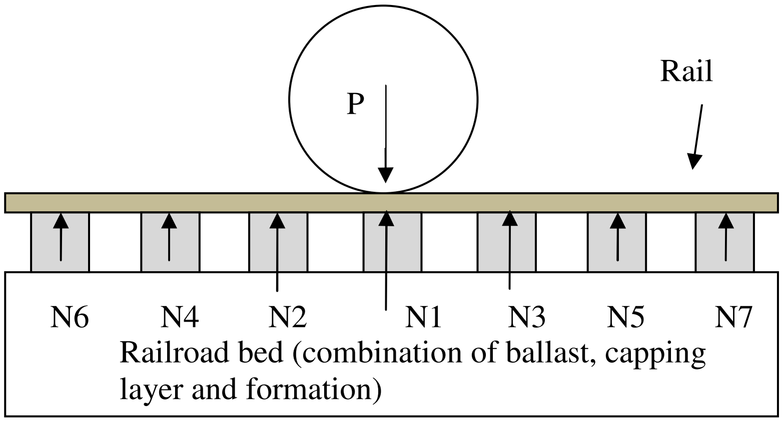

Figure 3. The theoretical analysis of wheel load distribution on sleepers

Table 1. Analysis of load on sleepers according to Figure 3

S/S, mm	***W/L***, kN	***N*1**, kN	***N2***, ***N3***	***N4***, ***N5***	***N6***, ***N7***	***N1-3***	***N1-5***	***N1-7***
685	128	72.9	31	0.8	-3	134.9	136.5	130.5

Where, S/S – sleeper spacing, W/L – wheel load, *N1-3* – total of *N1* to *N3*, *N1-5* – total of *N1* to *N5*, *N1-7* – total of *N1* to *N7*.

Table 2. The basic parameters of track components [16]

Basic parameters	
Wheel	Radius: 457.5mm; thickness: 72mm
Sleeper	Top width: 220mm; bottom width: 250mm; height: 208mm; length: 2200mm
Rail pad	Length: 180mm, width: 150mm, thickness: 7.5mm
Ballast	Top length: 1356mm; depth: 250mm
Sub-ballast	Top length: 2596.2mm; depth: 150mm
Formation	Depth: 1000mm

The materials of wheel and rail were defined as structural high strength steel; the sleepers were concrete; the ballast was defined as the material created by the properties described in [16]. The complete model with FEA meshes is shown in Figure 4.

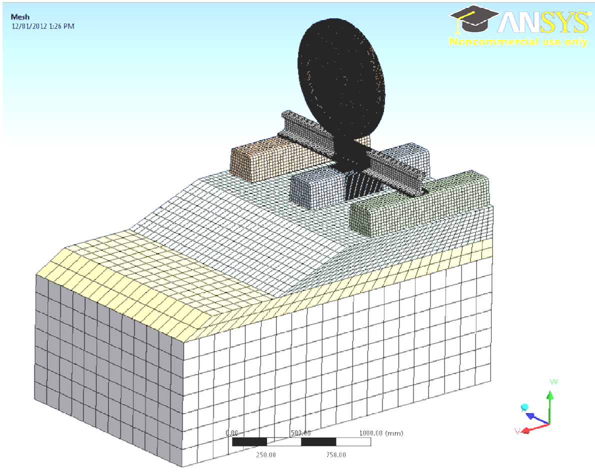

Figure 4. The simulation model of wheel-track structure

To analyze the impact force caused by wheel–rail interaction in a wheel flat situation, a flat surface 50mm in length was created on the wheel tread and its location is presented in Figure 5. As the wheel rolls around, the impact force caused by the flat surface is transferred from the top of the rail to the top of the tested sleepers and then into the ballast and formation layers.

