MODELING OF WHEEL/TRACK INTERACTION WITH WHEEL DEFECTS AND DIAGNOSIS METHODS: AN OVERVIEW

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ABSTRACT
The contact defects between wheel and rail which are originated from wheel profile irregularities and train overloading or unbalancing could deteriorate railway tracks. The interaction defects influence more drastically the maintenance of railways with high traffic like the ones of Eurotunnel tracks. Several works have been performed on different issues of dynamic interaction model between rail and wheel. Such a model can be used for analysis and numerical simulation for wheel profile geometry and train load characteristics. The aim of this paper is to review the major methods and models and classify them, so that different parts of the model are demonstrated: wheel profile geometry, wheel/rail contact model, vehicle and track structure and substructure models, calculation methods and transient simulation models. To achieve the accurate prediction by the model, different methods of exploiting the experimental data to identify the load and diagnosis the interaction defects are also presented.

Keywords: Track/train interaction model, wheel profile irregularity, track response, identification and diagnosis

1. INTRODUCTION

The reduction of railway operating cost is a key issue for infrastructure managers and railway operators. In the scope, maintenance cost reduction is a major target. Highest speed, increased traffic density and load result in accelerated degradation (fatigue). In the same time, interoperability, variety of different vehicles, induces a wider variety of potential degradation sources. One of the most important problems facing the railway maintenance is the monitoring of dynamic behavior of tracks subjected to moving loads (freight and passenger trains) and the defect diagnosis. The structures are therefore subjected to severe vibrations and dynamic stresses, which in turn are much more than the corresponding static stresses.

The dynamic force of railway interaction is influenced by geometrical characteristics of wheel and rail and dynamic characteristics of the load. The goal of diagnosis is to identify these characteristic and the defects related to them. The main geometrical defects are out-of-roundness of wheel profile, rail corrugation, rail joints discontinuity and wheel/rail roughness.

The out-of-roundness (OOR) defects concern the deviation of the wheel tread geometry from its circular shape. Different types of out-of-roundness are catalogued by the International Union of Railways (ETF 04). Two major types of OOR are wheel polygonal and wheelflat. When the brakes are applied to a railway wheel, it can sometimes happen that the wheel locks and slides along the rail. The reason for this may be poorly adjusted, defective or frozen brakes or lack of adhesion at the wheel/rail interface, for example, due to leaves on the rail head. This sliding causes severe wear of the part of the wheel in contact with the rail, leading to the formation of a wheelflat. Such flats on the wheel may be typically 50 mm long but can extend to over 100 mm long.

A gathering of several flats leads to the creation of a polygon. When the wheels rotate, wheelflats generate large impact forces between the wheel and track. Polygonal wheels with a few dominating harmonics (1 to 5 wavelengths around the wheel circumference) have previously been detected especially on high-speed trains. Simulation results show that the most important wavelength-fixing mechanisms of the wheel OOR are the vertical resonance of the coupled train-track system and the frequency region including the lowest vertical track antiresonance (Johansson and Andersson 05).

Wheelflats and rail roughness are very important in the context of dynamic wheel-rail interaction and track deterioration. Dynamic characteristics of the load as variable moving speed and unbalance of wagons load are other important issues to study subsequently. Hereby the parameters like train charge and its vertical position are considered to be fixed.

In order to investigate the track/train dynamic interaction with various load balancing and wheel profile defects, train speeds and static loads, a proper mathematical model is essential. Such a model can be used for analysis and numerical simulation for different train characteristics and parameters (speed, load …) and method of diagnosis. The aim of this paper is to review the major methods and models and classify them, so that different parts of the model are demonstrated: wheel profile geometry, wheel/rail contact model, vehicle and track structure and substructure models, calculation methods and transient simulation models. Such a model is then usable to identify the moving train characteristic and diagnosis the defects on profile of railway wheel. This paper reports on an overview within the framework of the wheel defect topic of the "Track Train System Availability" (TTSA) project of i-Trans competitiveness cluster.

2. PREVIOUS WORKS

The interaction defects between wheel and rail causes noise and safety problem. When wheel locks and slides along the rail, it produces wear and flattening the wheel. Consequently large vibration amplitudes are created which lead to the damage of track and propagation of fatigue cracks. Several works have been performed on different issues of dynamic interaction between rail and wheel. In this part the related more recent studies are addressed contingent on the following topics.

2.1. Geometry of Wheelflat

The force of contact is expressed as a function of the relative displacement between wheel and rail at the contact point, and it depends on the un-deformed wheel–rail geometry and the elastic characteristics of the wheel–rail contact. Two main kinds of model have been used to study wheel/rail interactions, a moving irregularity between a stationary wheel and rail, and a wheel rolling on the track.

