ANNALS OF FACULTY ENGINEERING HUNEDOARA
– International Journal of Engineering
Tome XI (Year 2013) – FASCICULE 4 (ISSN 1584 – 2673)

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CONSIDERATIONS ON THE STRUCTURAL NOISE AND VIBRATION CAUSED BY THE OSCILLATIONS OF RAILWAY VEHICLES FOR HIGH SPEED MOVEMENT

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ABSTRACT: Lateral oscillation of railway vehicles is a major problem for narrow gauge railways or metric because reduced diameter bogie wheels. This problem can be reduced by changing the Prud'homme limit on vehicles and improving resistance tread lateral, being able to use the checkrails. Running with lateral oscillations road of the railway vehicles passenger and freight is a known problem for standard gauge railways. The lateral movement represents a particular problem which causes passengers discomfort and in major cases, the operating speed must be reduced. Whilst the railways and trains may suffer because of this. A possibility to resolve these problems should as reducing the Prud'homme limit, which defines the permissible lateral load, repeated exerted on the rail. This formula was used for vehicles and standard gauge railways, but many companies and suppliers of rolling stock have assumed that can be applied on narrow gauge lines. On the basis of existing information, without being made field tests or computer simulations, Prud'homme limit can be modified to standard gauge rail networks, but to verify the load lateral of this type of gauge must be made more investigations. Inevitably have emerged more simplistic assumptions, but is unlikely to significantly alter the conclusions.
KEYWORDS: railway vehicles, structural noise, vibration, high speed movement

INTRODUCTION

The dynamic variable lateral forces existing between the vehicle and the rail track can be generated and the profile wheels which are designed to control lateral movement of the wheel and improve the wheel - rail. The suspension side and vehicle running characteristics contribute to lateral forces and the form of the oscillation movement may be controlled by design and efficient maintenance. The running characteristics depend on the suspension and accepted play between the axle, bogie and the vehicle structure and the effect of the engine axles. The railway geometry represents a critical element and effective maintenance is essential to reduce lateral oscillations, and this is further facilitated by a high standard of construction, a large mass per unit length and the corresponding rail boards. Other rail or vehicle defects may arise in the excessive swaying of the vehicle, its gallop or the vertical instability, all amplifying the lateral forces.

Under the pressure of lateral forces, the track can yield elastically and cause temporary deformation, can become deformed or has not returned to its original shape, depending on the size of the lateral force. Strains that remain are called the deviations from the zero scratch. Whether these deformations are accumulated due to the passage of rolling stock frequently poorly maintained, is very likely that will deteriorate gradually rail and lateral oscillations provoke of other vehicles. This would result to a vicious cycle which leads to wear or even destruction of the track and the vehicles suspension.

GENERAL ASPECTS BY THE LATERAL FORCES AND THE DYNAMICS OF WHEEL - RAIL, LIMITATION OF LATERAL FORCES

The research carried out by André Prud'homme have concluded that if the lateral force exercised by the axle on track path itself does not repeatedly exceeds a certain value H, the deviations from the scratch, are gathering indefinitely and ultimate deviation is determined acceptable limits. The limit value is represented by the value of force respectively $H = 10 + 0.33\ P$ [kN], where H represents the lateral force while P [kN] is the static axle load. This relationship was originally conceived for the rail with wooden timber sleepers in ballast fastened without thermal variations. Force H is the considered resistant of checkrails limit to acceptability criteria are met the deviations from scratch under repeated lateral load pressure competing with vertical load P. Prud'homme later has extended his researches to keep into account the thermal variations of welded the rail and applied a multiplying factor of 0.85, and the final form of the Prud'homme limit is $H = 0.85\ (10 + 0.33\ P)$.

The force H is defined as the maximum the lateral force that can be exercised over repeatedly the rail vehicle. The Prud'homme limit has not been exceeded during repeated measurements made to assess the maximum lateral forces exerted on the rail when tested at speeds of 308 km/h and 322 km/h. These have determined the validation used the Prud'homme limit by designers and suppliers of rolling stock for track gauge of 1435 mm and established tolerances accepted for vehicle maintenance. On the basis of recent investigations, SNCF has limited the lateral force resistance vehicle track system TGV lines. All these means that the lateral load, dynamic, repeated exercised vehicle axle must not exceed the value: $H = 0.85\ (10 + 0.33\ P)$. Concrete railway sleepers lines must be made in such a way, as well as static resistance of lateral checkrails may not be less than the value $H_1 = (24 + 0.41\ P)$ for railway track themselves buried and respectively $H_2 = (38 + 0.63\ P)$ for the dynamic stabiliser rails.