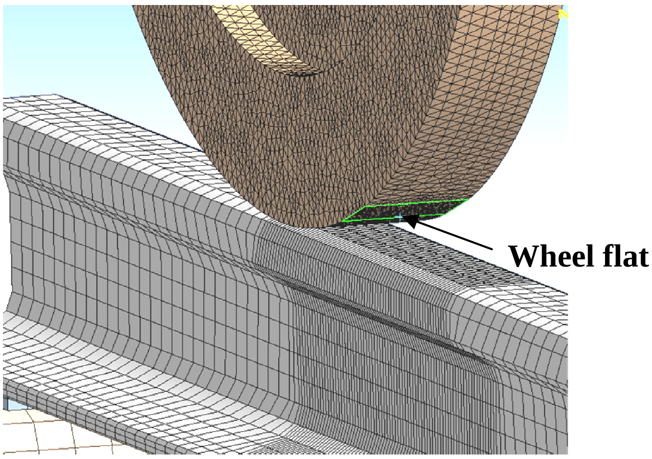

Figure 5. The Location of the wheel flat surface and the mesh detail

The entire model was meshed variously in different areas to ensure the results were as accurate as possible. The contact areas of wheel-rail and rail- sleeper are the areas where stress is concentrated. So, both were fine meshed, as shown in Figure 5. The remaining areas of the wheel, rail and sleepers were medially meshed; the ballast layer, capping layer and formation layer were coarsely meshed. All components in the model were meshed by 3-D solid elements. Therefore, the minimum finite-element mesh dimension was 5mm in the wheel flat area, as well as in the impacted area of the rail and the area of the middle sleeper directly below the impacted area of the rail because the middle sleeper was the sleeper of interest. The model consists of 109,148 nodes and 302,031 elements.

The applied conditions included defining types of contact areas, boundary conditions and applied loads. In this model, only the wheel is moving, the other components are in constant contact with each other. Because this study mainly aims to analyse the vertical wheel-rail impact force, the relative motions between wheel and rail do not significantly affect the analysis. To save computational cost, the relative motions are ignored. Therefore, all components in the entire track structure are bonded. The contact area between the wheel and the top of the rail is also bonded to ensure the wheel rolls on the rail without sliding.

The load in this model is the static vehicle weight applied as a constant force to the wheel which rotates at a given angular velocity. The static force on the wheel is the equivalent load converted from half of the axle load from gross mass of the car body, bogie frame and wheel shaft. According to the parameters of the heavy haul vehicle in [16] the converted force is approximately 127.9kN. The running speed of the heavy haul vehicle is 72km/h which, converted to an angular velocity of the wheel is 43.48rad/s.

4: FE model validation

An important consideration when using an FE model to do experiments for research instead of unavailable experimental devices is whether the results are valid representations of reality. Thus when using the model, it must be proven that the values from the simulations are representative. Therefore, the newly developed FEA model in Section 3 will be validated in the following section.

4.1: Validation with a perfect wheel by valid simulation

To validate the FEA model, the FEA results are firstly compared against those obtained from a software package named DTrack that was used for track dynamic analysis. DTrack is a software developed for track dynamic analyses based on DARTS which has been developed by Queens's Unversity back to 1992 [17]. It is not commercialised software, but its simulation results have been validated in the Manchester benchmarks recorded in [18] [19] and later described in [15]. DTrack has also been validated through an international benchmarking exercise against six widely used models, such as NUCARS, and against measured track data [16]. That means it is suitable to track dynamic analysis. Therefore, it is reasonable to employ DTrack as a reference in this validation.

To validate the numerical model, a base case of a perfectly round wheel was chosen. The relationship between the wheel-rail contact force and time is shown in Figure 6. The variation of the force with time is due to the wheel rolling over the spaces between sleepers as well as vibration of the rail and, to a lesser extent, to the railroad bed. The mean value of the trace is the 'quasi-static force' [20].The wheel–rail forces from the FE model are shown in Figure 6 with an additional (dotted) line representing the output from the DTrack model for the short section of track simulated in the FE model. It can be found from Figure 6 that the trace of the DTrack output and the moving mean of the FE model's trace have good correspondence between each other. Hence, the FEA model is validated.

