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Sensitivity analysis of track parameters on train-track dynamic interaction

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Abstract

Dynamic behavior of railway tracks when trains are running is influenced by several factors, i.e. rolling stock, the components of superstructure and their specifications. Usually, features like the sleeper spacing, rail pad stiffness, ballast damping and stiffness have an effect on the dynamic response of the track. The best method to study the dynamic behavior of the track is to model the track assembly and the train as a whole and carry out an analysis of dynamic interaction. Such analysis makes the identification of the track's dynamic behavior easer and helps to anticipate the deterioration of the track elements, and determines the effects of increase or decrease of mentioned parameters. This paper presents track–train dynamic interaction without considering irregularity of the rail face. A sensitivity analysis was carried out on the selected model. The analysis was undertaken with the view of varying one of the mentioned parameters and the results were presented to further identify the deterioration of the track elements.

The results indicate that reducing sleeper spacing, rail pad stiffness, ballast stiffness, and increasing ballast damping reduces wheel-rail, rail-sleeper, and sleeper - ballast contact forces.

Keywords: Train-track interaction; Track dynamics; Wheel-rail contact force; Track parameters; Sensitivity analysis

1. Introduction

Track components become damaged due to dynamic excitations. Such excitation arises from the train running over track irregularities. Provided that a sufficient number of trains pass over a site at a similar speed, the dynamic contact loads between rail, sleeper and ballast are similar from one train to another. The same irregularities excite each train, and the damages caused by one train tend to exacerbate vibration of subsequent trains leading to further damages [1, 2].

Likewise, increasing axle loads and train speed, which is the purpose of railway administrations, requires an accurate mathematical train-track interaction model and numerical solution for dynamic interaction problems for trains on their track. Generally, higher train speed and axle loads lead to increased magnitude of dynamic responses of the track and vehicle. The interactive forces between train and track (via wheel/rail contacts) and track responses depend on the dynamic properties (parameters) of the two, and also on the train speed, track components and wheel defects. Therefore, it is very important to understand the dynamic behavior of track, and to find important parameters, which reduces the dynamic responses of track components.

In this paper, the influence of sleeper spacing, unsprung mass of wheelset, railpad stiffness, ballast stiffness, and ballast damping on dynamic responses of track components are treated.

2. Model, equations and solution method

The train-track interaction system is illustrated in

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Fig. 1. The interacting train and track are both modelled as dynamic systems and the compound train-track system is treated as a whole. In the track model, the rail is treated as a continuous beam and discretely supported, via rail pads and flexible sleepers, to ballast. The rail structure is modelled by using a finite element model with two infinite elements treating the boundary conditions. The most complete sleeper model is a Timoshenko beam of variable thickness, which can readily be analyzed by using finite elements. Half of the sleeper is modelled by use of three uniform beam elements on a visco-elastic foundation. This model includes additional ballast masses below each sleeper, which are interconnected by springs and dashpots in shear [3]. The contact between the wheel and rail is modeled by non-linear Hertzian spring elements [4].

The vehicle is modeled with ten degrees of freedom including the vehicle body mass and its inertia moment, the two bogie masses and their moments of inertia, and four unsprung wheelsets masses. Each bogie frame is connected to its unsprung wheels through the primary suspension springs and to the vehicle body through the secondary springs [5, 6].

The equations of motion of the vehicle, which are regularly derived according to D'Alembert's principle, are second order ordinary differential equations. Equations of motion of the track subsystem consist of rail, sleepers and ballast masses. These equations have been explained in detail in [7, 8].

The general motion equations for a train – track system with N degrees of freedom are written as

$$[M]\{\ddot{v}\}+[C]\{\dot{v}\}+[K]\{v\}=\{F(t)\} \tag{1}$$

Where $\{F(t)\}$ denotes the externally applied forces, [M], [C] and [K] are mass, damping and stiffness matrices, respectively.

