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Influences of Railway Track Geometry Parameters on Ground-Borne Vibration

J. Sadeghi[1,*], M.H. Esmaeili[2], M. Shadfar[2],M. Ashari Ghomi[3]

[1]Centre of Excellence for Railway Transportation, Iran University of Science and Technology, Tehran, Iran
[2] School of Railway Engineering, Iran University of Science and Technology,Tehran, Iran
[3]Tehran Urban and Suburban Railway Department,Tehran, Iran

A B S T R A C T

One of the most important challenges in the development of urban railways network is the effects of railway vibrations on human and adjacent structures. Although various investigations into the sources of railway vibrations have been conducted, very few research works have been reported on the influences of track geometry parameters on the level of vibrations. This is addressed in this research. The influences of railway track geometry parameters including rail cant, track gage, super-elevation, curve radius, profile, alignment and twist on ground-borne vibration were investigated by sensitivity analyses of a numerical model developed in this research. The model was validated using data obtained from a comprehensive field measurement conducted in Tehran Metro Lines (Iran). Results indicate that the effects of geometry parameters on track vibration levels are considerable. Based on the results obtained, the current code of practice was improved by establishing new limitations on the allowable track geometry parameters tolerances. It was shown that consideration of these new limitations ensures that the induced track vibrations remain within the allowable limits.

Keywords: Ground-borne vibration, Railway geometry, Numerical modeling, Field measurement

1. Introduction

One of the main concerns in the development of railways particularly in the urban areas is vibrations induced by the railways. A review of the literature indicates that several investigations have been conducted into the effect of various track and rolling stock parameters on ground-borne vibrations in the last three decades. The effect of wheel flat and wear on track vibration was investigated by Torstensson and Nielsen [1]. Gupta and his colleagues studied the effects of soil and tunnel lining parameters on ground-borne vibration [2]. Further works through filed tests were made by Madhush and Kaynia in which the effect of soft soil parameters on vibration was investigated [3]. The effect of multi layered soil, bedrock depth, and mechanical properties of soil were analyzed by a number of researchers including Auersch [4-6], Yang and his colleagues [7,8].Using field tests and theoretical models, Cox and his colleagues have studied the effect of floating and fix-direction slab-track stiffness in vibration attenuation [9,10]. Effect of thickness and length of the slab-track was investigated by Yuan et al [11]. Yang and Lambaert focused on vibration prediction in conventional and high-speed lines through analyses of numerical models and field test results [7] and [12]. Zhang and his colleagues developed a FEM model to evaluate the role of track fastening system on ground-borne vibration [13]. Ho and his colleagues adapted the same approaches to investigate track vibration levels at turn-outs when compared with tangent lines [14] and [15]. A comprehensive review of the studies related to high-speed track vibrations was presented by Krylov [16].

Despite noticeable researches carried out to investigate the effects of train and track parameters on vibration levels, the effect of track geometry

*Corresponding Author
Email Address: javad_sadeghi@iust.ac.ir

parameters have not been fully studied yet. Curve radius, super-elevation, rail cant, track gauge, profile, alignment and twist are the track geometry factors that may influence the rail-wheel interaction and in turn track vibration level. The influences of these parameters on track vibrations were investigated in this research. This was made by parametric analyses of a numerical model of train-track system developed and validated in this research.

2. Modeling procedure

A widely used railway modeling software called Universal Mechanism (UM) was used to develop a model of train-track system [17]. Multi Body Dynamics (MBD) technique and Winkler theory were used to model the wagon and the track, respectively. The model includes two parts: trains and track superstructure. The train model includes car-body, bogie frame, wheel-set, axle-box, primary and secondary suspension systems and wheel-rail contact mechanism. The wheelset was considered as a 6 DOF system and axle-box was modeled as a 1 DOF mass connected to wheelset (revolute joint). Bogie frame was considered as a 6 DOF mass (Fig. 1) connected to the axle-box with a primary suspension system (Fig. 2). The primary suspension system was modeled using rubber elements presented by stiffness and damping.