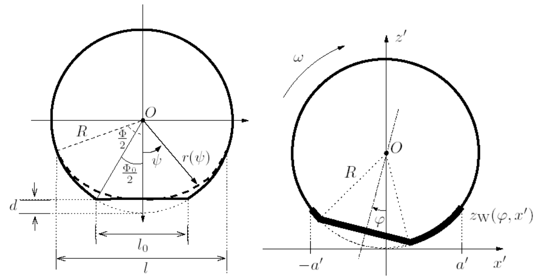
For a discretely supported rail the moving irregularity model cannot deal with the parametric excitation at the sleeper-passing frequency caused by the varying dynamic stiffness in a sleeper bay, and may underestimate the interaction force level at high speeds (Wu and Thompson 06). The moving wheel model is therefore essential to investigate the effects on wheel/rail interaction due to the parametric excitation (Sheng and Thompson 04). This model incorporates vehicles, a track and a layered ground, and uses the moving axle loads and the vertical rail irregularities such as wheelflat in its inputs (Baeza, Roda, and Nielsen 06b).

In a rough approximation, the relation between *d*, depth of a wheelflat, l_0, its length, and *R*, the radius of the wheel could be $d \cong \frac{l_0^2}{8R}$. The geometric equations of a wheelflat is detailed in (Baeza, Roda, Carballeira, and Giner 06a; Pieringer and Kropp 08b).

Two kinds of wheelflat geometry are generally considered: the fresh wheelflat with sharp edges as occurring right after formation and the rounded wheelflat, which rapidly develops from the fresh wheelflat as a result of wheel tread wear and plastic deformation. In the literature, a rounded flat is given the same depth as the fresh flat but with a greater length.

Above a certain critical train speed, the wheel separates from the rail when the interface encounters certain types of discontinuities. The strength of the impact and the frequency of repetition are proportional to the train speed (Vér, Ventres, and Myles 76). For a fresh wheelflat, of depth 2 mm and length 86 mm, loss of contact is found to occur for speeds above 30 km/h. For a rounded flat of the same depth but overall length 121 mm the speed at which loss of contact first occurs increases to about 50 km/h (Wu and Thompson 02).

Pieringer and Kropp (08b) derived the equations for the geometric parameters of a wheelflat related to the centre angle, Φ_0. Because the contact algorithm requires the wheel profile expressed in Cartesian coordinates in the wheel-following coordinate system (x',z'), the orientation of the wheel in this coordinate system was described by the angle φ, $0 \leq \varphi < 2\pi$, see figure 1.



(a) (b)

Figure 1. Geometry of wheelflats in φ=0 (a); and in angular position φ>0 (b)

In order to distinguish the fresh and rounded flats with simpler equations, Baeza et al. defined the irregularity function as the vertical displacement of the wheel form a reference point for bringing wheel and rail in contact (Baeza, Roda, Carballeira, and Giner 06a).

2.2. Model of contact

This still-relevant classical solution provides a foundation for modern problems in contact mechanics. In linear elastic context, where the area of contact is much smaller than the characteristic radius of the body, each body can be considered an elastic half-space. An elastic sphere of radius *R* indents an elastic half-space to depth *d*, and thus creates a contact area of radius *a*. The applied force *F* is related to the displacement d by $F = \frac{4}{3}E^*R^{1/2}d^{3/2}$ where $\frac{1}{E^*} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}$ and E_1, E_2 are the elastic modules and ν_1, ν_2 the Poisson's ratios associated with each body.

This equation is usable for interaction of a round wheel with a rail surface. To consider a wheelflat, it is considered that a rigid cylinder is pressed into an elastic half-space. So it creates a pressure distribution described by $p(r) = p_0(1 - \frac{r^2}{a^2})^{-1/2}$ where r is a the radius of the cylinder $p_0 = \frac{1}{\pi}E^*\frac{d}{a}$ and the relationship between the indentation depth and the normal force is given by $F = 2aE^*d$.

For the first time, Hertz solved the problem involving contact between two elastic bodies with curved surfaces. Based on geometrical effects on local elastic deformation, the Hertz formulation relates the normal transmitted force between the bodies by $F_c = K_H \delta^{1.5}$. The application of 2D Hertzian contact had been dominating related to the other methods just as these cases (Vér, Ventres, and Myles 76; Nielsen and Igeland 95; Zhai and Cai 97; Nordborg 02; Sun and Dhanasekar 02; Sheng, Jones, and Thompson 04; Ford and Tompson 06; Wu and Thompson 06).

Despite its popularity, limits of this approach are that geometric requirement in Hertzian model by which the non-deformed surface collide, should be elliptic paraboloids. Other assumptions which are made in determining the solutions of Hertzian contact problems are:

• the strains are small and within the elastic limit,

• each body can be considered an elastic half-space, i.e., the area of contact is much smaller than the characteristic radius of the body,

• the surfaces are continuous and non-conforming,

• the surfaces are frictionless.

This requirement is not satisfied when rail-wheel contact coincides with flat. Additional complications arise when some or all these assumptions are violated and such contact problems are usually called non-Hertzian. The theoretical results of a Hertzian and a non-Hertzian contact model are compared and it is found out that Hertzian model tends to overestimate the peak impact forces (Baeza, Roda, Carballeira, and Giner 06a; Pieringer and Kropp 08b). Non-linearities in the wheel/rail interaction cannot be neglected in the case of excitation by wheelflats because of the resulting large contact forces and the occurrence of loss of contact for train speeds above the critical speed.