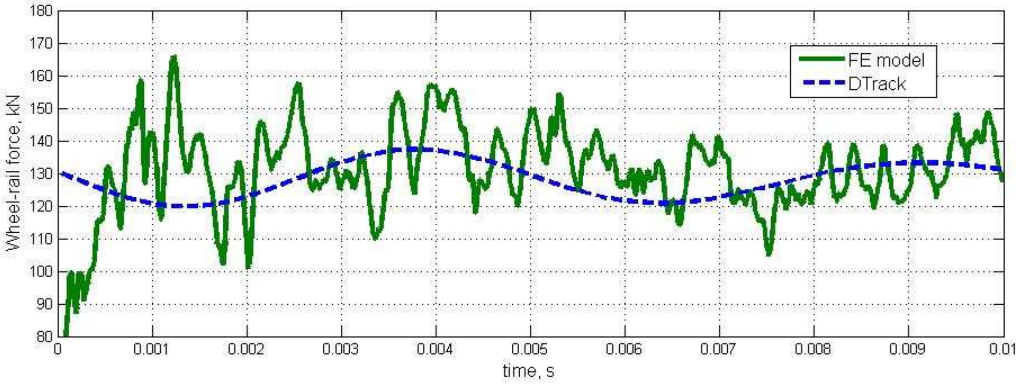

Figure 6. Simulation results for wheel–rail contact force from FE model and from D-Track

It should be noted here that Figure 6 shows that, at the beginning the force increases from zero and varies at a high frequency because the ANSYS dynamic simulation requires the model to commence with zero wheel load and zero speed but then it applies the specified load and speed immediately – as shown in the first 0.001 seconds in Figure 6. The DTrack model's simulation results in Figure 6 commenced before the zero time point in the figure and so no such transition is shown for its trace. The FE trace in Figure 6 suggests a high frequency vibration of the wheel on the rail of about 6 kHz. This frequency is much higher than frequencies measured with normal track instrumentation or modelled in dynamic packages such as DTrack. Although the vibration diminishes a little with time in Figure 6, it doesn't disappear which suggests that it is a vibration which may not have been considered or detected previously.

4.2: Validation with a wheel-flat by theoretical analysis

As demonstrated in [23], the FE model can be validated by comparing the FE model results with appropriate theoretical solution. From the solution of theoretical analysis that described in Section 2, the achieved values are the dropping distance h and the impact force F_{im}. However, the results which can be obtained from FE model are the impact force F_{im} and dropping time t (from A to B as shown in Figure 1). The dropping time t from theoretical analysis can be derived from h by the formula $h=g*t^2/2$. This dropping time is calculated for the first flat impact only. As shown in Figure 6, before the impact due to the flat, the track is under vibration but with relative smaller amplitude. To simplify the theoretical calculation, the spring-back of the rail is ignored and the equilibrium position is considered. For this case, the wheel flat size is 50mm, according to the relevant parameters presented in Table 2, the t and F_{im} can be obtained, and compare with the result from FE model as (shown in Figure 7 later) is presented in Table 3.

Table 3. The comparison between results from theoretical analysis and FE model

	Theoretical result	FEA result
Dropping time, ms	2.3	1.9
Impact force, kN	383.8	414.5

According to the comparisons in Table 3, the FEA result of impact force has only 8% difference to the theoretical results, and the dropping times calculated by these two methods are also very close. Therefore, it also validates the developed FEA model.

5. Results and discussion

The FE model, which was developed in Section 3 and validated in Section 4, was used for the analysis of wheel–rail impact forces on the track. The wheel–rail impact forces induced by wheel flats of various sizes on both the top of the rail and the sleeper were investigated. In addition, a wheel–rail impact force induces vibration in the track structure and this vibration is transmitted to all the components of the system. In this analysis, both vibration frequency and the amplitude of the dynamic impact force are analyzed.