The lateral rails strength represents the maximum load side stationary the rail can be supported under vertical load of the vehicle without causing permanent deflection of the rails, or in the case of using the metal bridge as shown schematically in Figure 1. For example, when P is the axle load of 150 kN, the limit values for standard gauge the rail are basically following namely maximum lateral force exerted by the axle whose value is calculated using the formula $0.85\ (10 + 0.33\ P) = 51$ kN, the minimum resistance of rails in the case of bury $= 24 + 0.41.P = 85.5$ kN and last but not least, the minimum resistance rails after using stabilizer, which is used for finding the following empirical relationship $38 + 0.63.P = 132.5$ kN.

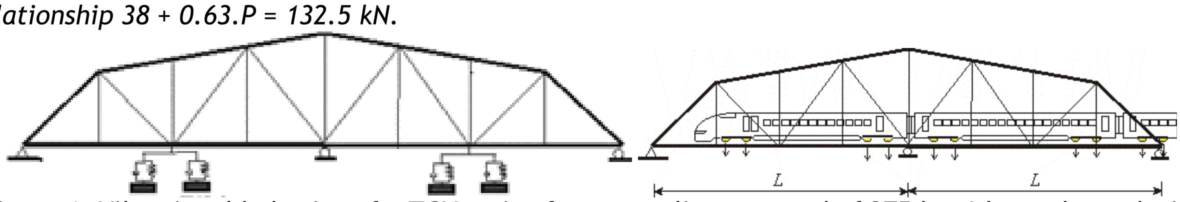

Figure 1. Vibrational behavior of a TGV trains from traveling at speed of 275 km / h overlapped with undulatory phenomena induced vibration and a metal bridge deck length on L

A reasonable limit, safety, between the resistance of and the pressure lateral rails running increases the comfort and increases the safety, ensuring at the same time the efficiency and the optimal maintenance of rail and vehicles. Therefore the Prud'homme limit became a starting point for construction engineers railway track and railway vehicles. One of the interaction safety limits wheel - rail is that the vehicle can not produce the lateral load per axle over static vertical (H / P) greater than 0.5. For a nominal axle load of 140 kN, horizontal the lateral force allowed is 70 kN, which is 50% higher than the Prud'homme.

THE LIMITS OF THE RAILWAY LINE

There are two possible independent approaches: one based on the thick side rail and the second on the axle load proportional. Taking the first approach, the thickness of the side rail ballast resistance and resilience the framework are. The verge trimming ballast resistance includes the resistance of ballast and sleepers heads acting frictional resistance acting on the sides and bottom of the beams. The resistance of the ballast embankment can be considered to be the same for both the standard gauge and the gauge meter, whereas the cross sectional areas of the beams are the same. The sleepers cross bars for standard gauge resistance is a function of the ratio between the lengths of the beams of 2.0 to 2.5 m. The resistance of the bogie frame can be considered to vary approximately in the same proportion as the distance between pivots, for instance, from 1065 mm to 1500 mm, resulting 0,71:1. Therefore, the strength of the bogie frame of a railway track metric may be considered to be 0.7 in the normal track. As a simplified approach, both components were given equal importance. The total lateral thickness gauge railways such metric is (0.9 + 0.7) / 2 or 0.8 of the normal track. The horizontal force H acting on the center of gravity, the height C of the guide bar, the resistance from the horizontal forces H1 and (H - H1) acting on the rail. It is also creates a variation of the axle load R = H c / g, where g is the distance between the center line. The increased axle load is the Q + R are actually dynamic overload outer axle and axle load is the Q-R. External load is calculated as the proportional R / Q = Hc / Qg.