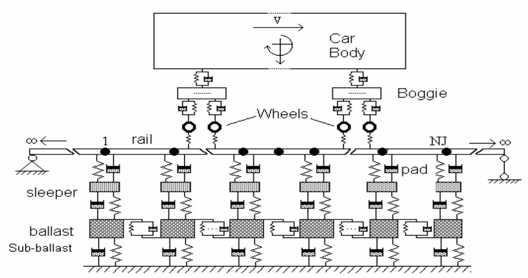

Fig. 1. Train-track interaction model

In Zakeri [7], these three matrices are described in detail. In order to solve the second order differential Eq. (1) with high degrees of freedom involving nonlinearity, due to the nonlinear contact forces between wheels and rails, the direct time integration method must be adopted to obtain numerical results in time domain. The time histories of displacement, velocities, accelerations of the whole system and dynamic forces between wheel/rail, rail/sleeper and sleeper/ballast can be completely calculated.

3. Application of DATI program

DATI computer software is designed to simulate the vertical dynamic interactions between railway tracks and vehicles. The time histories of displacement, velocities, accelerations of the whole system and dynamic forces between wheels and rails can be completely calculated.

Since the interaction problem is generally solved by use of numerical time-integration, the computer time when employing this type of track model substantially exceeds the time required in an analysis of less detailed models.

It is better to show the process of calculations by a flow chart. The flow chart of the complete simulation process is illustrated in Fig. 2.

In a personal computer a primary limitation is the available memory, and it is not possible to retain all of the necessary arrays in core at one time. Accord-

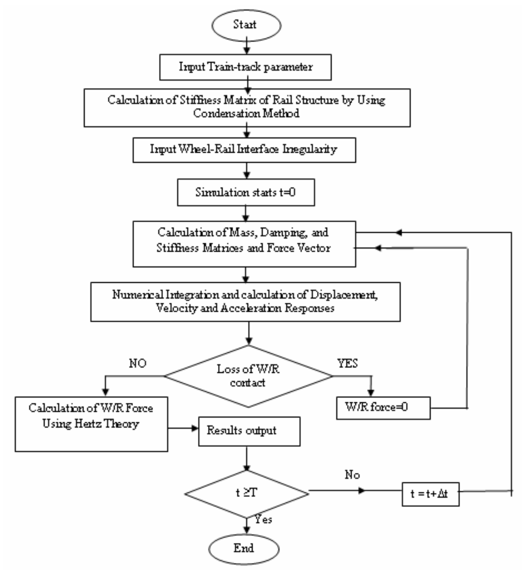

Fig. 2. Flow chart of DATI program.

ingly, an out-of-core frontal system is adopted to solve linear algebraic equations.

A simple memory management system utilizing the main memory and disk files is used to store large arrays resulting from the global coefficient matrix and any element history terms. For implementation on larger computers with virtual memory management, it is more efficient to avoid using disk files as much as possible. Accordingly, for these systems an in-core variable band solver is included as an equation-solving option.

The calculated results are well in accordance, both in response curves, in amplitudes and in distribution tendencies, with the existing experimental data [7], which verified the effectiveness of the analytical model and the computer simulation method.

4. Results

In this paper, UIC60 rail and ICE trains have been used, for which all of the technical parameters are shown in Table 1. A length of 53 sleeper-spacing is used in this model. According to Fig. 3, there are 59 joints, 52 rail supports and 58 rail elements. Displacements, accelerations and velocity of the joint No 30 (mid joint of selected track) and that of joint No 28 (mid support of rail track) are calculated. Also, wheel-rail vertical force, rail/sleeper and sleeper/ballast forces are calculated. The train speed is considered about 160km/h [8].

Table 1. Track and vehicle parameters.