A two dimensional model consisting of a Winkler bedding of independent springs between wheel and rail is introduced (Pieringer and Kropp 10b). Figure 2 shows that for the calculation of the normal contact force, this model takes into account one line of combined wheel/rail roughness, in the rolling direction. The springs in the bedding are independent and allow for loss of contact.

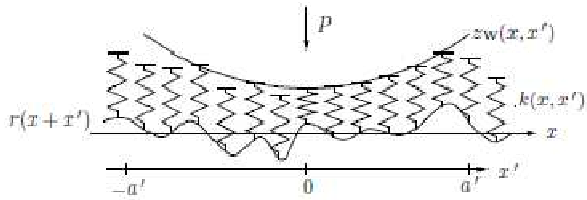


Figure 2. Bedding model of contact

The most general non-Hertzian model used in railway diagnostic is proposed by Kalker (Kalker 95). As mentioned, loss of contact which is occurred on wheelflats, passing over rail joints, and non-linearity could not be considered by Hertzian approach. Thereby the pretabulated Kalker model has taken much attention lastly (Baeza, Roda, Carballeira, and Giner 06a; Baeza, Roda, and Nielsen 06b; Mazilu 07; Mazilu 10). To summarize the process, Baeza, Roda, Carballeira, and Giner (06a) defined the potential contact area (PCA) in such way that it contains every point of the contact area and is rectangular. A discretization of the PCA is established in equal rectangular elements within which the magnitudes to be defined in each element are considered to be constant, as seen in the figure 3.

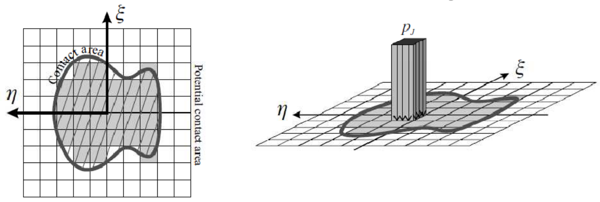


Figure 3. Definition of the potential contact area

2.3. Vehicle Model

In most of cases, a simple model for vehicle is used to achieve the necessary frequency band (Nordborg 02; Wu and Thompson 02; Mazilu 07; Pieringer and Kropp 08b; Steenbergen 08; Pieringer and Kropp 10a). The vehicle motion is governed by the wheel-rail contact. The simplest 1D vehicle model is an unsprung mass. In this model, as shown in figure 4(a), an unsprung mass which represents the wheel is connected to a sprung mass assimilated by a static load for the rest of vehicle. Because the primary suspension filters the high frequency vibration of contact, only the vertical dynamics are considered (Baeza, Roda, Carballeira, and Giner 06a). This model has 2 degrees of vertical movement and the masses are connected through a suspension.

(a)

(b)

(c)

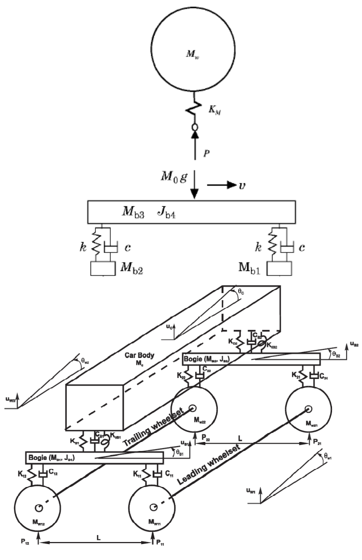


Figure 4. Vehicle model in 1D (a), 2D (b) and 3D (c)

Using just an unsprung mass to show whole vehicle is not the case in practice, where multiple wheels roll on the rail. It is shown that the high-frequency excitation from each wheel can be treated independently by using the superposition principle, provided that the rail vibration is considered as a frequency band average (Wu and Thompson 01).

Although a single unsprung mass on a nonlinear Hertzian contact was the most common vehicle model, it is shown that this model may largely underestimate the dynamic response (Nielsen and Igeland 95). Therefore a 2D vehicle model is proposed which contains 2 unsprung masses. Figure 4(b) presents the complete configuration of this model. The other parts of the model are 1 sprung mass and 2 suspensions and it has 6 dof. By comparing the results calculated for vehicle models including one and two unsprung masses, it was found that the interaction between the two wheels of the bogie model was primarily due to the vibration of the track structure, whereas the bogie frame was not much a effected by the imperfections studied. This is not surprising since bogie suspensions are designed to isolate the unsprung masses from the rest of the vehicle (Nielsen and Igeland 95).