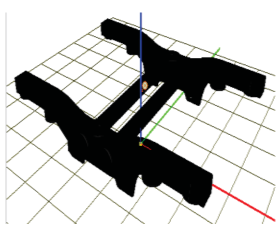

Figure 1: Modeling of bogie frame

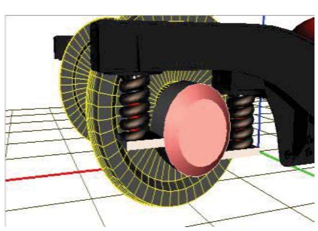

Figure 2: Modeling of spring and damperof primary suspension system

As presented in Fig. 3, the secondary suspension system which includes air spring and side bearers was modeled using Nishimura theorem [18]. To this end, a visco-elastic element with longitudinal, lateral and vertical stiffness in the vertical direction and parallel to the linear spring were considered. K1, K2, K3 and C parameters in Nishimura theorem (Fig. 3) were defined by the following equations [18]:

$$K_1 = \frac{nP_0}{V_b}A_e^2,\ K_2 = \frac{nP_0}{V_a}A_e^2,$$
$$K_3 = (P_0 - P_{atm})\frac{dA_e}{dz},\ C = GA_e^2\rho_0 \quad (1)$$

$$F_0 = (P_0 - P_{atm})A_e \quad (2)$$

In which A_e is the effective area of air spring, n is the polytrophic coefficient of the air, P_0 is the static pressure of the air spring reservoir, P_{atm} is the environment pressure, V_a and V_b are the spring and reservoir volume respectively, F_0 is the spring preload, G is an additional coefficient explaining stiffness to damping ratio, and ρ_0 is the air density. G is determined by Equation 3 as follows:

$$G = \frac{1}{R} = \frac{0.126g}{d^3} \quad (3)$$

Where g is the gravity and d is the bogie air spring orifice diameter. All dimensions are in SI system. Center pin was modeled by allowing car-body to rotate in the yaw direction (rotation around vertical axis), having 3 mm clearance in both lateral and longitudinal directions between center pin and bogie frame. The car-body mass was modeled as a 6 DOF system. Pre-calculated Contact Table method for nonlinear 3D theorem of Kalker with FASTSIM solver [19], was used to model the wheel-rail interaction. The rail head and rail side friction coefficients were considered 0.25 and 0.2, respectively. Finally, the track was modeled as a beam rested on an elastic foundation. The schematic view of the train model (with 50 DOF) and the model of train-track are presented in Figures 4 and 5.

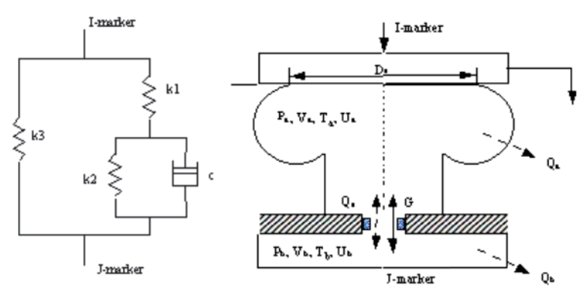

Figure 3: Schematic view of Nishimura model used for secondary suspension system [18]

3. Model Verification

In order to evaluate the validity of the model, comprehensive field tests were made in Tehran subway line. Tehran metro line 1 (Fig. 6) with a length of 39 km has been under operation for nearly 15 years. Vast damages to the adjacent buildings caused by ground-borne vibration have been reported in some locations of the line particularly in the vicinity of sharp curves [20]. Track vibrations in term of acceleration at several locations of the line were measured. For this purpose, the train was instrumented by installing two 3-axial accelerometers on the axle-box and on the bogie frame right up to the first accelerometer (Figs. 7 and 8). The axle and bogie frame accelerations were recorded along the line 1 of Tehran subway. Using recorded accelerations, the dynamic forces imposed on the track were calculated. The tests were carried out after ensuring that there is no corrugation on the rail surface. The results in time and frequency domains for the curves with the radiuses of 498, 401 and 298 meters are presented in Fig. 9. The vibration level is mostly presented in a logarithmic scale expressed in dB. This is made by using the one second following equation [21] where $F_{rms_{curve}}$ is the second root mean square of measured force and F_{ref} is the reference force value equal to 10^{-6} N [21].