5.1: Wheel-rail Impact force

In this case, a 50mm flat was applied to the wheel. The dynamic impact force induced by the wheel-flat is plotted in Figure 7. For the comparison, the results for a case with the perfect wheel (i.e. no flat) are also plotted in the same figure. It should be noted that as discussed in Section 3, the force values plotted in Figure 7 include the statics loading of 128kN applied on the wheel axle (including weight of wheel and presented by the straight line).

From Figure 7, it can be found that at the beginning of the trace, the high frequency 6kHz wheel–rail dynamic force (the increment from statics loading) induced by the smooth portion of the defected wheel is around 24kN, which is identical to dynamic component of the force induced by the perfect wheel until approximately 0.0057s, because the wheel flat surface was not interacting with rail during this period. After 0.0057s, the wheel starts to separate from the rail at the flat surface because of the inertia of wheel until at 0.0078s when the wheel impacts on the top of the rail. The wheel-rail force is reduced to zero during the separation from rail and increases rapidly at the moment of impact. The impact force induced by the wheel flat is dramatically greater than the dynamic component induced by a perfect profile. According to Figure 7, the incremental impact force from the flat is approximately 292kN – more than twice the wheel static load which is around 128kN. In summary, this flat profile significantly increases the wheel–rail contact force

In addition, from Figure 7, we can also find that the wheel flat impact induces a forced wheel-rail vibration after impact, with a frequency of wheel-rail force of approximately 1.1kHz.

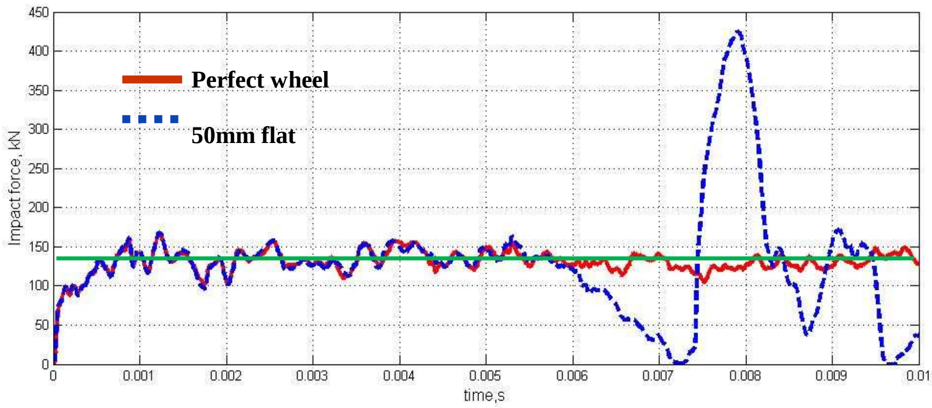

Figure 7. Comparison between wheel–rail contact forces induced by a perfect wheel and a wheel with a 50mm wheel flat

5.2: Relationship between dynamic impact force and wheel flat size

To investigate the effect of wheel flat size on the impact force, flats were studied with sizes of 30mm, 40mm, 50mm and 60mm. Their results are presented in Figure 8.

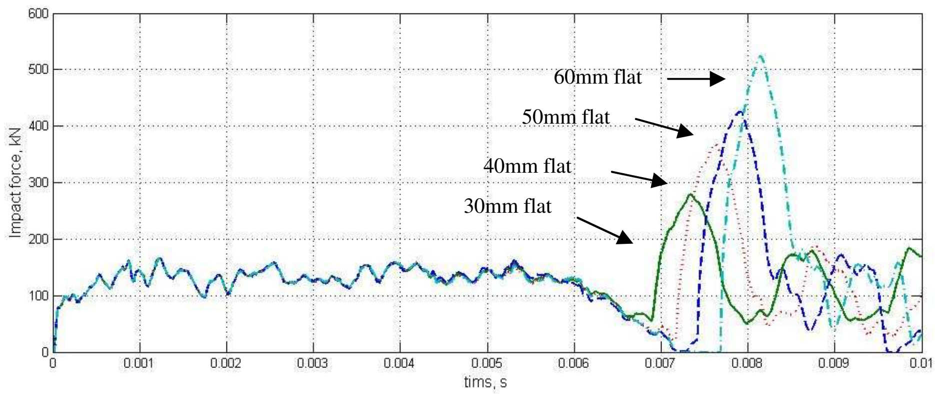

Figure 8. Wheel–rail impact forces induced by 30mm, 40mm, 50mm and 60mm wheel flats