For vibration analysis in dissymmetric load and stability analysis, 3D vehicle model is proposed (Zhai and Cai 97; Szolc 01; Sun and Dhanasekar 02; Hou, Kalousek, and Dong 03; Johnsson and Andersson 05). Figure 4(c) (Hou, Kalousek, and Dong 03) shows a complete bogie model which is consisted of two wheelsets and a bogie frame. The primary suspension between the wheelsets and the bogie frame consists of linear springs and viscous dampers (Johnsson and Andersson 05). The considered vehicle model is supported on two double-axle bogies at each end and is described as a 10 dof lumped mass system comprising the vehicle body mass and its moment of inertia, the two bogie masses and their moments of inertia, and four wheelset unsprung masses (Zhai and Cai 97; Sun and Dhanasekar 02).

2.4. Track Model

The model of rail and its supports is the most important part which affects the accuracy and speed of the simulation and has attracted much attentions. The simplest model which is widely used to represent an infinite rail is Euler-Bernoulli beam (Zhai and Cai 97; Szolc 01; Nordborg 02; Steenbergen 08; Hammoud, Duhamel, and Sab 10). The dynamic response of an Euler-Bernoulli beam under a moving load is analyzed (Lee 94). This model is limited for high frequencies. Timoshenko beam model was developed to consider the shear deformation and large deflections, and has been used to study vehicle/track dynamics to examine the effect of wheelflats since 1926 (Sun and Dhanasekar 02; Wu and Thompson 02; Baeza, Roda, and Nielsen 06b; Mazilu 07). Timoshenko beam model is known to provide a good representation of the vertical vibration of the rail up to about 2 kHz, above which the rail cross-section deformation should be taken into account. The shear deformations can be included not only for mechanical reasons, but also in order to optimize the discretization in the space-domain. So the physically more realistic Timoshenko beam model offers additional numerical advantages when dealing with transient dynamic problems in unbounded domains.

UIC60 is the most frequent rail type, so its characteristic is frequently used. To have a better representation of the rail, it is divided into two parts: the upper part representing the head and the lower part representing the foot. Both the head and the foot are represented by infinite Timoshenko beams in the rail axis direction. These two beams are connected by continuously distributed springs to allow relative motion between them (Wu and Thompson 99).

Although Euler-Bernoulli model is satisfactory at low frequencies, to consider the shear deformation and rotary inertia at higher frequencies a Rayleigh-Timoshenko finite beam elements is used (Pieringer, Kropp, and Thompson 10b). As an example, curve squealing of railway wheels occurs erratically in narrow curves with a frequency of about 4 kHz (Pieringer and Kropp 10a).

A simple model to fix the rail to the ground is the continuous foundation like an elastic half-space and a Winkler Bedding. The main difference between the elastic half-space and the Winkler bedding lies in the fact that in the elastic half-space the points are coupled with each other, while in the Winkler bedding a set of non-coupled springs is used. Because the points in the elastic half-space are coupled an iterative procedure is necessary to determine the displacements due to the roughness profile at each time step. Therefore this model is computationally more expensive than the Winkler bedding. However, the Winkler system cannot represent a real foundation. It's because the coupling between a beam and an elastic mass under a mobile charge is a problem. Complete comparison between maximum moment and maximum displacement obtained by Winkler bedding and halfspace showed that for a given maximum moment, the Winkler bedding yields 1.5 to 1.8 times higher spring constant than the halfspace (Fischer and Gamsjäger 08).

In figure 5, the vertical geometry of the rail, which is continuously supported, is described by *z(x)*. Continuously supported beam is a significant simplification, which will especially affect the model results when irregularities having a predominant wavelength that is comparable to the sleeper spacing distance (Steenbergen 08).

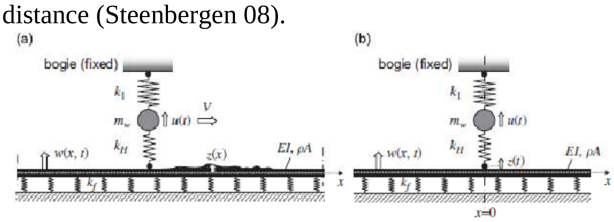


Figure 5. Moving wheel on irregularity z(x) (a) and equivalent excitation z(t) (b)

Figure 6 shows a slab track is considered as an infinite structure consisting of elastically supported double Euler-Bernoulli beams (Mazilu 10).

Some more examples of using the continuous support are presented here:

A simple model is developed based on essential cross-sectional deformation of a double Timoshenko beam in vertical vibration at high frequencies (Wu and Thompson 99).

The foundation consists of distributed non-interacting springs and dampers (Winkler foundation). A track model which has been discretized by use of standard polynomial finite elements will therefore be sought (Nielsen and Igeland 95).

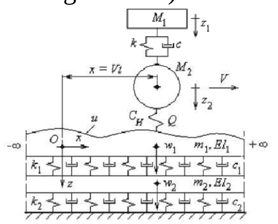


Figure 6. Mechanical model of a moving mass and an infinite homogenous structure

The dynamic behavior of the isolated substructures (rails and sleepers) is described by linear beam theory. The sleepers are modeled as Euler beams on a Winkler foundation, where the ballast is the elastic foundation. Therefore, initially, ballast is considered as having linear characteristics. However, it is possible to include nonlinearities by incorporating them as external forces as a function of sleeper displacements (Baeza, Roda, and Nielsen 06b).