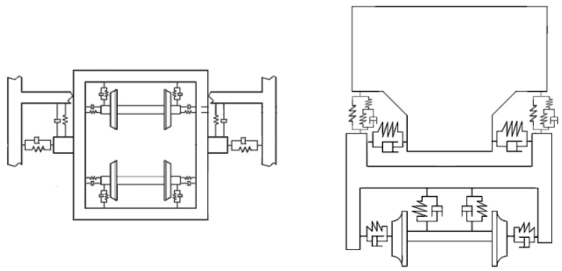

Figure 4: Schematic view of vehicle model

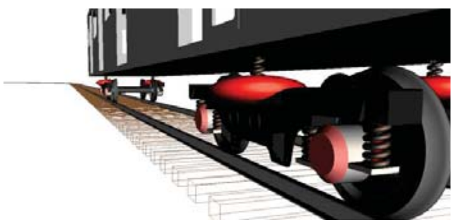

Figure 5: 3D model of train and track model

$$dB = 20log\frac{F_{rms_{curve}}}{F_{ref}} \tag{4}$$

As indicated in Fig. 9, the vibration level for tangent track is 219 db and for the curves with 498, 401 and 298 m radiuses are 219.6, 219.8 and 220 dB, respectively.

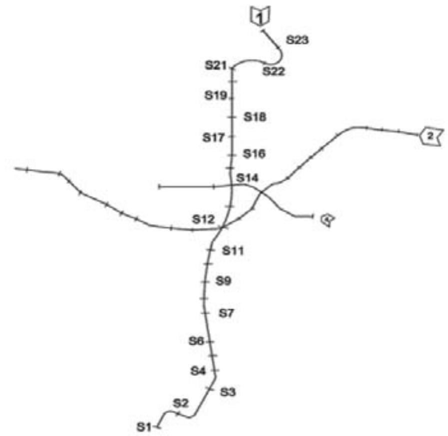

Figure 6: Tehran subway line 1, field test

The model (developed in the last section) was run based on the properties of the train and the track used in Tehran Metro Line 1. The masses, stiffness, damping and geometrical parameters of the cars were obtained from Tehran metro car manufacture [22]. They are presented in Tables 1 to 3. The slab-track modulus, the rail cant and the rail profile are 100 KN/mm, 1:20 and UIC 54, respectively.

Comparisons of the results obtained from the model and the field tests are presented in Fig. 10.

According to this figure, the differences between results obtained from measurements and the model are less than 5%.

Table 1: Train parameters [22]

Part	Mass (Kg)	CG (m) (from rail surface)	Ixx ($Kg.m^2$)	Iyy ($Kg.m^2$)	Izz ($Kg.m^2$)
Wheelset	1144	0.42	579.392	85.793	579.392
Axle box	54	0.42	1.32	1.02	1.6
Frame	978	0.595	705.513	328.168	1018.5
Carbody	28000	0.7	56800	1970000	1970000

Table 2: Suspension system parameters [22]

Part	Kx(N/m)	Ky(N/m)	Kz(N/m)	Cz(N.s/m)
Primary suspension	3533500	3533500	931950	541.31
Secondary suspension (Fig. 3)	710000	100000	K_1=347556 K_2=347556 K_3=670000	627840

Table 3: Wheel-set parameters [22]