As shown in Figure 8, it was found that the larger the wheel flat size, the greater the impact force. In addition, for a longer flat, the impact starts at a later time, because the flat length determines the drop distance and time of wheel flat impact on the rail, as shown in Figure 1.

The wheel–rail impact force was transferred to the sleeper through the rail pad at the rail seat and the induced impact forces on the sleeper are shown in Figure 9. The impact force on the sleeper has similar characteristics to the impact force on the wheel-rail, but it has more complicated oscillation property.

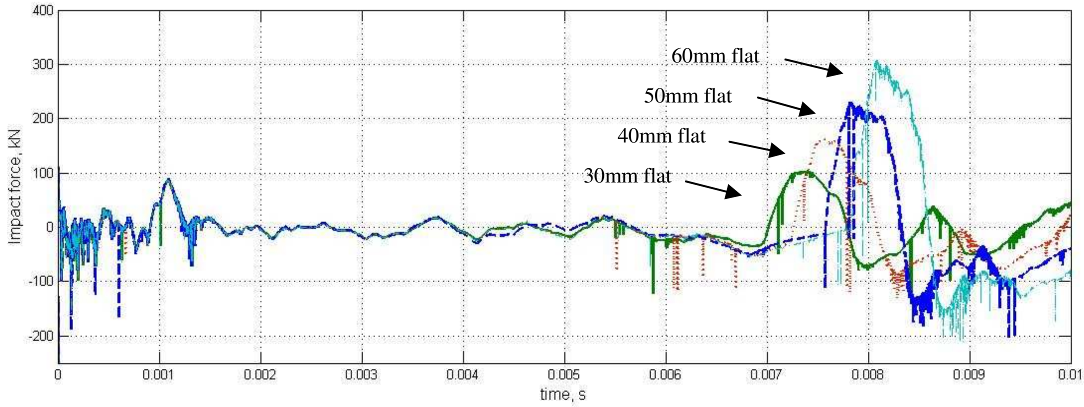

Figure 9. Incremental impact forces on sleeper by 30mm, 40mm, 50mm and 60mm wheel flat sizes

The detailed results are summarized in Table 4. Note that the impact forces listed in this table do not include the static force on the wheel.

Table 4. Incremental wheel-rail and sleeper impact forces and frequencies induced by various wheel flat sizes

Wheel flat size,	Wheel–rail incremental	Frequency of wheel–rail impact	Incremental impact force on sleeper, kN	Frequency of sleeper impact
30	151	714.3	110.5	666.7
40	240.3	869.6	162.2	740.7
50	292	1052.6	230.5	833.3
60	396.3	1176.5	307	909.1

The relationships between wheel–rail force and wheel flat size, and between sleeper force and wheel flat size are shown in Figure10. They were found to be non-linear and monotonically increasing. In Figure10, the wheel-rail impact forces between 30mm and 60mm present as a curve, whereas the sleeper impact forces follow an almost straight-line trend. Therefore, for track design purposes, it is reasonable to use a straight line to describe the relationship between sleeper impact forces and the wheel flat sizes. The specific form of these relationships will of course change depending on the rail, pad, sleeper and ballast characteristics of a given track.

