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COMPARATIVE STUDY OF RUNNING SAFETY AND RIDE COMFORT OF RAILWAY VEHICLE

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Summary: The paper deals with the running safety and the ride comfort of a railway vehicle. It presents methods used for investigation of these essential ingredients in the assessment of the vehicle running behaviour. The present study of the running safety and ride comfort is based on the results of numerical simulations for non-linear model of railway vehicle moving along a tangent track. Importantly, the applied model takes into account lateral and vertical irregularities of the track geometry with random character. The running safety is studied according to the UIC 518 code: through the ratio of the lateral Y to vertical Q forces in the wheel/rail contact point. The lateral and vertical vehicle body accelerations, which a passenger of the railway vehicle is exposed to, are used for the assessment of the ride comfort in accordance with the ISO 2631-1 standard. It is analysed and compared how various parameters of the railway vehicle-track system affect the running safety and the ride comfort.

Keywords: railway vehicle, safety against derailment, ride comfort

1. INTRODUCTION

Running safety and ride comfort are essential elements in the analysis of the railway vehicle dynamics and they have to be taken into account in modelling and assessment of the vehicle running behaviour. The main objective of running safety is the preventing the derailment of a railway vehicle which occurs when the wheels run off the rails. The derailment can be the result of various, sometimes very complicated conditions which lead to the lost of the lateral guidance, provided by the track during the normal vehicle operation. According to the review [1] by Wu and Wilson in [2], the four main causes of the derailment are: wheel flange climb, gauge widening, rail rollover and track panel shift. The risk of derailment by flange climbing over the rail can be assessed with the over hundred-year-old Nadal criterion [3] and its more contemporary modifications [4-9] partly adopted in regulations for testing running safety [10-12]. Ride comfort is a complex concept which involves numerous factors which have adverse effects on a travelling passenger [13-14]. They include vibrations, temperature, noise, air quality, lighting

conditions and others as well as the time that the passenger is exposed to these undesired factors. The effect of vibration depends on the magnitude of acceleration suffered by a passenger, its direction and frequency [15].

The present paper studies safety of derailment by flange climbing and the vibration comfort for a passenger car by using the methods included in the UIC 518 code [12] and the ISO 2631-1 standard [16] (and also in the Polish PN-91/S-04100 standard [17]), respectively. The investigations are based on the results of the numerical simulations for a non-linear model of railway vehicle moving along a tangent track with the random geometrical irregularities. The main objective is comparison how various factors, like ride velocity, suspension parameters and track irregularities, affect the running safety and the ride comfort.

2. RAILWAY VEHICLE - TRACK MODEL

The passenger car studied in this paper is described with the non-linear model of a railway vehicle which consists of seven rigid bodies: the car body, two bogie frames, and four wheelsets (27 degrees of freedom) [18]. The railway vehicle moves along a stiff, tangent track with constant speed v . The primary and secondary vehicle's suspensions are assumed to have linear characteristics. The applied stiffness constants and damping coefficients as well as masses and inertia moments of the car body, bogies and wheelsets are the same as previously used in [19,20]. The forces at the wheel/rail contact are found with the algorithm developed by Kalker within the simplified nonlinear theory of contact [21]. The positions and geometry of the contact points are determined by using the non-linear wheel (UIC 60) and rail (ORE S1002) profiles. The model includes the geometrical track irregularities $\xi = \xi(x)$ represented by: $y_w(x)$, $z_w(x)$ - lateral and vertical irregularities of the track centre line, $2l_0(x)$ - variable track gauge and $h_w(x)$ - local superelevation of the track. These parameters are random quantities which vary with position x measured along the track centre line and as such they can be treated as stationary and ergodic stochastic processes. The realizations of the track irregularities are directly included into the equations of motion. These equations are solved numerically to find the coordinates $\mathbf{y}(t) = (y_1, y_2, \dots, y_{27})$ describing the positions and rotations of all vehicle elements at a sequence of N equidistant times $t = i\Delta t$ during the vehicle motion over the track section of the length L ; thus total duration of the simulated motion is $T = L/v$ and the time step is $\Delta t = T/N = \Delta L/v$ where $\Delta L = L/N$. The realizations of track irregularities applied in simulations have the length of 4000m and their respective standard deviations are equal to $\sigma_{yw} = 0.003\text{ m}$, $\sigma_{zw} = 0.0033\text{ m}$, $\sigma_{2l} = 0.0011\text{ m}$, $\sigma_{hw} = 0.0015\text{ m}$. The power spectral densities of $y_w(x)$, $z_w(x)$ are shown in Fig. 1, the two other densities are given in [20].

