The impact of dynamic load from the wheel on the rail for high-speed trains in Uzbekistan

Saidburkhan Djabarov and Nodirbek Kodirov

1 Tashkent State Transport University, St. Temiryulchilar 1, 100167, Tashkent Uzbekistan

Abstract. On the basis of theoretical prerequisites and experimental studies, to substantiate the choice of factors that determine the formation of a dynamic load on the track from modern rolling stock for high-speed and high-speed traffic. As a result, a method for determining the vertical design force acting from the wheel on the rail is given.

Methodology. Fundamental structural changes in high-speed and high-speed trains, a significant increase in the speed of movement lead to the need to revise the methods for calculating the effect of rolling stock on the railway track, the ratio of standard deviations of various dynamic forces, and assessing the influence of various factors for different speeds of movement. The paper also investigates the change in the dynamic load based on the results of experimental measurements. Results. The dependences of the average and calculated vertical forces on the speed of movement for modern passenger trains. Using factorial dispersion analysis, numerical characteristics of the impact of various factors on the value of the effect of the vertical force of the wheel on the rail are obtained. The degree of influence of various factors on the formation of the value of this force is established.

1 Introduction

Path strength calculations are performed to solve problems such as determining stresses and strains in elements of the upper structure of the railway track from the impact of the rolling stock, justification of the design and assessment of the track capacity according to the established operating conditions, calculation of the permissible speeds according to the conditions of the track strength, setting the temperature regime for the operation of the seamless track, etc. Their technique has a long history that can be traced back to the 19th century. Many well-known domestic and foreign scientists have contributed to the development of this direction. The current document is the industry instruction "Rules for calculating the strength and durability of a railway track" [2].

“Rules for calculating the strength and durability of a railway track” [2].

* Corresponding author: nodir_kodirov_95@mail.ru
The current trend in the development of transport networks provides for the introduction of high-speed and high-speed passenger trains on the railways of Uzbekistan [4]. Fundamental design changes in such trains and a significant increase in the speed of movement lead to the need to revise the methods for calculating their effect on the railway track [14, 15].

When designing high-speed rolling stock, two concepts of traction drive were used: locomotive (concentrated traction) - with traction engines installed on electric locomotives, which in high-speed trains, as a rule, are placed one at a time in the main and tail sections of the train (exceptions are, for example, high-speed trains X2000 (Sweden) and ICE2 (Germany) with one electric locomotive each, or an ART-R train (Great Britain), in which two electric locomotives are located in the middle of the train), and multiple unit (distributed traction) - with a relatively uniform placement of traction means along the train.

High-speed lines use rolling stock with different structures for supporting bodies on bogies: articulated, when two adjacent cars are supported by one bogie, and independent, in which each of the cars is supported by two individual bogies. The rivalry between designers who adhere to one of the two concepts continues. For example, the international concern Alstom produces trains of the TGV family only with articulated cars, while intermediate single-axle bogies are used in trains with articulated cars Talgo-250 [21].

It is believed that a decrease in the rigidity of the spring suspension, other things being equal, has a positive effect on the smoothness of the movement of railway vehicles. Therefore, the existing trends aimed at improving the smoothness of the ride are primarily characterized by a decrease in the stiffness of the spring suspension and a corresponding increase in static deflection. The use of hydraulic devices with computer control does not fit into the identical transition to the indicators of spring-mechanical systems, incorporated in the existing calculation methods.

2 Methodology

The main direction of adaptation of track strength calculations for high speeds should be considered the adequacy of taking into account the dynamics of the movement process itself. Regarding the rails, in accordance with the accepted hypothesis of Petrov N.P., the rail deflection occurs instantly, and the outline of the rail bend under the influence of dynamic overload is taken to be the bend line from static overload, numerically equal to the value of the dynamic force at a given time [9].