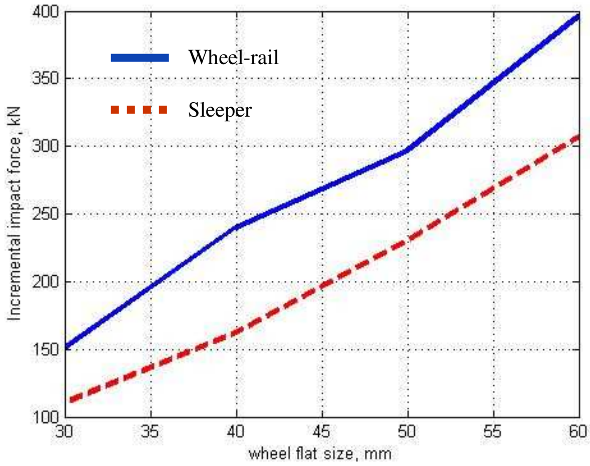

Figure10. Relationship between incremental impact force and wheel flat sizes for wheel-rail (solid line) and sleeper (dashed line) at the rail seat

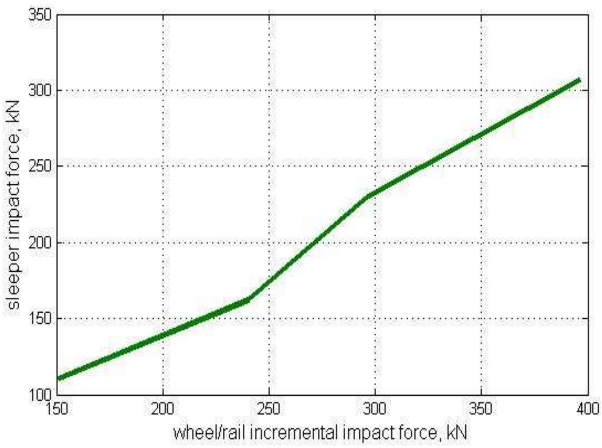

Figure11. Relationship between wheel-rail incremental impact force and impact force on a sleeper at the rail seat

The relationship of the incremental impact force on the sleeper and the incremental wheel-rail impact force was found based on values listed in Table 1, and is presented in Figure11. As shown in this figure, the impact forces on the sleepers are always smaller that the relevant wheel-rail forces. This is easy to understand because only a partial impact forces are transferred from track to the sleepers. From Figure11, we can also find that the relationship presents as a curve rather than a straight line which is traditionally assumed in the design of the track-sleeper system. Such non-linearity could be due not only to the inclusion of non-linear materials properties in the track model, but also to the dynamic interaction of track components. Under the dynamic impact, the track and the sleeper will vibrate at their own natural frequencies. Their different natural frequencies lead to their motions sometimes augmenting each other, and sometimes counteracting each other. It is also interesting to find that in the middle segment. i.e., when the wheel-rail impact force increases from 245kN to 300kN, the sleeper impact force grows faster than it does in other stages. This implies that more attention should be paid in the design of the sleeper for this case if the wheel-rail impact force is between 245kN to 300kN.

5.3: Relationship between impact force frequency and wheel flat size

As discussed above, the wheel flat induces a significant change in the dynamic impact forces which have complicated periodic properties, as shown in Figure 8 and Figure 9. Therefore, the presence of a wheel flat will change not only the amplitude of the dynamic impact forces, as discussed in Section 5.1, but also the frequency of the dynamic impact forces. The relationship between the frequency of the impact force and wheel flat size is shown in Figure12 for both wheel-rail impact and impact on sleeper at the rail seat.

The figure shows that the frequency of the impact force is non-linear and monotonically increasing for both wheel-rail impact and sleeper impact. The non-linearity and the increasing rate of the impact forces for wheel–rail impact are larger than for the sleeper impact. The reason could be due to the effect of the rail pad and the different non-linear materials properties. However, the transfer for the frequency of the impact forces between sleeper and wheel–rail was not strongly non-linear which is indicated in Figure13. From Figure13, we can find the frequency of the wheel-rail impact forces is larger than that of the sleeper impact forces. It is because the rail pad between the track and the sleeper plays a function as a damper.