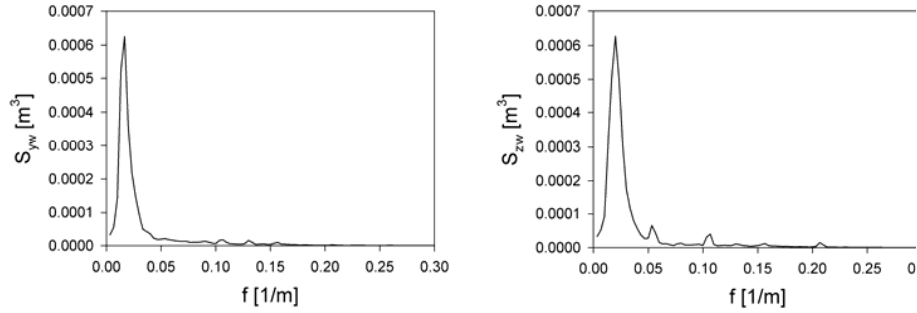


Fig. 1. Power spectral density of lateral (y_w) and vertical (z_w) irregularities of the track centre line vs spatial frequency f

3. ANALYSIS OF RUNNING SAFETY

As it has already been mentioned in the introduction, there are several causes of the derailment of a railway vehicle. One of the main scenarios of derailment is realised when during the vehicle motion a large lateral force acting on a wheelset leads to the wheel flange contact with the rail – as a result flange climbs up the rail rapidly (especially after the contact angle attains its maximum value) and the wheelset derails. The occurrence flange climb is related to the ratio Y/Q of the lateral to vertical force components at the wheel-rail contact.

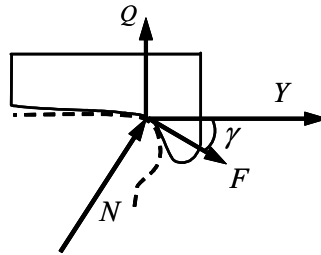


Fig.2. Force components at a wheel/rail contact: lateral (Y) and vertical (Q), normal force (N) and lateral rolling-friction force (F)

This ratio is used directly in the Nadal criterion [3] which gives its maximum value

$$\left| \frac{Y}{Q} \right| < \frac{\operatorname{tg} \delta - \mu}{1 + \mu \operatorname{tg} \delta} \quad (1)$$

obtained under the condition of the saturated friction force (depending on the friction coefficient μ) at the contact point and calculated for the maximum flange contact angle δ . The Nadal criterion is easy to implement and it is widely used for assessment of the safety against derailment. In particular, this criterion is applied – in a modified form – in the UIC 518 code [12] which is used for testing and approval of railway vehicles. The main modification adopted in the UIC 518 code is the requirement that the ratio Y/Q exceeds the assumed critical value of 0.8 over a 2m track interval. This is done because derailment of a vehicle can take place only if the ratio Y/Q exceeds the limit value for a sufficiently long time interval. Similar requirements of the minimum duration of the ratio Y/Q exceeding the Nadal limit value are imposed in other regulations regarding the running safety that are used for certification testing of railway vehicles. In particular, the time limit of 50 ms is adopted together with the limit value of 1.0 for Y/Q by Association of American Railroads (AAR) [10], while the flange climb distance limit of 5 ft is adopted by the Federal Railroad Administration (FRA), U.S., [11] for high-speed tracks of the class 6 and higher. Modifications of the Y/Q limit by increasing its values for short duration of the lateral force impulse have been proposed in the reports of Japanese National Railways (JNR) [4] and General Motors Electromotive Division (EMD) [5]. The occurrence of wheel climbing also strongly depends on the angle of attack (the yaw angle ψ of the wheelset) so that the limit Y/Q value should be increased for ψ smaller than 5 mrad as it was proposed by Elkins and Wu [7] and investigated further in the reports [8,9]. These works also propose bi-parameter expressions for the flange climb distance as a function of both ψ and Y/Q . Another extension of the Nadal criterion is obtained when, instead of the Y/Q ratio for flanging wheel only, the sum of the Y/Q ratios for both wheels on the same axle is considered. The resulting Weinstock [6] criterion is less conservative and regarded to be more accurate than the Nadal criterion, but it is not so widely applied.