Wheel-base space (m)	2.2
Tap circle distance (m)	1.5
Wheel profile	S1002

Figure 7: Train instrumentation

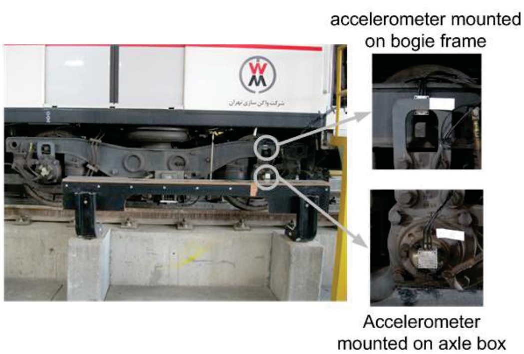

Figure 8: Accelerometers Mounted on front bogie

4. Parametric analysis

The effect of the main track geometry parameters are investigated by parametric analyses of the model developed and validated in the last section. These parameters are: rail cant, gauge, curve radius, super-elevation, profile, alignment and twist. Rail cant is the inward inclination of the rail. Gauge is the right angle distance between the two rails at a given location, measured 14 mm below the top surface of the railhead. Super-elevation is the difference between vertical elevations of the two parallel rails. Super-elevation helps compensate for the centrifugal force on cars rounding the curve. It is not considered as a defect unless it deviates from the desired super-elevation. If super-elevation is greater or less than the desired value, it is considered positive and negative, respectively [17]. Profile is the track geometry of each rail or the track centerline projected onto the longitudinal vertical plane. Deviation in profile is the change in elevation of the two rails along the track relative to a designated grade. Alignment is an indicator of how well positioned the rails are horizontally along the intended route [23].Alignment deviation is the difference between the designated alignment and what is actual; it is measured as the horizontal distance between the gauge side railhead (measured at 14 mm below the top) and center of a certain chord length [23]. Twist is the variation in track super-elevation along the track line.

To conduct the sensitivity analyses, a reference model was defined. The properties of the reference model were identical to the track properties in the site described in section 3. In the parametric analyses, variations in track vibration levels due to changes in each track geometry parameters of the reference model were evaluated. The results obtained are presented in Figs. 11 to 16.

