Assessment of the impact on the railway track of a special purpose passenger car model 61-934

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Abstract. In this work, theoretical studies have been carried out in order to assess the strength of structural elements of the track superstructure from the impact of a special-purpose passenger carriage. The assessment of the impact of the wagon on the railway track was carried out according to the stresses in the edges of the rail base, stresses in the main area of the subgrade and the stability of the track against lateral shear. The calculations showed that the strength and stability of the structural elements of the track superstructure as a result of the impact on it of a special-purpose passenger carriage model 61-934 meets the regulatory requirements.

1 Introduction

The main priority tasks for the further development of railway transport in the Republic of Uzbekistan, the implementation of which will create the necessary prerequisites for the development of the constantly growing volumes of freight and passenger traffic in the absence of reserves for increasing the carrying and path capacity of cargo-intensive sections of the railway network, are the introduction of modern rolling stock [1-4].

The possibility of introducing new rolling stock into operation is determined by assessing the stress-strain state of the structural elements of the permanent way and the subgrade, which are indicators of the impact of rolling stock on the railway track [5-9].

2 Methods

This calculation was carried out to assess the strength of the structural elements of the track superstructure under the impact on it of a new special-purpose passenger carriage model 61-934, manufactured at JSC "Tashkent plant for the construction and repair of passenger cars" [10-13].

Wherein, the fulfillment of the following regulatory requirements was checked [14, 15]:
– to the edges of the rail foot;
– to the strains in the main site of the roadbed;

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- the ratio of the maximum horizontal load acting on the sleeper to the average vertical load on the sleeper.

The calculation was carried out following [14, 16].

Following the requirements, a track with the following design was adopted as the design railway track: rail type – R65; reduced rail wear – 6 mm; the number of sleepers per 1 km of track – 1840; type of sleepers - reinforced concrete; a kind of ballast - crushed stone.

The initial data for the calculation are given in Table 1.

Table 1. Initial data for the calculation

Parameter name	Indication	Numeric value
Statical load of a wheel on a rail, kgf	P_{CT}	7450
Wheelbase, m	L_T	2.5
Weight of unsuspended parts of the car, referred to the wheel, kgf	q	705
Carriage speed, km/h: in a straight line in a curve	V	160 80
Static deflection of spring suspension, mm	f_{cm}	0.202
Wheel diameter, cm	d	95
Coefficient of friction between wheel and rail	μ_{TP}	0.15

Calculated track characteristics with typical rail spacers are shown in Table 2.

Table 2. Calculated track characteristics

Parameter name	Indication	Numeric value
Coefficient of relative stiffness of the rail base and the rail, cm $^{-1}$	k	0.01536
Coefficient of relative stiffness of the rail base and rail in the broadside direction, cm $^{-1}$	k_Z	0.01462
Resisting moment of a worn rail along the rail foot, cm^3	W	417
Coefficient taking into account the influence of the rail and sleepers type, the type of ballast, track mass	L	0.261
Distance between the axles of sleepers, cm	L_S	55
Elasticity modulus of the rail base, kgf / cm^2	U	1500
Coefficient characterizing the ratio of unsprung wheel weight and track weight	α_0	0.403
Coefficient taking into account the irregularity in the distribution of pressure along the sleeper	ω	0.7
The width of the lower bed of the sleeper, cm	b	27.6
Ballast layer thickness under the sleeper, cm	h	55
Half sleeper area, taking into account bending correction, cm^2	Ω_a	3092
Friction coefficient of sleepers against ballast	F_S	0.45
Transition coefficient from axial to edge stresses: in a straight line in a curve	f	1.35 1.65

The assessment of the impact of the car on the track was carried out in accordance with [15] according to the following parameters: stresses in the edges of the rail foot σк; ballast stresses at depth h (σh); the stability of the path against lateral shear, determined by the value of the coefficient α.

In this case, the fulfillment of the following conditions was checked:

$$\sigma_{\kappa} \leq 240 \text{ MPa}; \sigma_h \leq 0.08 \text{ MPa}; \alpha \leq 1.4.$$

Calculation of stresses was carried out according to [14], calculation of track stability - according to [16].