Table 1. Results obtained from measurements to determine acoustic vibrations propagate at the speed of 275 [km / h] of a TGV over a metallic bridge (Figure 1)

	Values	
30" [760mm]	ls	kg
The standard model		
30" x 30" [760 x 760mm]	850	380
30" x 36" [760 x 910mm]	850	380
30" x 48" [760 x 1210mm]	750	330
30" x 60" [760 x 1520mm]	650	290
The high capacity model		
30" x 30" [760 x 760mm]	2,600	1170
30" x 36" [760 x 910mm]	2,600	1170
30" x 48" [760 x 1210mm]	2,500	1120
30" x 60" [760 x 1520mm]	2,400	1080

When the H force reaches the limit for normal gauge Prud'homme will result in a certain amount of external load has been shown that it does not even discomfort while running, no derailment. Therefore it is advisable to adopts the same forward level of external load equal to the upper limit. It should be noted that the limit Prud'Hommme can be increased only by improving the sustainability of *the railway side, whereas the effects of external load can also be critical. In this case, increasing sustainability metric gauge track to be completed by low vehicle center of gravity height.*

On the majority railway infrastructure, passenger and freight trains share the same line. The freight vehicles produce damage lines, leading to a less smooth running of passenger trains and the possible economic consequences. For this reason, Prud'homme limit change to be implemented uniformly for both passenger and freight wagons and locomotives. Using this information, the resistance under static side rails railway track with concrete sleepers metric can be calculated by multiplying the appropriate gauge 0.8. The calculated values for metric gauge railways are (19 + 0.33 P) for recently buried rails and ballast prism (30 + 0.5 P) for tracks that have already been stabilized.

Table 2. The values of measured in dB sound intensity after smoothing acoustic vibrations

Noise level [dB]	Linear smoothing	Noise level [dB]	Linear smoothing
0	1	30	31
3	1.4	36	60
6	2	40	100
10	3.1	50	310
12	4	60	1000
18	8	70	3100
20	10	80	10,000
24	16	100	100,000

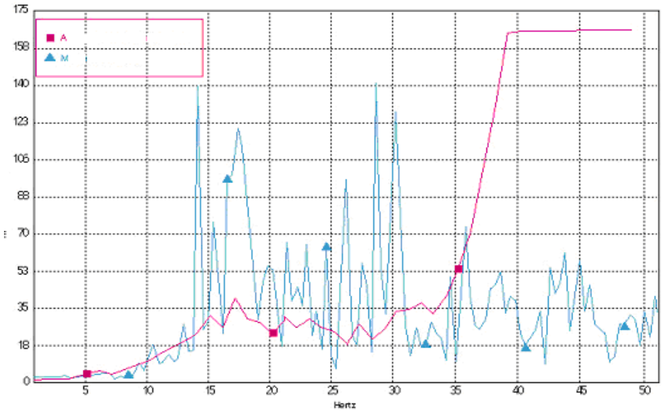

Figure 3. The weightings of vibration frequency spectrum. A = The threshold limit value of level vibrations that should not be exceeded; M = The level reached of vibration resulting from the experimental measurements

Some analyzes suggest that the swinging movement of trains, provide quality-related benefits provided runtime. This may be due to lateral forces exerted on the track, set the track reaching normal levels. Tilting mechanisms are designed to improve passenger comfort by compensating for the centrifugal force unstable although they do not reduce the lateral forces acting on the track. Principle operational solutions to reduce rolling noise is short term in reducing defects "roughness" on wheels and on rails, so to act upon excitation. Frequency weighting curve based approximate range values respectively for frequencies between 1-80 Hz. Frequency weightings in Table 2 are close to the curves in Figure 3 for an approximate quantification of the effects of the different frequencies of vibration on discomfort people or the interference with activities. One of the most ingenious methods of reducing noise caused by high-speed train traffic is to use fitted with axle bogies whose wheels have a movement independent of each other, advanced powertrain systems whose operation is substantially diminishes the prospect of instability terms of the flow oscillations occurring in curves. But not decrease system and does not involve reducing hunting type oscillations, whose nature arises even in the case of ideal self in alignment and landing.

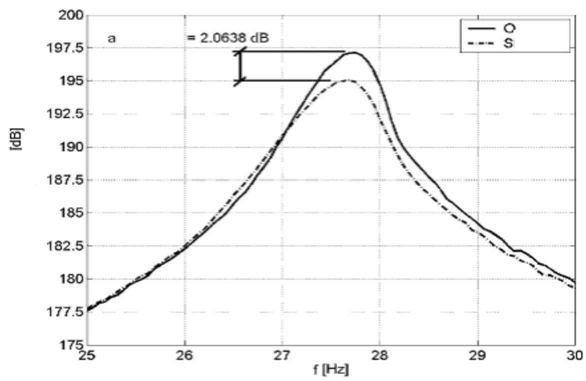

Figure 5. Variation of critical speed noise from traffic or train high speed

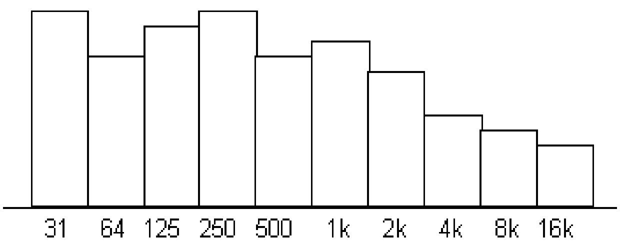

Figure 6. Simplified Diagram own pulsation and gallop bounce oscillations measured in dB