Track Model Parameters	Vehicle Model Parameters
$E = 206\times10^6 \quad KN/m$	$M_C = 49500 \quad Kg$
$I = 32.2\times10^{-6} \quad m^4$	$J_C = 1.7\times10^3 \quad T.m^2$
$M_r = 60 \quad kg/m$	$L_C = 19.0 \quad m$
$K_P = 240\times10^3 \quad KN/m$	$M_t = 10750.0 \quad Kg$
$C_P = 248.0 \quad KN.S/m$	$J_t = 9.60 \quad T.m^2$
$M_S = 320 \quad Kg$	$L_t = 2.5 \quad m$
$K_B = 70\times10^3 \quad KN/m$	$K_T = 1720.0 \quad KN/m$
$C_B = 180.0 \quad KN.S/m$	$C_r = 300.0 \quad KN.S/m$
$M_B = 1400.0 \quad Kg$	$M_W = 2200.0 \quad Kg$
$K_F = 130\times10^3 \quad KN/m$	$K_w = 4360.0 \quad KN/m$
$C_F = 62.3 \quad KN.S/m$	$C_w = 220.0 \quad KN.S/m$
$L = 29.15 \quad m$	$K_H = 2.4\times10^5 \quad KN/m$

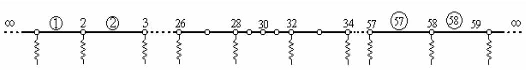

Fig. 3. Rail structure of track.

For better understanding of the dynamic behavior of track, the factors affecting on track responses are investigated. The following results are obtained for variation of one parameter at a time.

4.1 Effect of sleeper spacing

Having acknowledged the Winkler theory of rail on non-continuum supports, an increase in the distance of sleepers would raise the stiffness of the supports and consequently cause a higher reaction from the

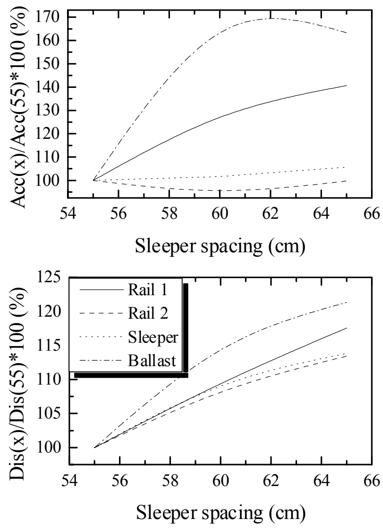

Fig. 4. Variation of displacement and acceleration of track components with sleeper spacing.

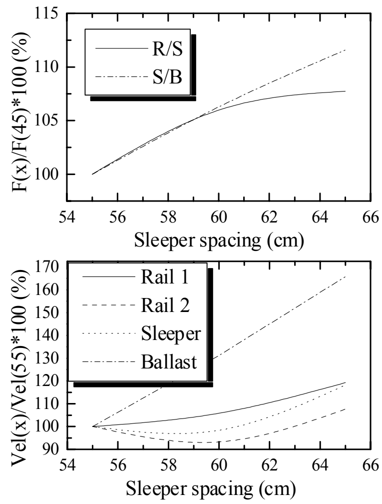

Fig. 5. Variation of velocity of track components, R/S and S/B dynamic forces with sleeper spacing.

supports and result in displacement of the rail. A sensitivity analysis was performed by distancing the sleepers and measuring the effects in a track-train interaction modeling. The results illustrated in Figs. 4 and 5 show that an increase in the distance of sleepers from 55cm to 65cm would create an increase of 12% in the reaction forces of the supports, an increase of 18% in the displacement of mid span of the rail, an increase of 12% in the displacement of rail on sleeper and displacement of the sleeper. This issue also boots the acceleration of vibration relative to that of base (55cm gap), for both mid span rail and ballast reaching 35% and 65%, respectively. The effect of such acceleration is noticeable in declining of fasteners.

4.2 Effect of railpad stiffness

Rail pads, which lie between rail and concrete or steel sleepers or sometimes positioned on steel plates of wooden sleepers, play an elastic role in the structure of track. Since contact surfaces between rail and sleepers are rigid, this elastic rubber is necessary. Stiffness of pad rails also plays an important role. Sensitivity analysis on the stiffness of existing rail pads through track-train interaction modeling shows that by increasing rail pad stiffness from 50MN/m to 250MN/m, interaction forces of ballast-sleepers and sleeper-rails increase by about 9%. Also, vibration acceleration of rail at supports does not vary perceptibly, whereas this acceleration is reduced by about 10% at the mid span rail. Noticeable changes also occur with regard to the displacement of both mid span and supports. Increase of rail pad stiffness, from 50MN/m to 250MN/m, will cause a decrease of 45% in displacement of the rail (Figs. 6, 7), which, in turn, is of paramount consideration [5].