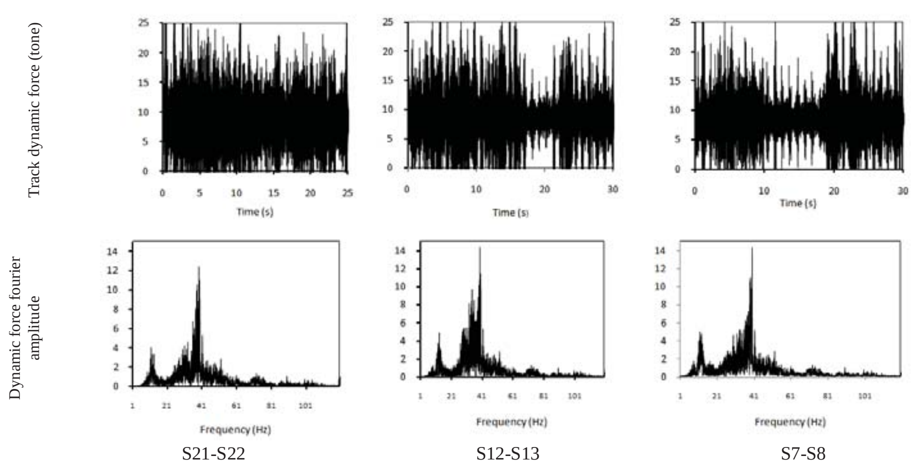

Figure 9: Track dynamic forces in time and frequency domains for various curves

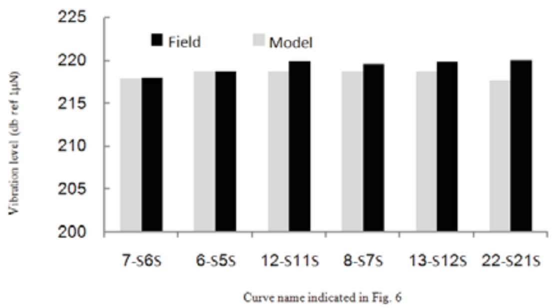

Figure 10: Field test measurements compared with theoretical results

Track vibration levels against curve radius in various amounts of super elevation are plotted in Fig. 11. As illustrated in this figure, an increase in curve radius, results in decreases in vibration levels. Negative super elevation deficiency has caused an increase of 0.2 dB, while the effects of positive ones are negligible. This means that running at the curves with the speeds higher than the designated value has considerable impacts on the induced vibration forces. However, if the radius of the curves is greater than 500 meters, the effect of curve radius is no longer considerable. The effect of track gauge variations is not considerable when the train speed is less than 100 km/h (Fig. 12). As the train speed exceeds 100 km/h, increases of track gauge excess from 0 to 20 mm cause 2 db increases in vibration levels. It means that if the train speed exceeds the hunting limits, its vibration impact is intensified.

Track vibration level versus train speeds is plotted in Fig. 13. The vibration level has a linear correlation with the train speeds for rail cants of zero and 1:40; this correlation has a parabolic shape for the rail cant of 1:20. Results indicate that the impacts of rail-cant on vibration levels are noticeable when the train speeds is greater than 100 km/h. This is more considerable for the rail cant of 1:20 when compared with zero and 1:40 rail inclination. As indicated in Fig. 14, the effect of track twist for a range of 1 to 7 mm/m is highly dependent on the train speeds. Track twist does not affect ground-borne vibration when the train speed is less than 70 km/h or if the track twists are less than 7 mm/m. If the twist reaches 8 mm/m, there is 3 db increases in vibration levels.

Track vibration levels are plotted against alignment deviations for the train speeds of 40 to 160 km/h. As it is shown in Fig. 15, when alignment deviations vary from 5 to 45 mm, the vibration level increases up to 4.5 db (2 %). If the alignment deviation is more than 30 mm, its impacts on vibration levels increase substantially (i.e., the more alignment deviation, the higher rate of impact on vibration). The graphs given in Fig. 16, presents the influence of profile deviations on vibration levels for the train running at various speeds. Increasing profile deviations from 2 to 20 mm causes an approximately 9 dB increases in vibration levels. The rate of changes in vibrations levels because of changes in profile deviations substantially increases when the deviations exceed 15 mm.

5. Recommendations on codes of practice

The results of parametric studies indicate that track geometry deviations can have noticeable impacts on the track levels of vibration. This is not considered in the current codes of practice where the allowable tolerances for the track geometry deviations (deficiencies) are suggested. Since the track vibrations can have considerable harms to surrounding environment (human and structures) particularly in the urban railway networks and high speed lines, there is a need to revise the current codes of practice by taking into account the track induced vibration as a criterion. Based on the results obtained in this research, new limits were developed to consider the vibration allowable limits in the track allowable geometry tolerances. This is summarized in Table 4.

Table 4 was developed based on the influences of each track geometry parameters on track vibration. According to this table, track gauge tolerances are limited to 15 mm which is obtained from the maximum allowable vibration levels. The use of rail cant of less than 1:40 is limited to the speed of 100 km/h or less. As indicated in the table, the allowable alignment and profile clearances were limited to 30 and 16 mm, respectively. Current codes such as EN-13803 [24] and Iranian national codes [25] allow a curve radius of 250 meter. TCRP suggests the use of 150 m curves for conventional tracks [26]. The minimum curve radius of 500 meters is recommended in the table based on the maximum allowable level of vibration. The limitations for super-elevation deficiencies and twist tolerances indicated in the current code are satisfying from the vibration point of view.