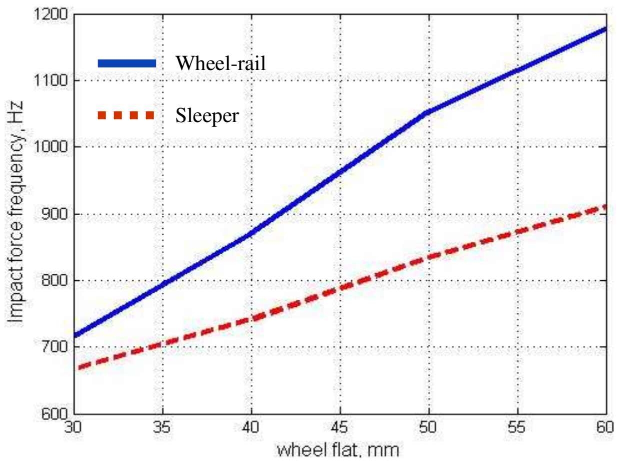

Figure12. Relationship between impact frequency and wheel flat size for wheel-rail impact (solid line) and sleeper impact (dashed line) at the rail seat

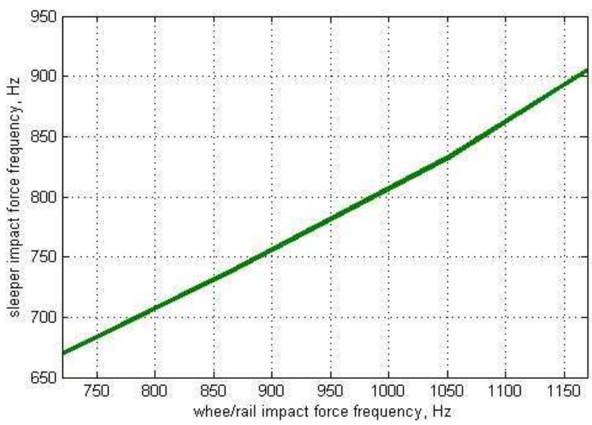

Figure13. Relationship of frequency of sleeper and of wheel-rail impact force at the rail seat

6: Conclusion

A 3-D finite element model for analysis of the behaviour of railway track structures under impact forces induced by wheel flats has been developed and used to conduct several important analyses. The FE model was firstly validated by comparing its results with the results obtained by the DTrack computer model and by theoretical analysis. It has been proven that the established FE model is effective and reliable. Then, the FE model was used to perform a dynamic wheel–rail impact analysis of heavy haul railway tracks and sleepers under wheel flats. The important conclusions can be drawn as follows.

(1) It has been found that the wheel flat will cause significant dynamic impact forces. The larger the wheel flat size, the greater the impact force.
(2) The relationships between wheel–rail impact force magnitudes and wheel flat sizes, and between wheel-rail impact force frequencies and wheel flat sizes, were found to be non-linear and monotonically increasing on both rail and sleeper. These non-linear relationships imply that both the impact magnitude and frequency of the impact force on the track structure cannot be simply predicted by a linear function used in the traditional design method, and the design impact load on track components should be categorized by the wheel–rail incremental impact force.
(3) The characteristics of the frequency of the impact forces are different from the characteristics of the impact force peak value. This needs to be taken into account in the fatigue analysis of sleepers and other track components in the future. According to the characteristics of frequency, track designers can better categorize the critical impact force for the design of track span life. The characteristics of impact frequency also have implications for track maintenance.

The FEA model established in this paper can be a powerful tool for numerical analysis of the dynamic properties of the wheel-track system, especially with wheel flats. Some important conclusions drawn from these analyses in this paper may serve as guidance for the design and maintenance of the wheel-track system.

However, it should be noted here that the model is presently limited to vertical loading analysis of track structure. In order to include lateral (i.e. horizontal) loading, a more detailed model needs to be developed.

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