From a mathematical perspective, the movement of the axle oscillations in the curve are of unstable oscillations that occur in the wheel - rail contact where friction forces interact and combined with lateral sliding friction wheel in the curve where the track gauge widens slightly from the baseline level of alignment. In practical experiments it was found that the rate of lateral movement of the axle to flow in curves is constant. It was also found that the sliding friction forces are proportional and in phase. Theoretically, the noise caused by the movement of the vehicle wheel vibrations can be reduced or even eliminated only by preventing disturbing factors causing maximum amplitude of oscillations due and powerful phenomena encountered friction between the wheels and tires at high speed traffic.

An important source of noise is the vibration induced wheel and worn wheel tread. Noise levels is dominated by vibration wheel vibrating. Current models, the wheel resonance frequencies are in the range 1000 - 3000 Hz, the human hearing is most sensitive. From calculations revealed that for determining and identifying the actual vibration modes of the disc wheel and oscillating system stability, an important share a strong influence and factors that have to be constantly maintained. Also perceived discomfort caused by noise is reduced if the resonant frequency of the wheel is moved above the 3000 Hz [1]. Another way to reduce noise is to use quieter braking systems. Normally freight trains are equipped with cast iron brake blocks, while passenger trains are equipped with disc brakes. Experience has shown that the friction brakes of iron, already after a short use, cause wear of the wheel, to the profile of the wear, which generates noise in the range of 800-2000 Hz. Measurements, for example, shows the difference in sound pressure level of 10-15 dB in the frequency range. The most effective measure for reducing noise is usually directed towards the noise source, in this case, the wheel - rail. Research indicates a difference in the level of noise of 15 dB (A) between a rail and a worn smooth as possible. One way to reduce the noise caused by the wheel - rail is increasing vibration energy dissipation. The new concept line, developed for heavy and high-speed lines, the track is suspended on a continuous elastic foundation or concrete. In most developed systems, a substantial part is molded rubber track. In some cases, the propagation area is reduced by about 50%, implying a possible reduction in the acoustic energy of 3 dB in the vicinity of the line, where all the other elements are the same.

The sound can propagate through the three environments or gas liquid and solid, and in terms of quantum, it can be identified and the fourth state of matter, namely plasma, but as a paradox, he they can propagate in vacuo. The sound is produced by compression and decompression layers from the environment (disturbance is transmitted step by step through the elastic, creating elastic waves, which take the form of alternating compressions or rarefaction).

It is worth mentioning that any vibrating body can be a source of elastic waves in the environment where they are, that can be a sound source. From the analytical point of view, the general form of the equations of forced vibrations of a system with viscous and hysteretic damping is of the form $[M]\{\ddot{q}\}+[C]\{\dot{q}\}+\frac{1}{\omega}[H]\{\dot{q}\}+[K]\{q\}=\{f\}$. *If the excitation is harmonic then* $\{f\}=\{\hat{f}\}.e^{i\omega t}$ *and* $\{q\}=\{\widetilde{q}\}.e^{i\omega t}$, *and if there is a column matrix* $\{\hat{f}\}$ *the actual elements which define an excitation force "in phase" so that any pulsation* ω, *the displacements* q_j *all be "in phase" with each other but not mandatory and forces* $\{\widetilde{q}\}=\{\hat{q}\}.e^{i\varphi}$, *where* φ *is the phase difference between force and displacement), then the matrix* $\{\hat{q}\}$ *has its merits and to determine methods to eliminate the phenomenon of resonance, the following issues should be considered, namely that resonance occurs when the dominant frequency of the input signal (ie when the frequency leading to extreme values of the power spectral density of the excitation) coincides with the natural frequency of the oscillating system.*