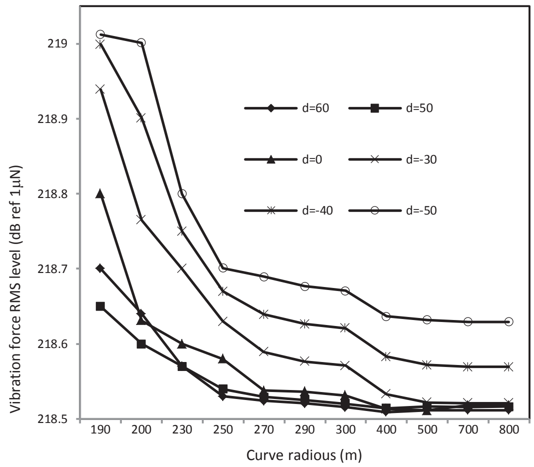

Figure 11: Track vibration plottedagainst curve radius in various super-elevation (d)

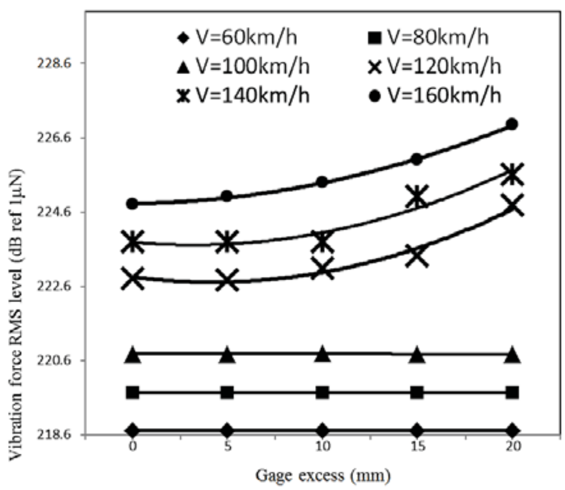

Figure 12: Track vibration plotted againstgauge excess in various train speeds

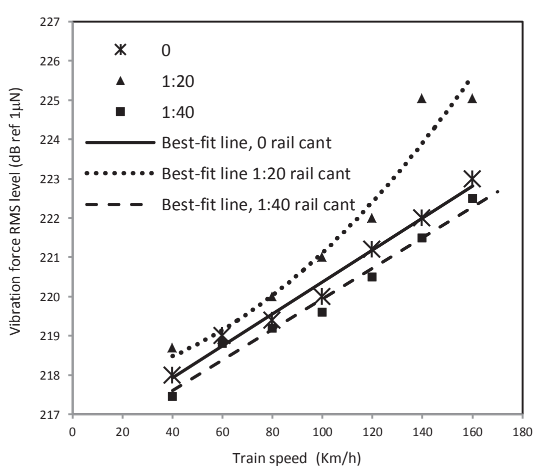

Figure 13: Effect of rail cant on vibrationlevel for various train speeds

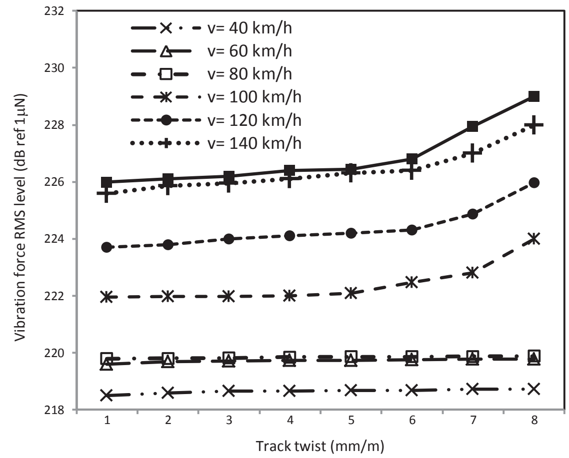

Figure 14: Effect of track twist on trackvibration for various train speeds

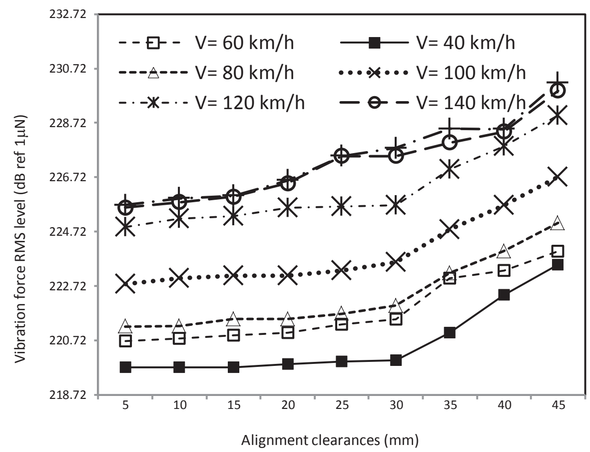

Figure 15: Effect of alignment deviations on vibration level for various train speeds