4.3 Effect of ballast stiffness

Ballast stone material and its grade play an important role in varying the ballast layer stiffness. In practice, during operation and moving of the freight trains, falling of small particles over the ballast layer causes an increase of stiffness in the layer of ballast. Also, a gradual deterioration of ballast affect on size of ballast grade and forming powder due to friction exists between stones rubbing together, which affect the stiffness of ballasts. For this reason, a sensitivity analysis was conducted on the ballast stiffness by a sampled track-train interaction modeling and the outcomes are as follows:

Increasing ballast stiffness from 60MN/m to 100 MN/m resulted in 20% reduction of rail displacement at mid span, on supports and also displacement of sleepers. Interaction forces between ballast-Sleepers and sleepers-rails also increased by 6%. The biggest change was in vibration acceleration of ballast reaching to about 16% (Figs. 8, 9).

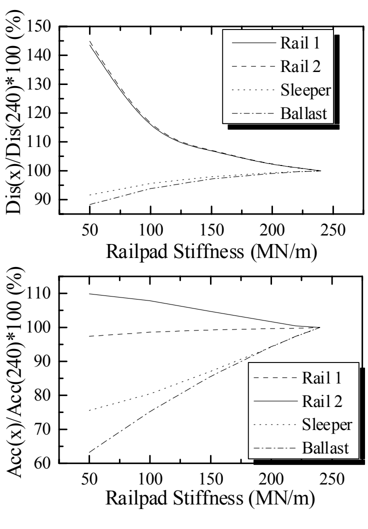

Fig. 6. Variation of displacements and accelerations of track components with railpad stiffness.

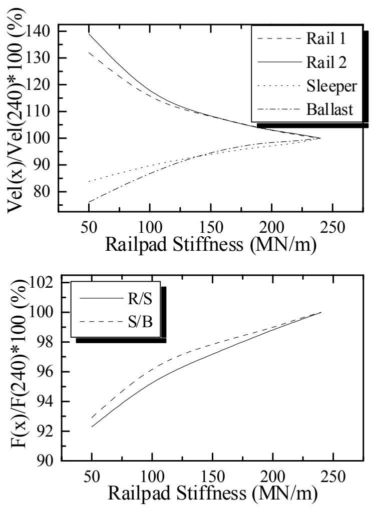

Fig. 7. Variation of velocities of track components, R/S and S/B contact force with railpad stiffness.

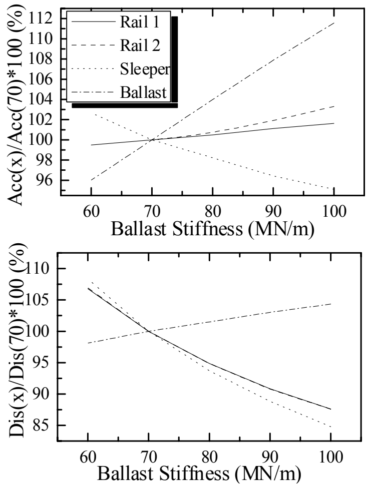

Fig. 8. Variation of displacements and Accelerations of track components with ballast stiffness.

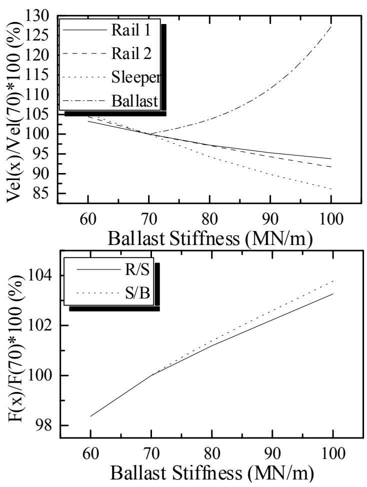

Fig. 9. Variation of velocities of track components, R/S and S/B contact force with ballast stiffness.