Despite these numerous applications, continuously supported beam is a significant simplification, which will especially affect the model results when irregularities having a predominant wavelength that is comparable to the sleeper spacing distance, and so is not preferred related to discretely supported beams. In cases when loss of contact occurs the effect of the discreteness of the supports becomes important.

While conventional Timoshenko beam model may be used only up to about 2000 Hz, the discretely supported rail model could consider the vertical vibration from about 1000 to 6500 Hz. For high frequencies (at least 5 kHz), the rail cross-sectional deformation is significant. The equation of Timoshenko beam with discrete supports is given in (Sun and Dhanasekar 02).

A discretely supported rail could consider higher frequency vibration modes. Properties associated with the discrete supports are sleeper-passing frequency $f_s = v/l$, and pinned-pinned frequency f_{pp} and the wavelength $\lambda_{pp} = 2l$, where v is the train's speed and l is the distance between 2 consecutive sleepers. Numerical simulations show that it is necessary to include discrete supports in rail modeling to describe the response at low frequencies, determined by the sleeper-passing frequency f_s, and around the pinned–pinned frequency f_{pp}, usually around 1 kHz-in particular if the rail is very smooth or has a corrugation with a wavelength corresponding to the pinned–pinned frequency. If the rail has a corrugation it may also be necessary to include the nonlinear contact spring, since loss of contact occurs for great corrugation amplitudes, e.g., if the corrugation amplitude r_0 is greater than 15 μm when the preload P is 65 kN.

This model is a flexible approach which could be developed in different degree of freedom for the support. The rail is normally described as an infinitely long beam discretely supported at rail/sleeper junctions by a series of springs, dampers and masses. In (Zhai and Cai 97), the three layers of discrete springs and dampers represent the elasticity and damping effects of the rail pads, the ballast, and the subgrade, respectively. The two layers of discrete masses below the rail represent the sleepers and the ballast, respectively. In order to account for the shear continuity of the interlocking ballast particles, shear springs and dampers are introduced between the ballast masses to model the shear coupling effects in the ballast. Also the transient differential equations for 3 layer of supports and a 10 dof car model is developed.

Railpad and ballast/subgrade have a large influence on the track dynamics at low frequencies. In figure 7, two different visco-elastic models of the rail pad is shown (Nielsen 08).

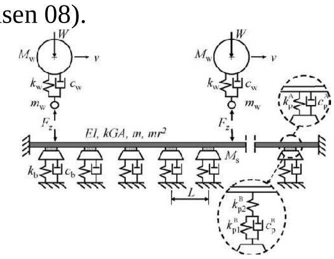


Figure 7. Vehicle model (2 wheelsets) and2 different visco-elastic models of the rail pad

Two track models that include different models of rail pads and ballast/subgrade are compared (Nielsen 08). In track model A, each rail pad is modeled as a discrete linear elastic spring and a viscous damper in parallel (Kelvin model). In track model B, each pad is modeled by a three-parameter visco-elastic model (standard solid model), see figure 7. In track model A, the support under each sleeper is modeled by a Kelvin model, whereas in track model B, a four-parameter visco-elastic model is adopted. The four-parameter model means two spring–damper sets coupled in series, with each set containing one elastic spring and one viscous damper coupled in parallel.

A model is proposed in (Baeza, Roda, and Nielsen 06b) for wheel-rail contact, railpad and ballast with the interaction forces between them and the vehicle. It provides a model with a reduced number of coordinates, so a low computational cost. The developed track model is based on a sub-structuring approach, where a modal description of each isolated rail and sleeper is adopted. Ballast and railpads are considered as connection elements, where the ballast connects sleepers and ground and the railpads connect sleepers and rails.

2.5. Calculation in Time and Frequency Domain

The issue of wheel/rail interaction may be studied by applying the frequency-domain analysis or the time-domain analysis. The most wide spread prediction model for railway noise is a frequency-domain model. It establishes a relationship between receptance and external force at different frequencies using a mathematical transformation under set assumptions, thereby avoiding the solution of complicated differential equations. It is in the nature of frequency-domain models that they can only include a linear contact model. So the frequency-domain analysis requires the assumption that the track is a linear (steady state) structure. Steady-state interaction for an asymmetrical vehicle/track model is calculated by (Hou, Kalousek, and Dong 03). For steady state, each wheel/rail force is a periodic function of time and can be expressed as a Fourier series. The noise is usually presented in frequency bands such as one-third octaves.

Frequency-domain method mainly proposed by Remington is premature, requiring evidence from more advanced description of the generation mechanism. To study the non-linear effect of roughness on interaction force and provide an excitation input to exciting linear model of noise, 2 procedures are examined (Remington and Webb 96). Alternative excitation mechanisms include nonlinearities and parametric excitation due to charging rail and contact receptance with position. For continuously supported track, nonlinear wheel/rail contact effects are mostly unimportant, unless the roughness level is very high or the static load is very small.