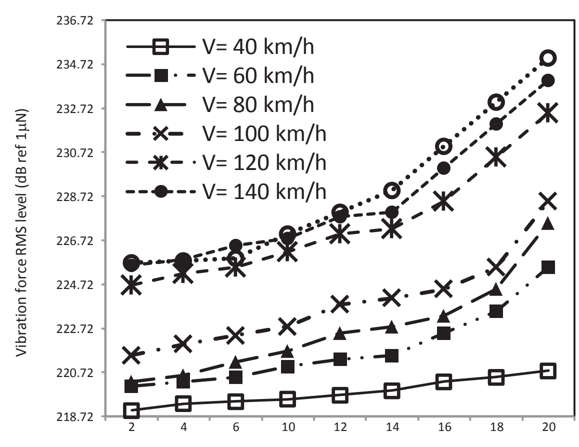

Figure 16: Effect of profile deviations on vibration level for various train speeds

Table 4: Track geometry allowable tolerances (suggested in this research)

Parameter	standards	Speed (km/h)	Current tolerances	Maximum level of amplification (db)	Adjustment
Gage (mm)	EN 13848-5 (2008) [27]	V<80	Min -6 Max +25	1.5	The maximum values should be limited to +15 mm for speeds of 120 km/h of higher
		80<V<120	Min -5 Max +22.5		
		120<V	Min -2.5 Max +16		
	TSI (2008) [28]	-	Min -9 Max +35		
Rail cant	No criterion founded	-	-	1 *	Use of rail cant of 1:20 should be limited to the speed of 100km/h, should be considered in each case
Alignment (mm)	MPO (2004) [29]	-	Max 41	3	The maximum values should be limited to 30 mm
	EN 13848-5 (2008) [27]	V<80	Max 15		
		80<V<120	Max 11		
		120<V<160	Max 9		
Profile (mm)	EN 13848-5 (2008) [27]	V<80	Max 18	10	The maximum values should be limited to 16 mm
		80<V<120	Max 16		
		120<V<160	Max 15		
Curve radios (m)	TCRP (2012) [22]	-	Min 150	0.5	The minimum curve radius should be increased to 500 meters
	EN-13803 (2009) [27]		Min 250		
	MPO (2004) [29]		Min 250		
Super elevation (mm)	TSI (2008) [28]	-	+/- 20	0.4	The current criteria are satisfying
Twist (mm/m)	TSI (2008) [28]	-	7	0	The current criteria are satisfying

*In the case of 1:20 rail cant and S1002 wheel profile

6. Conclusions

Railway ground-borne vibrations particularly in urban areas have been considered as one of the most challenging railway engineering practices in the recent decades. Although considerable number of researches has been conducted on the effect of track and train parameters on ground-borne vibration level, the effect of track geometry parameters has not fully studied yet. This is addressed in this research. A numerical model of train-track system was developed. It was validated through field investigations. The influences of the track geometry parameters on level of ground bourn vibration were investigated by sensitivity analyses of the model. It was shown that track geometrical parameters including rail cant, super-elevation, curve radius, profile, alignment, twist and gauge can have substantial influences on the amount of track induced vibrations. The rate of changes in the vibrations level at various conditions of track geometry parameters was discussed and consequently correlations between track geometry parameters and track level of vibrations were developed. Based on the results obtained, recommendations were made to set new limitations for the track geometry parameters tolerances. It was shown that consideration of the new limitations ensures that the induced track vibrations remain within the allowable limits.

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