4.4 Effect of ballast damping

The damping coefficient of ballast also plays an important function in the behavior of track vibration. As we know, the determination of such coefficient is practically very difficult. For this reason, a sensitivity analysis was carried out on a track-train interaction model so that if there was an error in evaluating the damping ratio, changes in parameters such as displacement, vibration acceleration and tracks element interaction forces could be identified. In this study, ballast damping ratio was considered to be between 30 kN.s/m and 180 kN.s/m. The following considerable results have been deduced.

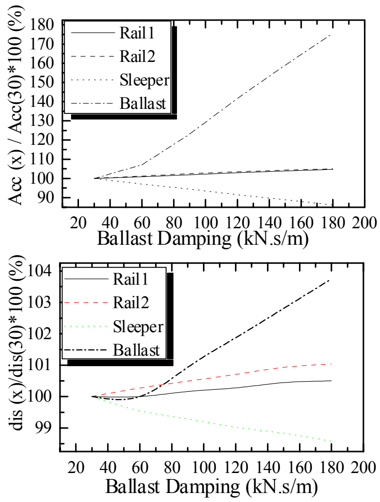

Fig. 10. Variation of displacements and accelerations of track components with ballast damping.

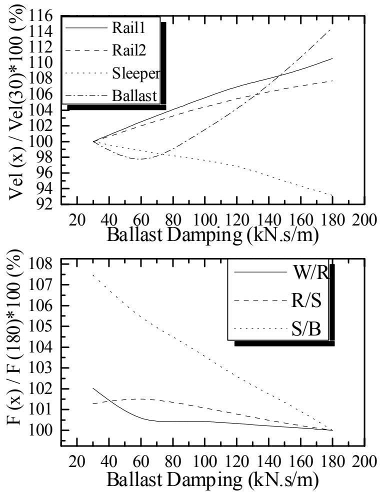

Fig. 11. Variation of velocities of track components, W/R, R/S and S/B contact forces with ballast damping.

- Rail displacements on both supports and at mid span approximately remained.
- Displacement of ballast layer rose by about 5%.
- Rail vibration acceleration also remained roughly unchanged.
- Ballast vibration acceleration grew by about 75%.
- As anticipated, sleeper vibration acceleration was reduced by about 15%.
- Contact forces of wheel and rail were down by 2%, up to damping ratio of 80 kN.s/m and after that remained unchanged. This change, as was foreseen, was due to the consistency of displacement of rail and sleeper (Figs. 10, 11)..

- Interaction forces between sleeper and rail also have had a 2% rise.
- Interaction forces between sleeper and ballast dropped by 7.5%.

Hence, it seems that measuring and controlling the ballast layers vibration acceleration could be beneficial in evaluating the ballast damping coefficient.

5. Conclusions

In this paper, for a better understanding of the dynamic behaviour of track, the factors affecting on track responses were investigated. The numerical results for W/R, R/S and S/B interactive forces, track component displacements, velocities and accelerations were presented. The following results were obtained.

Reducing sleeper spacing increases the variation of the overall track stiffness and hence reduces all of the track responses and interactive forces.

Increasing the unsprung mass of wheelset increases all of the responses, especially wheel/rail contact force, rail displacement and ballast acceleration.

Reducing the railpad stiffness reduces ballast and sleeper displacements, ballast and sleeper accelerations, ballast and sleeper velocities, W/R and R/S interactive force, but increases rail displacement, rail acceleration and rail velocity.

Increasing the ballast stiffness reduces rail and sleeper displacements, sleeper acceleration, rail and sleeper velocities, but increases ballast displacement, rail and sleeper accelerations slightly, ballast velocity, R/S and S/B interactive forces.

Increasing the ballast damping reduces sleeper displacements, sleeper acceleration, sleeper velocities, W/R, R/S and S/B forces, but increases the ballast displacement and acceleration, rail and ballast velocity.

It is observed that the magnitude of W/R contact force mainly depends on the unsprung mass, sleeper spacing, and rail stiffness.

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