In the present work, safety against derailment is assessed with the Nadal criterion modified according to the UIC Code 518. In the first step of the safety analysis, the ratio Y/Q obtained from simulations is averaged at each track point x over the surrounding 2m track section (window) $[x-1\text{m}, x+1\text{m}]$. Thus, the running average $(Y/Q)_{2m}$ is calculated – it is done to satisfy the discussed requirement of minimum flange climb distance necessary for derailment. Further, as it is recommended in the UIC Code 518, the 99.85 percentile value $(Y/Q)_{2m}|_{0.9985}$ is found. The obtained values $(Y/Q)_{2m}|_{0.9985}$ have been compared to the limit value 0.8 adopted in the UIC Code 518 [12]. This method of assessment of safety against derailment - described in more detail in [22, 23] - is presented schematically in Fig.3.

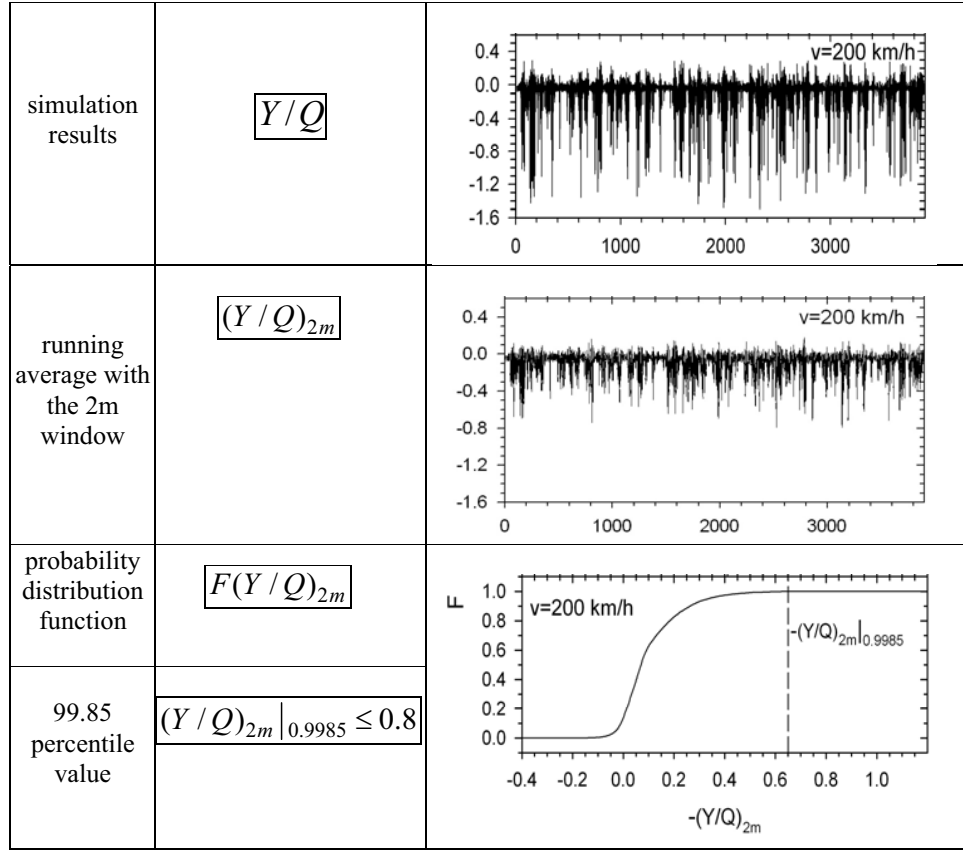


Fig.3. Scheme of evaluation of safety against derailment according to UIC Code 518 [12] and the illustration of its consecutive steps based on the simulation results ($v = 200$ km/h)

On the basis of the simulation studies [22-24], it can be concluded that the values of $(Y/Q)_{2m}|_{0.9985}$:

- grow with the increase of the ride velocity but they do not exceed the limit value 0.8 for the investigated riding speeds in the interval from 80 to 200 km/h (see Table1),
- grow with the increasing amplitude of the lateral irregularities, while the effect of the vertical irregularities is negligible (see Table 2); $y_{w,exp}, z_{w,exp}$ - denote experimental realizations,
- are practically identical for the front and rear bogies, and significantly larger for the leading wheelset than for the trailing one.

Changes of stiffness and damping coefficients of the vehicle suspension modify the values of Y/Q very slightly, except for the case of the breakdown of the damper in the

vertical primary suspension. The effect of ride velocity, track irregularities and various suspension parameters is summarized in Table 3.