If the input signal is characterized by a damped harmonic correlation, then the system will show an increase of vibration, which will be manifested by a maximum mean square acceleration when the frequency is close correlation harmonic frequency of the oscillating system . In this case, the power spectral density of the input signal is of the form:

$$S_{x_0}(\nu)=\frac{\sigma_{x_0}^2}{\pi}\frac{\alpha_1\left(\alpha_1^2+\beta_1^2+\nu^2\right)}{\nu^4+2(\alpha_1^2-\beta_1^2)\nu^2+\left(\alpha_1^2+\beta_1^2\right)},\ \text{for } |\nu|\prec\nu_1 . \tag{1}$$

In the case of a system dry friction dissipative as appropriate suspension coil spring, the equation of motion can be described as: $\ddot{x}+R\,\mathrm{sgn}\,\dot{x}+g(x)=0$, *where* $g(x)$ *represents the elastic force and is a positive constant. Likewise,*

$$\mathrm{sgn}\,\dot{x}=\frac{|\dot{x}|}{\dot{x}}=\begin{cases}1; if:\dot{x}\succ 0\\ -1; if:\dot{x}\prec 0\end{cases} \tag{2}$$

Also, if in this case, the driver no matter regarded is balance. In this case, the speed is positive first and negative second semioscilation and later, after determining extreme stretching can be determined times. If you know the spectral density of a stationary random function w *of "* j *"often differentiable, then the standard deviation of this function and its derivatives up to order* m *including quadrature are determined by the following:*

$$\sigma_{w^{(j)}}^2=2\int_0^\infty S_{w^{(j)}}(\nu)d\nu=2\int_0^\infty \nu^{2j}S_w(\nu)d\nu , \tag{3}$$

where the function $S_w(\nu)$ is a symmetrical function and represent the power spectral density of $p_T(\nu)$ which is stationary, ergodic and implementation time by Fourier transform. It is worth mentioning that satisfy the previous conditions, thus leading to the appearance of white noise, Fourier integral $x_0(t)$ being divergent and therefore identification the function $p(t)$ white noise consists in explicitly defining white noise as random function $x(t)$ which has identically zero mathematical expectation, specifying the symbol δ Green function, Dirac distribution $G(t)$ a nonrandom function representing the intensity of white noise. If you consider a random variable X that may only take real values and has the probability density $f_x(x)$ then, if $\alpha_r = \int_{-\infty}^{\infty} x^r . f_x(x)dx$ si $r \in N$, so α_1 is called the math expected value of x and is denoted by m_x. In the this case, the function $S_{x_0}(\nu)$ represents the power spectral density of the function $p(t)$. To note is that the expected value if considered random signal is identically zero, then the function that describes the input signal is retarded potential. In the general case, the correlation function and power spectral density of stationary random function $x_0(t)$ the spectral band and diminishes the power spectral density of various sizes considered as outputs of an oscillating mechanical system given. Likewise, the functions of $S_{x_0}(\nu)$ and $S_{\ddot{x}_0}(\nu)$ represents power spectral density of the input that the standard deviation of a stationary random the function "j" or derivatives, previously determined. If the random function $x_0(t)$ has an exponential correlation, the average squared correlation evolves and if amortized harmonic, while making the change of variables, then if the correlation function is a linear combination of exponential correlation between damped harmonic correlation in mean square of the response appears intensified regime vibration, which will be manifested by a maximum mean square acceleration when the frequency is close correlation harmonic frequency of the oscillating system. In this case, it was considered as an oscillating system analysis model with one degree of freedom to the dissipative force is proportional to some power of the oscillation speed $\dot{x}$, by the next form: $P = k\,\mathrm{sgn}\,\dot{x}|\dot{x}|^i$. For this oscillating system, the input signal is represented by a random process, which takes an exponential correlation and therefore is simplified differential equation of motion of the form $\ddot{x} + A\,\mathrm{sgn}\,\dot{x}|\dot{x}|^i + \omega^2 x = -\ddot{x}_0$ where $A = k/m$.

Whether the oscillating system could be described by the equation: $x(t) = A.x_0(t) \Rightarrow x_0(t) = A^{-1}.x(t)$, where $x_0(t)$ represents external signal acting on the system, then the function $x(t)$ is the mechanical system responds to external action and the parameter A is the set of elements that compose the the oscillating mechanical system considered. When would consider that the term A^{-1} is an linear differential operator of the following form:

$$A^{-1} = \sum_{i=0}^{n} a_i(t)\frac{d^i}{dt^i}, \quad (4)$$

then A there is linear and for the determination of a linear mechanical system, use linearization methods for operators equations describing motion (oscillation system).

