INTERACTION OF THE TE33A DIESEL LOCOMOTIVE AND THE RAILWAY TRACK ON CURVED SECTION WITH RADIUS 290 m

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Resume

The purpose of the research was an experimental study of the dynamic processes of interaction between a TE33A diesel locomotive and a railway track when passing along a curved section with a radius of 290 m. To study the dynamic and driving properties of the rolling stock, the analysis of the experimental data obtained was carried out. When performing the study, the following results were obtained: the indicators of the frame forces of each wheel were determined when moving the TE33A locomotive along a curve with a radius of 290 m; the coefficients of the vertical dynamics of the second suspension stage when moving the TE33A locomotive along a curve with a radius of 290 m are determined; axial stresses in the rails under the influence of the TE33A locomotive on the track with rails of grades R50 and R65 are determined.

Article info

Received 11 March 2023
Accepted 16 August 2023
Online 12 September 2023

Keywords:

railway track
curved section
rolling stock
locomotive wheelset
vertical dynamics
axial stresses in rails

Available online: https://doi.org/10.26552/com.C.2023.069

ISSN 1335-4205 (print version)
ISSN 2585-7878 (online version)

1 Introduction

One of the decisive factors ensuring clear and rhythmic operation of the railways is the stable operation of the locomotive economy, improving the technical condition, maintenance, and use of the locomotive fleet. The high intensity of locomotive use, high weight and speed of trains require an increase in the traction and speed qualities of locomotives operating under conditions of large dynamic loads. To ensure the safety of trains, the crew of the locomotives must have sufficient reliability.

Subsequently, the need to increase the speed and power of diesel locomotives, as well as improve the conditions for fitting them into the curves, led to the appearance of such a structural element of the mechanical part as carts, on which the body frame rested and was able to turn relative to it [1].

The use of bogies in the design of the mechanical part of locomotives required a number of related technical tasks, including: resting the body on a bogie movable relative to the vertical axis, transmitting longitudinal and transverse forces in conditions of relative movements of the body and bogie and connecting bogies with each other to improve the conditions of movement in curves. To reduce the likelihood of wheel skidding, devices were provided to help equalize the loads between individual driving wheelsets and prevent a significant decrease in vertical loads on the front axles of the wheelsets of each bogie when the maximum traction force is realized when the locomotive is starting.

The design of the trolley, as a separate device, forced, in order to improve the vibration protection of the body, to introduce elastic vertical links (body stage of spring suspension), and to improve the indicators of dynamic qualities during the horizontal vibrations (smoothness and lateral force), to abandon the rigid link of the body with the trolley in the transverse direction, introducing quasi-elastic crosslink devices - the so-called «return» devices. At the same time, it turned out that the bogie frame began to perform loading functions

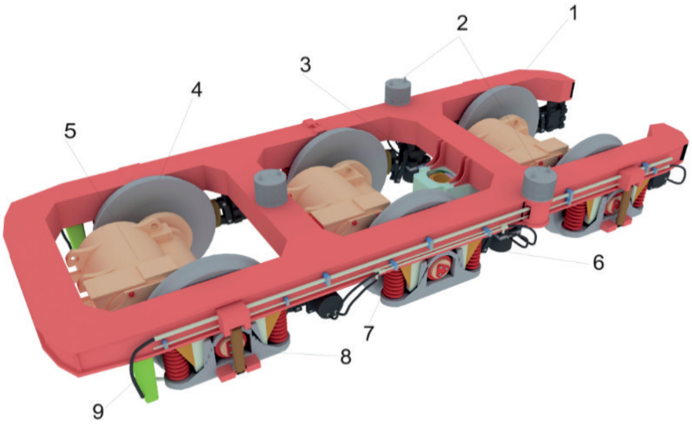

Figure 1 *General view of the three-axle trolley: 1 - trolley frame, 2 - elements of the second stage of spring suspension, 3 - the place of laying the pin, 4 - wheelset, 5 - traction electric motor, 6 - brake pad, 7 - elements of the first stage of spring suspension, 8 - axle box assembly, 9 - sand supply*

Table 1 *Specifications of the trolley*

Name of indicators	Parameter values
Load from wheelset on rail, kN	233
Design speed, km/h	120
Traction motor	5GEB30B
Number of traction motors	3
Traction drive	One-sided with support-axial suspension of TED (TM)
Gearing	One-way, spur, with UZK
Gear gear ratio	5.3125
Design pressing of brake pads on axle at pressure of 0.38 MPa, kN	136
Body support system	Three-point, supports with rubber-metal elements

unusual for its purpose, which led to a complication of the design of the bogies, an increase in their weight and, as a result, to a deterioration in the conditions for the locomotive to pass curved sections of the track [2-3]. All these ultimately led to an increase in the horizontal forces of interaction between wheels and rails, and to an increase in the wear of wheelset bands and rails.