The limits and bordering conditions for the application of such an assumption were considered by D. M. Kurgan [8]. The dynamic load is taken into account through the appropriate definition of the design force, which is taken as the maximum probable force with a probability of not exceeding 0.994, consisting of a static load and a complex of dynamic additives, i.e.

\[ P_{раз} = \bar{P} + 2,5S \]

where \( \bar{P} \) - the average value of the force acting from the wheel on the rail; \( S \) - standard deviation of the dynamic vertical load of the wheel on the rail, kg; \( P \) - the static force acting from the wheel on the rail; \( P_p \) - the dynamic force acting from the wheel on the rail; \( S_p \) - the standard deviation of the static vertical load of the wheel on the rail, kg; \( S_{нк} \) - the standard deviation of the dynamic vertical load of the wheel on the rail, kg; \( S_{онк} \) - the standard deviation of the static vertical load of the wheel on the rail, kg.

\[ S = \sqrt{s_p^2 + S_{пн}^2 + 0,05S_{пк}^2 + 0,95S_{онк}^2} \]
Where

\( P_{\text{ст}} \) - the weight of the vehicle, referred to one wheel (static load);

\( \bar{P}_p \) - average the value of the force from fluctuations of the overspring weight of the crew;

\( S_p \) - root-mean-square deviation of the force from fluctuations of the overspring weight of the crew;

\( S_{\text{пнк}} \) - root-mean-square deviation of the force from rolling the wheel along the rail with unevenness;

\( S_{\text{инк}} \) - root-mean-square deviation of the force from the presence of an isolated unevenness on the wheel;

\( S_{\text{бнк}} \) - standard deviation of the force of the presence of continuous unevenness on the wheel.

The maximum value of the force from fluctuations in the bolster weight of the vehicle (body) can be determined through the maximum and static deflections of the springs, as well as through the coefficient of vertical dynamics \([9]\) using the following formula

\[
P_{\text{р}(\max)} = k_d (P_{\text{ст}} - q_k)
\]

Where

\( k_d \) - the weight of the unsprung part of the vehicle, related to one wheel, kN.

\( q_k \) -

\[
\begin{align*}
\bar{P}_p &= 0.75P_{\text{р}(\max)} \\
S_p &= 0.08P_{\text{р}(\max)}
\end{align*}
\]

\[
S_{\text{пн}} = 1.81 \times 10^7 \alpha_1 \beta \epsilon \gamma I \sqrt{\frac{U_k}{k} \bar{P}V}
\]

Where

\( \alpha_1 \) - a coefficient depending on the type of sleepers;

\( \beta \) - coefficient taking into account the type of rails (depends on the moment of inertia of the rail);

\( \epsilon \) - coefficient depending on the type of sleepers;

\( \gamma \) - coefficient depending on the type of ballast;

\( U \) - modulus of elasticity of the under-rail base, MPa;

\( k \) - coefficient of relative stiffness of the rail, cm

\( V \) - movement speed, km/h.

\[
S_{\text{инк}} = 0.05 \alpha_0 \xi_0 \frac{U}{k}
\]

Where

\( \alpha_0 \) -

\( \xi_0 \) -

\( \epsilon \) -
The standard deviation of the force from the presence of a continuous inequality on the wheel in accordance with \[10\] Recommended determine by formula

\[S_{\sigma \text{HK}} = \frac{1.63 \times 10^{-2}a_0U\sqrt{q_kV^2}}{d^2/kU-32k^2q_k}\]

Here \(o\) is the diameter of the wheel, cm.

As can be seen from the above formulas, the dynamic components of the force acting from the wheel on the rail have complex dependencies on many parameters. The performed studies have shown that for most varieties of modern units of high-speed and high-speed rolling stock, the dependence of the contribution of each dynamic additive to the total value of the force is characteristic. The determining initial data can be considered the speed of movement and the modulus of elasticity of the under-rail base.

3 Results

The value of vertical forces is determined using formulas (4) – (7) and presented in the form of graphic images in fig. 1 – 7.

On fig. 1 shows the results of calculations to determine the change in the values of standard deviations of dynamic forces included in practical strength calculations, depending on the speed of movement, performed according to formulas (4) – (7). At the same time, the modulus of elasticity of the under-rail base is taken to be 50 MPa [6].

Fig. 1. Change in standard deviations of dynamic forces from the speed of movement, 1 -- ; 2 -- ; 3 -- ; 4 -- ;

Analysis of dependencies shown in fig. 1 made it possible to establish that the ratio of standard deviations of dynamic forces, and, accordingly, the influence of various factors, is not the same for different speeds. So, it is possible to separate the zone with speeds up to 80-100 km/h (typical for freight trains, conditionally zone I) and zones with speeds of 120-250 km/h (zone II) and more than 250 km/h (zone III).

On fig. 2 shows the dependence of the components of the calculated force (total standard deviation and the average value of the force on fluctuations in the bolster weight of the crew) depending on the speed of movement.
Fig. 2. Change in the components of the design force depending on the speed of movement:

1 − 2,5S; 2 − P₀.