To determine the oscillation of a high speed motor vehicle (for example, electric locomotive class BR 182) during its movement will take into account that it runs (in addition to the main forward movement) and a series of movements that tend limiting parasite the movement of the vehicle.

The purpose of determining the strain wave action disturbing the runway vertically, determine the resultant of all external forces and static moment relative to an axis through the center of gravity of the vehicle.

Table 3. Determination of sound vibration amplitudes propagated in the environment (measured in dB), depending on the frequency

dB	Amplitude [mm]	dB	Amplitude [mm]	dB	Amplitude [mm]
60	.0006	90	.018	120	.56
62	.0007	92	.022	122	.70
64	.0009	94	.028	124	.88
66	.0011	96	.035	126	1.1
68	.0014	98	.044	128	1.4
70	.0018	100	.056	130	1.8
72	.0022	102	.070	132	2.2
74	.0028	104	.088	134	2.8
76	.0035	106	.11	136	3.5
78	.0044	108	.14	138	4.4
80	.0056	110	.18	140	5.6
82	.0070	112	.22	142	7.0
84	.0088	114	.28	144	8.8
86	.011	116	.35	146	11.1
88	.014	118	.44	148	14.0

Due to the movement of the contact point wheel - rail or wheel center and the boundary points of arcs on a sinusoid (a phenomenon that leads to the raising and lowering alternative wheel respectively vertical to the horizontal plane) of the vehicle suspension assembly this box is would be transmitted through the suspension a disruptive force that can be decomposed into a Fourier series whose fundamental harmonic has the form: $z_s = (h/2).\sin((2.\pi)/l_s).x$*, for which the largest amplitude is equal to* $L = h/2$ *and the wavelength is equal to the length of the track (Figure 1).*

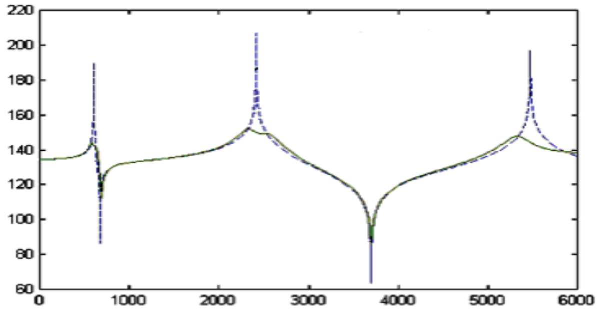

Figure 7. The oscillation frequencies in specific vertical locomotive BR 182 (Vmax = 350 km / h)

CONCLUSIONS

Has been demonstrated that high speed trains can run at a higher speed and can reduced the Rayleigh wave propagation speed in the wave function of backfill soil. Strong vibrations are generated into the soil. Train speed is also close to the so called critical speed which occurs due to the interaction of backfill and ground. The critical velocity, displacement and, therefore, velocity and acceleration will be high embankment.

The environmental problems caused by freight and passenger trains are probably more important than those caused by high-speed trains. The consequence is that vibrations with frequencies higher than normal must be taken into account geotehnology as human being is sensitive to vibration frequencies up to 80 Hz. Normally only frequencies up to 20 Hz are considered in geotechnology. Many problems jolts include many factors, such as an energy source environmental issue that transmits energy from an object, a receiver. In terms of geotechnical train rails, bed ballast, sleepers and backfill embankment are considered the source. The medium is the soil, backfill is placed and in which it is above the ground. Object of the receiver, it is often a house or more.

A railway vehicle running causes vibrations propagated path structure and in adjacent lands. There are two basic mechanisms of task qvasistatical and irregularities in the system moving wheel - rail. The vibrations with a frequency of 20 Hz are limited to systems wheel - rail and causes the dispersion of noise. Vibrations of less than 20 Hz may spread on surrounding surfaces may cause discomfort to residents and cause damage to the track, embankment and adjacent buildings. It was found that the two key factors of low frequency vibration emission are speed and weight train.

Vibration severity reach a building in proximity to railways depends on a number of factors. The most important thing is, of course, the distance to the line. Another important factor is the dynamic properties of the soil. It is known that alluvial soils, such as clay and silt mud, vibrations lead to lower frequencies and higher amplitudes than the more stable soils. Other factors are influencing soil layer depth to existence of layers of rock and earth. Not only soil, but also the characteristics of buildings such as foundation and height affect the propagation of vibrations in buildings. Measurements showed that the vibration amplitudes measured at ground level are lower than those on the upper floors. Buildings with rigid foundations are less sensitive than those with elastic foundations, etc.

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