The elasticity modulus of rail steel E in the calculations was taken to be $2{,}1\times10^5$ MPa.

The calculation was carried out for an equipped carriage taking into account the effective load. Simultaneously, its effect on the track from the side of the most loaded bogie was considered.

The calculation of the edge stresses was carried out for two calculation cases:

– for a speed of 160 km / h (movement in a straight line, $f = 1{,}35$);

– for a speed of 80 km / h (movement in a curve, $f = 1{,}65$).

Here, a speed of 80 km / h corresponds to movement in a curve with a radius of 300 m with a canting of the outer rail of 150 mm and an unbalanced acceleration of 0.7 m / s^2.

The values of the coefficients f are taken according to [14]; and besides, as in the specified source, the movement with speeds up to 140 km / h is considered, for the case of direct movement, the coefficient is increased by the value of the ratio (160/140).

3 Results and Discussion

The stresses in the edges of the rail foot were determined by the formula

$$\sigma_K = f \cdot \sigma_0 \tag{1}$$

where f is the coefficient of transition from axial stresses to edge stresses; σ_0 is maximum stresses in the rail foot from its bending, determined by the formula

$$\sigma_0 = \frac{P^I_{EKV}}{4kW} \tag{2}$$

where k is the coefficient of the relative stiffness of the rail foot and the rail; W is the resisting moment of the rail relative to its foot; P^I_{equ} is maximum equivalent load for calculating the stresses in rails, determined by the formula

$$P^I_{Ekv} = P^{max}_{din} + \sum m_i \text{Ч} P_{cpi} \tag{3}$$

where μ_i are the ordinates of the line of influence of the rail bending moments in the track sections located under wheel loads from the vehicle axles adjacent to the design axis.

The value of the ordinate μ_i is determined by the formula

$$\mu_i = e^{-kl_i}\left(\cos\ kl_i - \sin\ kl_i\right), \tag{4}$$

where l_i is the distance between the center of the axle of the design wheel and the wheel i is the axle, adjacent to the design one, is taken following [14]; e is base of the natural logarithms; $P^{\max}_{дин}$ is dynamic maximum load from the wheel to the rail, determined by the formula

$$P^{\max}_{din} = P_{CP} + \lambda S \tag{5}$$

where P_{AV} is the average value of the vertical load of the wheel on the rail, determined by the formula

$$P_{CP} = P_{CT} + P_P^{CP} \qquad (6)$$

where P_{ST} is the static load of the wheel on the rail, кг; P_P^{CP} is the average value of the dynamic load of the wheel on the rail from the vertical vibrations of the sprung parts of the car, kg, determined by the formula

$$P_P^{CP} = 0.75 \times P_P^{\max} \qquad (7)$$

where $P_P^{\max}$ is the maximum dynamic load of the wheel on the rail from vertical vibrations of the sprung parts of the car, determined by the formula

$$P_P^{\max} = k_D (P_{CT} - q) \qquad (8)$$

where C_D is the coefficient of vertical dynamics, determined by the formula:

$$k_D = 0.1 + 0.2\frac{V}{f_{cm}} \qquad (9)$$

where V is travel speed, km / h; f_{sm} is the static deflection of the spring suspension; q is the weight of the unsprung parts referred to the wheel; λ is normalizing factor that determines the maximum dynamic load occurrence probability, following [14] λ = 2,5; S is the standard deviation of the dynamic load from the wheel on the rail, determined by the formula

$$S = \sqrt{S_P^2 + S_{IP}^2 + 0.95 S_{NNK}^2 + 0.05 S_{INK}^2} \qquad (10)$$

where S_P is the standard deviation of the dynamic load from vertical vibrations of the sprung parts of the car, determined by the formula

$$S_P = 0.08 \times P_P^{\max} \qquad (11)$$

where $P_P^{\max}$ is maximum dynamic load of the wheel on the rail from vertical vibrations of the sprung parts of the car; $S_{НП}$ is the standard deviation of the dynamic load from the vertical vibrations of the unsprung parts of the car, determined by the formula