To study the change in the dynamic load from the wheel of the rolling stock, i.e. locomotives and wagons per rail, depending on the speed for passenger trains, calculations were made to determine their values using formula (10) given in [10]. The results of calculations in the form of graphic images are shown in Figs. 3-5. At the same time, it was assumed that passenger trains were driven by locomotives of the Talgo-250 and Skoda series.

The given values of the average and calculated (Fig. 3) forces were determined by the results of statistical data processing. To obtain numerical characteristics of the influence of various factors on the value of the vertical force of the wheel on the rail, factorial dispersion analyzes were performed using the results of a sufficient number of measurements for the experimental train Talgo-250 [3, 11].

Fig. 3. Dependence of the average and calculated vertical force on the speed of movement for the train locomotive of the train Talgo-250 (=105.5 kN): 1 − average value of the force; 2 − calculated value of force Pᵣ.
In the same way, the influence of the number of axles on the value of the vertical force of the wheel on the rail was established, which characterizes the influence of body vibrations and unevenness on the wheels (as an example, 8 axles of passenger cars with approximately the same load were considered) (Fig. 4.a).

The influence of the place where the rail was cut was also analyzed on the value of the vertical force of the wheel on the rail, which characterizes the effect of wheel movement along dynamic track irregularities resulting from rail vibrations (as an example, 8 sections are considered on both threads with different distances in a section with a total length of 42 meters without significant deviations in retention) (Fig. 4.b).

Fig. 4. Dependence of the average and calculated vertical force on the speed of movement for the passenger car of the train $P_{av}$, $P_{cr}$.
Similar calculations were made for the trailer car of the Skoda train at $P_{st}=84$ kN. The dependence of the average and calculated vertical force on the speed of movement for a trailer car is shown in Fig. 5.

Fig. 5. Dependence of the average and calculated vertical force on the speed of movement for the trailer car of the Skoda train: 1 - average value of force; 2 - calculated value of force.

To identify the regularities of the influence of vertical forces on the speed of trains, the movement of a high-speed train was simulated Talgo-250 with different speeds at a constant axle load from the locomotive. The total number of measurements in the calculation matrix fluctuated for different levels of speed, for example, for a speed of 200 km/h it was 560 values. The results obtained obey the normal distribution law. On fig. Figure 6 shows examples of distribution laws obtained from experimental data for speeds of 80 and 200 km/h with a signal sampling step of 12.5 kN.

For a speed of 200 km/h, the degree of impact (according to the Fisher coefficient) was set to the axle number of the car at the level of 1.696; the degree of influence of the rail section number is 122.4 at the level of the critical value of the Fisher coefficient of 2.02. For other values of the speed of movement, the ratio of the given indicators does not fundamentally change. For example, at a speed of 80 km/h, Fisher coefficients of 0.296 and 120.1 were obtained for the axle and rail section, respectively.

Thus, we can conclude that body vibrations in modern passenger cars are damped qualitatively and do not lead to a significant increase in the vertical pressure force of the wheel on the rail. The main factor of perturbation of the dynamic force can be considered the passage of the wheel-dynamic unevenness of the track, which is formed even in the absence of significant geometrical irregularities due to vibrations of the rail on an elastic underbase.
Therefore, it is advisable to consider a method for determining precisely the dynamic force from the passage of the wheel along the rail unevenness. In its primary form, formula (5) can be represented as follows

$$S_{ph} = 0.707 \frac{n^2}{\sqrt{2g}} \frac{U_{qk}}{\sqrt{k}} \sqrt{V_i} \ldots$$

(8)

Here 0.707 is the coefficient of transition from the maximum value of the force to its standard deviation, corresponding to the trigonometric function used to describe harmonic oscillations; g is the acceleration of free action; i - dynamic slope of the rail unevenness; a0 - coefficient, which entered the formula later, is determined by the ratio of the mass of the wheel and the mass of the track, reduced to the point of their contact.

Formula (8) is the result of processing the data from the solution of the differential equation of vibrations of the mechanical pair “wheel-rail”, originally obtained for the under-rail base with wooden sleepers [2, 4, 13]. For reinforced concrete sleepers, another formula was proposed [1, 13]

$$S_{ph} = 9.1 \times 10^{-3} P \sqrt{q_k} \sqrt{\frac{E_I}{U^2}}$$

(9)

Here E - modulus of elasticity of rail steel; I is the moment of inertia of the rail.