Table 1

The 99.85 percentile value $(Y/Q)_{2m}|_{0.9985}$ for various ride velocities v .

bogie	wheelset	$v=80$ km/h	$v=120$ km/h	$v=160$ km/h	$v=200$ km/h
front/rear	leading	0.266	0.321	0.411	0.650
	trailing	0.105	0.134	0.236	0.482

Table 2

The 99.85 percentile value $(Y/Q)_{2m}|_{0.9985}$ for various amplitudes of the lateral y_w and vertical z_w track irregularities; ride velocity $v=160$ km/h.

bogie	wheelset	$y_w = y_{w,exp}$ $z_w = z_{w,exp}$	$y_w = 2y_{w,exp}$ $z_w = z_{w,exp}$	$y_w = 3y_{w,exp}$ $z_w = z_{w,exp}$	$z_w = 2z_{w,exp}$ $y_w = y_{w,exp}$	$z_w = 3z_{w,exp}$ $y_w = y_{w,exp}$
front/rear	leading	0.411	0.538	0.670	0.414	0.418
	trailing	0.236	0.377	0.447	0.237	0.237

4. RIDE COMFORT

The passenger ride comfort related to vibrations is of vital importance among a variety of other factors involved in comfort evaluation. The principal quantity that is relevant to the vibration aspect of the ride comfort is the acceleration that the passenger is subject to during the motion of a railway vehicle. The perception of the ride comfort depends on both amplitude and frequency f of the suffered acceleration as well as on its direction (lateral y or vertical z).

The frequency dependence arises since various organs in human body are sensitive to vibrations from different frequency ranges. In the standard ISO 2631-1 [16] and the Polish standard PN-91/S-04100 [17], the ride comfort is evaluated quantitatively on the basis of lateral and vertical components of acceleration a_y , a_z (measured or obtained in simulations). It is done by determining the rms acceleration values $a_{y; rms}(f)$, $a_{z; rms}(f)$ for the centre frequencies of the 1/3 octave bands chosen in the ISO and PN standards [16,17].

Table 3.
Effect of ride velocity, vehicle suspension parameters and track irregularities on running safety and ride comfort. Blue up and down arrows denote increase or decrease (respectively) of the analysed parameter values.

Effect on:			RUNNING SAFETY	RIDE COMFORT	
parameter	direction	parameter change	$(Y/Q)_{2m} _{0.9985}$	$a_{yb;rms}(f)$	$a_{zb;rms}(f)$
velocity	along track	 v	strong	strong	strong
track irregularities	lateral	 y_w	significant	strong	negligible
	vertical	 z_w	negligible	negligible	strong !
suspension parameters	lateral	 k_{zy} primary suspension	negligible for leading wheelset significant for trailing wheelset	significant	negligible
		 c_{ny} secondary suspension	negligible		
	vertical	 k_{nz} secondary suspension	negligible	negligible	significant
		 c_{nz} primary suspension			
		$c_{zz} = 0$ primary suspension	significant	negligible	very strong !

Subsequently, the obtained rms values are compared with the reduced comfort boundary, the fatigue-decreased proficiency boundary, and the exposure limit [16,17]. Other methods for evaluation of the ride comfort related to vibrations, like those applied in the standards UIC 513R [25] and ISO 10056 [26], can also be used and synthetic indices of ride comfort can be obtained from the rms accelerations by using suitable frequency-dependent weighting functions. A comparison and correlation between the results of the methods used in various standards is discussed thoroughly in [14, 27].

The accelerations, experienced by a passenger and used for assessment of the ride comfort according to the ISO 2631-1 standard [16], are usually identified with the lateral and vertical accelerations of vehicle body (centre of mass) a_{yb}, a_{zb} . The dependence of the ride comfort on the passenger's position can also be studied by using accelerations at various locations within the car body, see, e.g., [18]. In the reported research, the accelerations a_{yb}, a_{zb} have been obtained numerically by solving the equations of motion over the track interval $0 \leq x \leq 4000$ m. The rms accelerations values of $a_{yb;rms}(f)$, $a_{zb;rms}(f)$ in 1/3 octave bands $(f - \Delta f / 2, f + \Delta f / 2)$ of width $\Delta f = 0.231f$ are found with the numerical Fourier transformation [17]:

$$a_{\eta b;rms}(f) = \left(\int_{f-\Delta f/2}^{f+\Delta f/2} \frac{2}{T} \left| \int_0^T a_{\eta b}(t) e^{i2\pi f' t} dt \right|^2 df' \right)^{1/2} \quad (\eta = y, z); \quad (2)$$

the band centre frequencies f are within the interval from 0.1 Hz up to 80 Hz as assumed in the standards [16,17]. For each 1/3 octave band, the obtained $a_{yb;rms}(f)$ and $a_{zb;rms}(f)$ will be compared with the boundary values for reduced comfort and fatigue-decreased proficiency given in this standard.