Because of the fact that the model is simplified and its linearity, it is not suitable for quantitative predictions with a high level of accuracy. So the nonlinear characteristics of the track structure and the non-linear wheel/rail contact should be modeled in time-domain. However, the time-domain models are more complex than frequency-domain models.

Based on a derivation of the translational and rotational dynamic stiffness of both infinite Timoshenko and Euler-Bernoulli beams on Winkler foundation in the frequency-domain and time-domain beam models, the use of Timoshenko's beam model leads to an asymptotic behavior in the frequency-domain which is linear with respect to $i\omega$. Thus, the corresponding expression in the time-domain is a first-order time derivative. Contrary to Timoshenko's model, Euler-Bernoulli's beam theory generates rational powers of $i\omega$ in the frequency-domain and consequently fractional derivatives in the time-domain with memory integrals to be solved. Their evaluation asks for nonlocal time-solvers with much higher computational effort than local solvers (Ruge and Birk 07).

Two methods of time & frequency-domain are presented to diagnosis the noise generation in wheel/rail contact including discrete supports, parametric excitation, and nonlinear contact spring (Nordborg 02).

A wavenumber-based frequency-domain calculation method is used for the response of a periodically supported rail to a moving harmonic load, allowing dynamic wheel/rail forces to be calculated for single or multiple wheels moving over an initially smooth or rough track. It is shown that this approach is more efficient than a full time-domain approach, but extends earlier work by including the effect of forward motion explicitly.

Another application of time-domain method is that the roughness spectrum is not derived from the wheelflat geometry directly, but from the results of the time-domain calculation. Thus, it is used as a means of converting the wheel/rail interaction force into an equivalent roughness input. It is shown that a time-domain wheel/rail interaction analysis model gives similar results to quasi-static roughness filtering with a constant load for moderate roughness, but dynamic effects became significant when the roughness amplitudes were large, particularly with dipped rail joints (Ford and Thompson 06).

In order to predict the consequent noise radiation, the wheel/rail interaction force is transformed from time-domain into the frequency-domain and then converted back to an equivalent roughness spectrum. This spectrum is used as the excitation to a linear, frequency-domain model of wheel/rail interaction to predict the noise. This hybrid approach has been shown to be adequate by comparing direct and hybrid calculations for a wheel with a single, lightly damped resonance (Wu and Thompson 02).

3. TRANSIENT SIMULATION MODELS

Displacements in physical coordinates, required to calculate the forces transmitted through railpads and contacts, and the values of the force terms which appear in the differential equations (Baeza, Roda, and Nielsen 06b). From the geometry, the kinematics, and the dynamics of the wheel/rail system, analytical models could be developed to identify the major variables controlling the generation of impact noise. The coupling between normal and lateral directions was introduced through the track dynamics due to an offset of the wheel/rail contact point from the rail centre line which is assumed as an input to the model (Pieringer, Kropp, and Nielsen 08a).

The beam model could be considered in time-domain via its receptance. Vertical vibration receptances of the rail have been calculated using both continuously and discretely supported rail models (Wu and Thompson 99). Nielsen studied the vertical dynamic behavior of a bogie moving on a rail discretely supported via railpads by sleepers resting on an elastic foundation. The transient interaction problem is numerically solved by use of an extended state-space vector approach in conjunction with a complex modal superposition for the track (Nielsen and Igeland 95).

Hammoud showed that the fully continuous solutions could not give a satisfactory response for the frequency response model. A formulation for coupling discrete and continuum models for both dynamic and static analyses was given, which offered the better simulations of material properties than the discrete calculations (Hammoud, Duhamel, and Sab 10).

Consideration of wheels moving along a discretely supported rail is normally achieved in the time-domain by solving differential equations as an initial-value problem. Pretension in the rail affects the dynamic response of the rail. The wave propagation velocity in the track is greater than 1000 km/h. For train speeds not exceeding 350-400 km/h the proper estimations performed have shown that the wave effects can be neglected and the most significant influence of vehicle-track relative motion is expressed by periodic fluctuation of track properties during run over successive sleepers.

In analytic solution for a continuously supported rail equation, the derivations related to time and positioned are replaces with its Laplace and Fourier transformed equivalent. But the analytical method will never give a simple and precise solution for a discretely supported rail. To obtain the satisfactory results, one method is using the Green's function; where wheel and rail are represented by impulse response functions. Otherwise the time-domain approaches require the track to be truncated into a finite length. To minimize wave reflections from the truncations and to be able to account for high-frequency vibration, the track section must include at least 100 sleepers and the rail must be modeled using either the finite element method or the modal superposition method employing more than 100 modes (Sheng, Li, Jones, and Thompson 07). The mentioned solutions are explained in this part.

3.1. Finite Element Method

It is generally found that the infinite beams representing the rail resting on an elastic foundation provides only a limited insight into the dynamic response of various track components. An improvement to such models is achieved by accounting the discrete spacing of the sleepers. The discrete support models and the finite element model allow improved prediction of the rail response and offer the potential for refinement by including all conceivable track components as layers.