$$S_{IP} = 0.565 \times 10^{-8} \times L \times l_{Ш} \times \sqrt{\frac{U}{k}} \times \sqrt{q} \times P_{CP} \times V \qquad (12)$$

where the values of L, l_{sh}, U, k, q see Table 2; V - see Table 1; S_{NNW} is the standard deviation of the dynamic load from the inertia forces of the unsprung parts of the car when the wheel moves with a smooth continuous unevenness on the rolling surface, determined by the formula

$$S_{NNK}=\frac{0.052\times\alpha_0\times U\times V^2\times\sqrt{q}}{d^2\times\sqrt{k\times U-3.26\times k^2\times q}} \tag{13}$$

where the values of α_0, U, k, q, d see Table 2; V – see Table 1; S_{INI} is the standard deviation of the dynamic load from the inertial forces of the unsprung parts of the car, arising from the presence of isolated irregularities on the rolling surface, determined by the formula

$$S_{INK}=0.735\times\alpha_0\times\frac{U}{k}\times e \tag{14}$$

where the values of α_0, U, k see Table 2; e is coefficient taking into account the depth of a smooth isolated unevenness (following [14], it is taken equal to 0.067).

The results of calculating the stresses in the edges of the rail foot are shown in Table 3.

Based on the data given in Table 3, the maximum stresses in the rail feet are 62.3 MPa, which is less than the permissible stresses of 240 MPa.

The calculating formula for determining the normal stresses σ_h in the ballast (including the top of the subgrade) at a depth h from the foot of the sleeper along the design vertical has the form

$$\sigma_h=\sigma_{h1}+\sigma_{h2}+\sigma_{h3} \tag{15}$$

where σ_{h1} and σ_{h3} are stresses on the subgrade under the design sleeper (Sleeper № 2) under the influence of sleepers № 1 and № 3, adjacent to the design one, determined by the formula

$$\sigma_{h1}=\sigma_{h3}=0.25\sigma_B A \tag{16}$$

where A is design coefficient, determined according [14], where its dependence on h, b and l_{sh} is shown in tabular form (for the accepted design of the path A = 0.255); σ_B is average values of stresses along the foot of sleepers, determined by the formula

$$\sigma_B=\frac{k\times l_{sh}}{2\Omega_a}P^{II}_{EKV} \tag{17}$$

where P^{II}_{EQU} is the maximum equivalent load for calculating the stresses in the sub-rail base, determined by the formula

–calculating the average stresses under the design sleeper (under sleeper No. 2)

$$P^{II}_{EKV}=P^{\max}_{DIN}+P_{CP}\cdot\eta \tag{18}$$

–calculating the average stress under sleepers adjacent to the calculated one (under sleepers № 1 and № 3)

$$P_{EKV}^{II} = P_{DIN}^{\max} \cdot \eta_{sh} + P_{CP} \cdot \eta \tag{19}$$

where η is the ordinate of the influence of neighboring wheels, determined by [14], where the dependence of η on the product of kx is given in a tabular form. In this case, $x = L_T$ – when calculated by the formula (18); $x = L_T - l_S$ – for sleeper №1 when calculated by the formula (19); $x = L_T + l_S$ – for sleeper №3 when calculated by the formula (19); η_{sh} is the ordinate of the influence of neighboring sleepers, determined from the dependence of η on the product of kx, given in [14] for $x = l_{sh}$; σ_{h2} are stresses on the subgrade under the impact of sleepers № 2 (design sleepers), determined by the formula

$$\sigma_{h2} = \sigma_B \omega \left[2,55C_2 + (0.635C_1 - 1.275C_2)m\right] \tag{20}$$

where C_1 and C_2 are calculated coefficients, following [14] for the adopted track design $C_1 = 0.241$, $C_2 = 0.116$); m is coefficient of transition from the pressure on the ballast averaged over the width of the sleeper to the pressure under the axis of the sleeper, determined by the formula