The obtained results [18,22-24] are analysed in a synthetic way in Table 3. In particular, it shows that both the increasing vehicle velocity and growing track irregularities lead to the increase of vehicle body acceleration and, as a result, to the decrease of the ride comfort. As an effect of these factors, values of the body accelerations can exceed the reduced comfort boundary for frequencies below 5 Hz, but they usually stay below the fatigue-decreased proficiency boundary. The frequency values corresponding to the maximums of the rms lateral body acceleration $a_{yb;rms}(f)$ increase, roughly linearly, with the increasing velocity, while the position on the frequency axis of the maximum vertical acceleration $a_{zb;rms}(f)$ does not change. The positions of these frequencies do also not depend on the magnitude of the track irregularity amplitudes.

The nominal values of parameters of the lateral stiffness of secondary suspension and the vertical damping of primary suspension seem to be chosen optimally with respect to the ride comfort. Similarly, it can be concluded that the nominal value $c_{zy} = 0$ (no damping in the lateral primary suspension) is chosen properly since changing c_{zy} has been found to have no significant effect on the body acceleration. Lateral (a_{yb}) and vertical (a_{zb}) body accelerations depend significantly on the position in the car body. It has been found in [18] that values of accelerations at points distant from the body centre of mass are even a few times larger than at the body centre of mass. The largest effect is found when the values of the body acceleration are compared between different points in the longitudinal symmetry plane. The change of the points within the lateral symmetry plane has much smaller effect and it is present only for the vertical acceleration.

5. CONCLUSIONS

The obtained results lead to the expected conclusion that with increasing ride velocity there is an increasing risk of derailment and a decrease of the comfort. The effect of the track geometrical irregularities on the running safety is different than in the case of the ride comfort. For the running safety, the effect of the vertical irregularities on the values of $(Y/Q)_{2m}|_{0.9985}$ is negligible, while the magnitudes of both the lateral and vertical track irregularities have strong effect on the vehicle body accelerations, which determine the ride comfort. The investigation of the rms body acceleration for the chosen points of the car body, allows us to evaluate and analyse the ride comfort for the passenger at various locations.

Summarizing, the performed analysis shows that the effect of various parameters of the railway vehicle-track system is very diverse both for the running safety and the ride comfort. Simultaneously, the two investigated aspects of the running behaviour can be

affected by changes of particular parameters in a very much different way. This confirms the conclusion that an improvement of running safety can sometimes result in a decrease of the passenger's ride comfort.

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ANALIZA PORÓWNAWCZA BEZPIECZEŃSTWA I KOMFORTU JAZDY POJAZDU SZYNOWEGO

Streszczenie. Praca dotyczy zagadnień bezpieczeństwa i komfortu jazdy pasażera, stanowiących istotne elementy analizy dynamiki pojazdu szynowego oraz przedstawia metody ich badania. Analiza bezpieczeństwa i komfortu jazdy zawarta w niniejszej pracy została przeprowadzona na podstawie rezultatów symulacji nieliniowego modelu pojazdu szynowego poruszającego się po torze prostym. Istotną cechą użytego modelu jest uwzględnienie poprzecznych i pionowych nierówności toru o charakterze losowym. Bezpieczeństwo przeciw wykolejeniu badano według normy UIC 518, obliczając iloraz siły poprzecznej Y do pionowej Q w punkcie kontaktu koło/szyna. Ocenę komfortu jazdy zgodnie z normą ISO 2631-1 przeprowadzono na podstawie wyznaczonych numerycznie poprzecznych i pionowych przyspieszeń nadwozia. Przedstawiono analizę porównawczą nierówno-miernego wpływu parametrów układu pojazd szynowy-tor na bezpieczeństwo i komfort jazdy.

Słowa kluczowe: pojazd szynowy, bezpieczeństwo przeciw wykolejeniu, komfort jazdy

Recenzent: Włodzimierz Choromański