The advantages of finite element modeling is that the dynamic analysis solved by use of numerical time-stepping routine, non-linear components and contacts may be included. Using real valued modal analysis and U-transformation, to find the dynamic response of an infinite uniform beam, by exact or approached methods. Disadvantage of finite element is that non-physical discontinuity in slope over element boundaries may occur.

A finite element time-domain model for is proposed to determine the dynamic responses to wheel/rail interaction (Hou, Kalousek, and Dong 03). The solution of the wheelflat case reveals that the wheel/rail impact on one rail significantly affects the wheel/rail interaction on the other side of the track.

3.2. Assumed Mode Simulation for Discrete Support

The modal properties of rail and sleepers can be calculated from an Euler-Bernoulli or Timoshenko beam model. The equation of motion in matrix form is formulated by using Hamilton's principle and the assumed mode method (Lee 94). The intermediate point constrains are located arbitrarily along the beam and are modeled as linear springs of very large stiffness.

From a modal basis viewpoint, the vertical displacement of a point located through the longitudinal coordinate x at the instant of t is $v^i(x,t) = \sum_{m=1}^{N_i} \phi_m^i(x) q_m^i(t)$, where $\phi_m^i(x)$ is the mth vibration mode and $q_m^i(t)$ is a set of modal coordinates.

Problem size depends on the number of rails and sleepers, the number of vibration modes considered in the modal descriptions of these elements, and the number of coordinates considered in the vehicle model (Baeza, Roda, Carballeira, and Giner 06b).

3.3. Green's Function

As described, one approach for track model is a finite/boundary method which should be used for a truncated model. To achieve a high precision in this method, the track model is sufficiently long, which is time consuming. Other approach based on analytical models and Green's function method, consider the track as an infinite structure.

Because the time-dependent stiffness of the track is obtained by inversing the receptance due to a unit stationary harmonic load, some authors have tried to study the transient simulation model by Green's function (Mazilu 07; Pieringer, Kropp, and Nielsen 08a; Mazilu 10). Mazilu investigated the interaction between a moving vehicle and a slab track by using new forms of the time-domain Green’s functions for both slab track and vehicle, that are suitable to account for the nonlinear wheel/rail contact and the Doppler effect, due to substructure technique. From the receptance of the rail and slab in the stationary coordinate system, and applying the inverse Fourier transform, the time-domain Green’s functions of the rail and slab are calculated step-by-step taking into account the moving impulse force and assembled in the so-called the Green’s matrix of the track, invoking the damping feature of the track structure. From this equation, the contact force results and then, the wheel and bogie displacements and the rail and slab deflection may be calculated separately. More accurate results for the high-frequency range might be obtained by replacing the Euler–Bernoulli beam model with the Timoshenko beam model.

On the other hand, the Green’s functions of the two-mass oscillator (the wheel and the suspended mass of the bogie) for the time-domain analysis are expressed via the Laplace transform. Starting from the equations of the wheel and rail displacement, the Green’s matrix of the track (only the vector corresponding to the rail) and the Green’s function at wheel meet in the nonlinear equation of the wheel/rail contact. From this equation, the contact force results and then, the wheel and bogie displacements and the rail and slab deflection at the section of the contact point may be calculated separately. No limitative condition regarding the wheel/rail contact or the irregularities of the rolling surfaces is required. In the quasi-static conditions, all Green's functions are set to zero. The quasi-static approach proposed by Wu is adequate to account for the nonlinearity of the wheel/rail contact, but they are not able to simulate the presence of the Doppler Effect.

Green function could be used for both frequency-domain and time-domain models. The time-model proposed by Heckel's determines the vertical rail deflection by time integrating Green function. The frequency-domain model solves a descretized integral equation by coupling the frequency components which are the Fourier coefficients of expansion of the varing receptance along the track, described by the track's Green function. Transforming of Green's function in frequency-domain into time-domain is down by a discrete Fourier transform.

Pieringer also presented the wheel Green's functions and the track by moving Green's functions (Pieringer, Kropp, and Nielsen 08a). One major advantage of the presented interaction model is its high computational efficiency. Even if combined with a complex finite element model of the track, the calculation time for a simulation is typically less than 20 seconds. Because the Green's matrix of the track has to be calculated once for a particular speed and then it can be used to simulate the interaction between the track and the vehicle for any set of vehicle parameters. One drawback of the moving Green's functions is that it should be calculated for each train speed.

4. VEHICLE PARAMETERS IDENTIFICATION AND WHEEL DIAGNOSIS

One method to evaluate the life assessment of the rail is direct measurement of the forces exerted by the train. A transducer called MPQY is developed for measuring the interaction train/track forces to forecast life of the rail (Delprete and Rosso 09). If the interaction forces could be measured, a damage model can be used to evaluate the track life. So in a continuous procedure, every vehicle should be equipped by a force measuring instrument. Since this solution is not suitable for infrastructure managers and is expensive for railway operators, a stationary instrumentation on the rail is often preferred.