$$m = \frac{8.9}{\sigma_B + 4.35} \tag{21}$$

Table 3. Calculation of stresses in the edges of the rail foot

Parameter name	Formula number	Unit of measure	Numeric value	
			in a straight line	in a curve
S_{INK}	14	kgf	1938.1	1938.1
S_{NNK}	13	kgf	496.7	124.2
K_D	9	-	0.26	0.18
$P_P^{\max}$	8	kgf	1743.0	1208.8
P_P^{CP}	7	kgf	1307.3	906.6
P_{CP}	6	kgf	8757.3	8356.6
S_{IP}	12	kgf	943.9	450.4
S_P	11	kgf	139.4	96.7
S	10	kgf	1149.1	643.3
$P_{DIN}^{\max}$	5	kgf	11629.9	9964.9
kx	-	-	3.69	3.69
μ_i	4	-	-0.0084	-0.0084
P^I_{EKV}	3	kgf	11555.9	9894.3
σ_0	2	kgf /cm^2	432.4	370.2
σ_k	1	MPa (kgf /cm^2)	59.5 (583.7)	62.3 (610.8)

The calculation results are shown in Table 4.

Table 4. Calculation of stresses on the top of the subgrade

Parameter name	Formula number	Unit of measure	Numeric value
η: - sleepers №1 - sleepers №2 - sleepers №3	 — — —	 — — —	 - 0.0422 -0.0218 - 0.0047
η_s	—	—	0.6065
P^{II}_{EQU}: - under the sleeper №1 - under the sleeper №2 - under the sleeper №3	 19 18 19	kgf	 5691.3 9782.6 14400.6
m	21	—	1.42
σ_{h1}	16	kgf /cm^2	0.05
σ_{h2}	20	kgf /cm^2	0.28
σ_{h3}	16	kgf /cm^2	0.13
$\sigma_{3\Pi}$	15	MPa (kgf /cm^2)	0.047 (0.46)

As follows from the data given in Table 4, the maximum stresses on the top of the subgrade are 0.047 MPa, which is less than the permissible stress of 0.08 MPa.

The coefficient α (the ratio of the maximum horizontal load to the average vertical load) is determined by the formula

$$\alpha = \frac{H_{ShP}}{P_{ShP}} = \frac{Y}{P_{CP}} \times \frac{k_Z}{k} \tag{22}$$

where Y is the shear force acting from the wheel on the rail.
The Y / P_{CP} ratio is determined by the expression

$$\frac{Y}{P_{CP}} = \frac{2C_0}{P_{CP} \times l_{sh} \times k_Z} + 2f_{sh}\frac{k}{k_Z} + \mu_{TP}\,, \tag{23}$$

where C_0 is the force of the initial resistance to the pumping of the sleepers in ballast, which is determined by the formula

$$C_0 = 0.1 \times \frac{k \times l_{sh}}{2} P_{CP}\,. \tag{24}$$

The value of P_{CP} is given in Table 3 for the case of the movement in a straight line.

The calculation results are shown in Table 5.

Table 5. Calculation of stresses on the top of the subgrade

Parameter name	Formula number	Unit of measure	Numeric value
C_0	24	kgf	38.29
Y / P_{AV}	23	-	1.30
α	22	-	1.24

As follows from the data given in Table 5, the value of the coefficient α (the ratio of the maximum horizontal load to the average vertical load) is 1.24, which is less than the maximum allowable value for crushed stone ballast equal to 1.4.

4 Conclusions

The calculations showed that the strength and stability of the structural elements of the track superstructure as a result of the impact on it of a special-purpose passenger carriage model 61-934 meets the regulatory requirements. In this case, the following results were obtained:

– the stresses in the edges of the rail foot are 62.9 MPa (permissible stresses are 240 MPa);

– the stresses on the top of the subgrade are 0.047 MPa (permissible stresses are 0.08 MPa;

– the ratio of the maximum horizontal load acting on the sleeper to the average vertical load on it is 1.24 (the maximum allowable value is 1.4).

Also, to increase the accuracy of measurements of the force effect of rolling stock on a railway track, it is proposed to apply methods of piecewise continuous recording of vertical and lateral forces from the interaction of a wheel with a rail by measuring stresses in two sections of the rail in tests on the impact on the track [17-20].

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