However, in modern practical track strength calculations [5], formula (8) is used, obtained from formula (5) with the corresponding coefficients, which take into account, first of all, the change in dynamic inequality parameters when reinforced concrete sleepers are used, caused by an increase in the rigidity of the under-rail base. and changing the reduced mass of the path

$$\begin{align*}
\text{i} &= A \alpha \beta \gamma \sqrt{D} \\
A &= \frac{1.788 \times 10^4 \sqrt{2g}}{n^2 a_0}
\end{align*}$$
Curves of dependences of the standard deviation of the force when passing along the rail inequality from the modulus of elasticity of the under-rail base, calculated by formulas (5) and (8), for the levels of train speeds of 40, 160 and 250 km/h, shown in Fig. 7. Analysis of the results obtained (Fig. 7) shows that at low speeds, the discrepancy in the calculations is insignificant. The moduli of elasticity of the under-rail base, calculated for different speed levels according to formulas (5) and (8), in the range of speeds of 40-50 km/h coincide with sufficient accuracy. A significant increase in the modulus of elasticity leads to a difference in results, which becomes especially noticeable at high speeds.

Fig. 7. Dependence of the standard deviation of the force on the passage of the wheel along the rail unevenness relative to the modulus of elasticity of the under-rail base: 1 – according to formula (5); 2 – according to formula (8) for speeds of 40, 160 and 250 km/h.

In this study, it is not intended to solve the problem of determining the reduced mass of the path. On the contrary, the above considerations show that such a parameter is artificial and can only be used for low speeds at the level of static calculation schemes [7].

4 Conclusion

Analysis of the results of statistical processing of data from computational studies of the movement of modern passenger trains with speeds up to 200 km/h relative to the level of the dynamic vertical force of the wheel on the rail made it possible to establish the degree of influence of the above factors on the formation of the value of this force. The presence of discrepancies in the obtained calculated values with the theoretical ones indicate the need to revise the methodology for determining the design force for practical calculations of the track for strength in the conditions of high-speed passenger trains. The revision should consist not only in the adjustment of existing methods, but also in the formation of new approaches based on modern mathematical models or relevant statistical data, taking into account foreign experience [8].

This approach hides the influence of the detailed characteristics of the track and the rolling stock on the formation of the dynamic force, but allows obtaining adequate numerical values for the corresponding calculations of the strength of the railway track.

The presented results provide further development of the scientific task of creating modern methods for calculating the stress-strain state of a railway track for high speeds. On
The basis of theoretical studies and calculated measurements, the factors of excitation of the dynamic component of the vertical force for modern passenger trains are established. The recommendations received will improve the existing track strength calculations, solving the practical problem of their application for speeds of more than 160 km/h.

The value of the calculated vertical force acting from the wheel on the rail depends on many factors. In practical calculations of the path for strength, this indicator is formed from several components, the influence of which is not the same at different speeds. Based on the analysis of the results of analytical calculations, it was found that the following speed zones can be separated by the influence of different dynamic components: up to 80-100 km/h, 120-250 km/h, more than 250 km/h.

Both theoretical studies and analysis of statistical data from experimental measurements show that the main factor in excitation of the dynamic component of the vertical force for modern passenger trains moving at a speed of 120 km/h or more is oscillations in the "wheel-rail" system or the so-called wheel passage dynamic rail unevenness.

Existing calculation methods of dynamic forces need to operate with an indicator, like the reduced mass of the path. The above studies show that this parameter is artificial and can only be used for low speeds at the level of static calculation schemes.

References

1. M.F. Verigo, The main provisions of the methodology for calculating forces acting on reinforced concrete sleepers (Tr. Vniizhta, Moscow, 1963)
2. E.I. Danilenko, Rules for calculating the railway track for strength and stability: CP-0117
3. I. Danilenko, V.V. Rybkin (Transport of Ukraine, Kiev, 2004)
10. TST-52/14. Methodology for assessing the impact of rolling stock on the track according to the conditions for ensuring its reliability (Ministry of Transport of the Russian Federation, Moscow, 2000)
14. B. Likhtberger, Zheleznodorozhnyy put [Railway track] E3S Web of Conferences 402, 06009 (2023) https://doi.org/10.1051/e3sconf/202340206009

TransSiberia 2023


20. Z. Kakharov, Z. Yavkacheva, E3S Web of Conferences 371, 02042(2023) DOI: https://doi.org/10.1051/e3sconf/202337102042