4.1. Vehicle parameters identification

In recent years, the technique of moving load identification has been developing very rapidly. Although different systems have been developed for measuring static weigh of the moving vehicles, the important problem is to identify the dynamic load railways. This problem has been discussed with different approaches for bridge deck. Here some cases are presented and they illustrate the possibilities for identifying a vehicle moving on ground-jointed rail. Other parameters like the friction coefficient could be characterized by the load and contact conditions.

The implementation of bridge weigh-in-motion technology is an inexpensive method to measure vehicle characteristics and true dynamic response (Liljencrantz, Karoumi, and Olofsson 07). An existing method of moving load identification on a single-span bridge deck was generalized for a continuous bridge deck with general boundary conditions (Zhu and Law 06). Based on modal superposition and regularization technique, this method was used to identify the moving loads on the elastically supported bridge deck. Numerical simulations showed that the method has been effective to identify accurately the moving loads on the bridge with elastic bearings using different types of measured responses. Measured acceleration gives better results than those from strains. Therefore, data acquisition is based primarily on the use of accelerometers.

Au, Jiang, and Cheung (04) simulated the acceleration measurements from the solution to the forward problem of a continuous beam under moving vehicles, together with the addition of artificially generated measurement noise. The identification was carried out through a robust multi-stage optimization scheme based on genetic algorithms, which searches for the best estimates of parameters by minimizing the errors between the measured accelerations and the reconstructed accelerations from the moving vehicles.

While most of moving force identifications are based on modal decomposition and modal truncation error, a finite element based method was developed (Law, Bu, Zhu, and Chan 04). The measured displacements were formulated as the shape functions of the finite elements of the structure are modeled as straight beam. In this method, the identified results are relatively not sensitive to the sampling frequency, velocity of vehicle, measurement noise level and road surface roughness.

4.2. Railway wheel diagnosis

Wheel defects like OOR and wheelflat cause high impact forces to train and track components. The accelerometer based methods for determining the wheel impact load resulting from wheel defect could prevent catastrophic failure of these components. This approach relies on the dynamic response of the track in determining the magnitude of the impact load imposed on the track by a defective wheel (Wasiwitono, Zheng, and Chiu 07).

The wavelet transform could be used to develop a diagnostic tool for quantifying the wheelflat defect in different train speeds. It is claimed that this diagnostic method is very effective to detect all the damaged wheel and to measure the train speed with a single rough sensor set-up (Belotti, Crenna, Michelini, and Rossi 06). To detect the performance robustness, they varied the threshold value of 10% and verified how results modify.

Skarlatos, Kleomenis, and Trochidis (04) proposed a fuzzy-logic method for wheel defects diagnosis. First, the vibration signatures caused by the rail-wheel interactions were recorded on the rail both in case of healthy wheels and wheels with defects known a priori. The measurements were made on new rails without any defect. Consequently, any expected change in the vibration signatures would reflect the condition of the wheels. Next, the measured data were statistically analyzed and confidence intervals for the wheel condition depending on train speed and frequency of analysis were established.

Another method of detecting the wheelflats by accelerometer is energy and cepstrum analysis of rail acceleration. Energy analysis is useful to estimate the global stress which the undergoes, while ceptrum analysis, a signal processing technique capable of detecting echoes even in strongly noisy signals, allows the detection of the independently from the presence of other defects, even when their effects are hidden (Bracciali and Cascini 97).

Recently one flat detector method using Doppler Effects is presented (Brizuela, Ibanez, Nevado, and Fritsch 10). This system analyses the rail/wheel contact by frequency and phase shifts. When a wheel moves at constant speed, the receiving signal presents a regular shift related to the movement speed. The difference between the emitted and the received frequencies changes if any defect on the wheel tread is detected.

5. CONCLUSION

In this review, some different models which provide an efficient prediction of wheel/rail dynamic behavior relating to the interaction defects were presented. Despite the large volume of researches which have been done on this subject, few work analyze comprehensively the problem for an experimental case.

In order to specify a global dynamic model for railway vehicle/track interaction, some precise and quick methods will be developed. Following the overview, the models comprise 3 different sub-models for track, contact and vehicle, where each part could be refined with increasing complexity. The rail is modeled by Euler-Bernoulli, Timoshenko and double Timoshenko beam methods, considering the necessary frequency bandwidth. For the same reason, type of beam support could be continuous or discretized. For the model of contact, after using the nonlinear Hertzian force, a Winkler bedding contact will be implemented. Two central defects of interaction stated in our study are different types of wheel contour irregularity and asymmetrically loaded vehicle. To analyze the wheel profile irregularities, a 1D vehicle model with 2 dof will be used to reduce the calculation time of the model. But the asymmetrical loading problem could only be analyzed by implementing a 3D vehicle model.

Firstly, each model should face to theoretical results reported in the literature. These models will be corrected and their parameters will be identified later by experimental results that are given via different experimental campaign on Eurotunnel platform. Based on the experimental data, the proposed methods of diagnosis will predict the defects of profile of railway wheel and identify the train load.

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