In this work, the test object is the TE33A main freight diesel locomotive with asynchronous traction motors and electric drive. The locomotive was manufactured by Lokomotiv Kurastyru Zauyty Joint Stock Company in Astana in 2010. The axial formula of the locomotive is 30-30. Design axle load on rails (227.65 ± 2%) kN. The service weight of the diesel locomotive (138 ± 2%) tf. Design speed 120 km/h. The diesel locomotive is designed to work with freight trains and has a two-stage spring suspension (Figure 1). Static deflection of the first stage of suspension is 131.5 mm. The second suspension stage consists of one central and two side support and return supports (on the trolley). Static deflection of the central support is 13.7 mm, side supports 10.3 mm. The first stage is helical springs, the second stage is rubber-metal blocks [4]. Traction and braking forces from the bogies to the body are transmitted by a pivot. The diesel locomotive uses support-axial suspension of traction motors and motor-axial rolling bearings. The bogie frame is connected with wheelsets through jaw boxes. The main technical data are given in Table 1.

Wheelsets are formed from solid-rolled wheels pressed onto the forged axle. Each wheel pair corresponds to an individual brake unit with a cylinder and shoes, the operating parameters of the unit are automatically adjusted. Such a system is the most efficient and easy to operate and repair [5-6].

The wheelsets of the locomotive sustain and transfer their own weight, weight of the body and bogies of the locomotive to the rails [7]. When rotating, the wheel pair, interacting with the rail, realizes the clutch force of the locomotive. Each wheel pair is driven by individual traction motor 5GEB30.

Minimum dimensions for the wheel profile are given in accordance with Figures 2, 3 and 4.

The main reasons for the wear and undercutting of the ridges of the wheel pairs of locomotives are the passage of curved sections of the railway track [4]. Determination of permissible speeds and dynamic parameters of interaction between a TE33A series diesel locomotive and a railway track, when passing on a curved section with the smallest radius, is an urgent research issue.

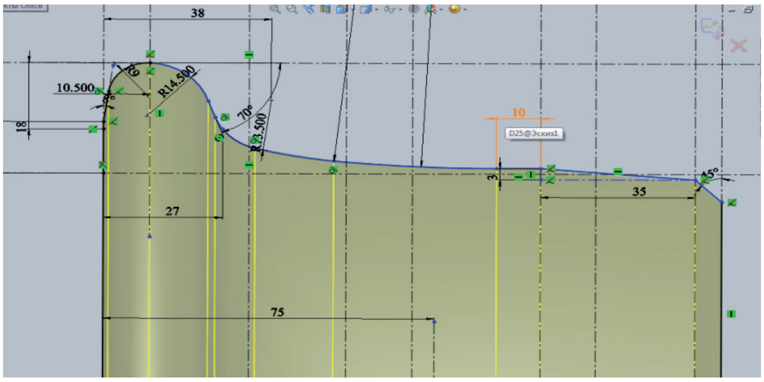

***Figure 2** The profile of the locomotive wheel TE33A with a crown thickness of 27 mm*

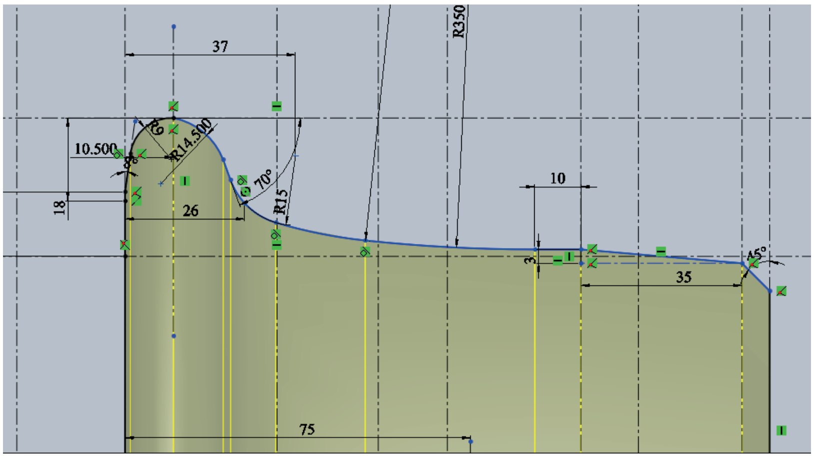

***Figure 3** The profile of the locomotive wheel TE33A with a ridge thickness of 26 mm*

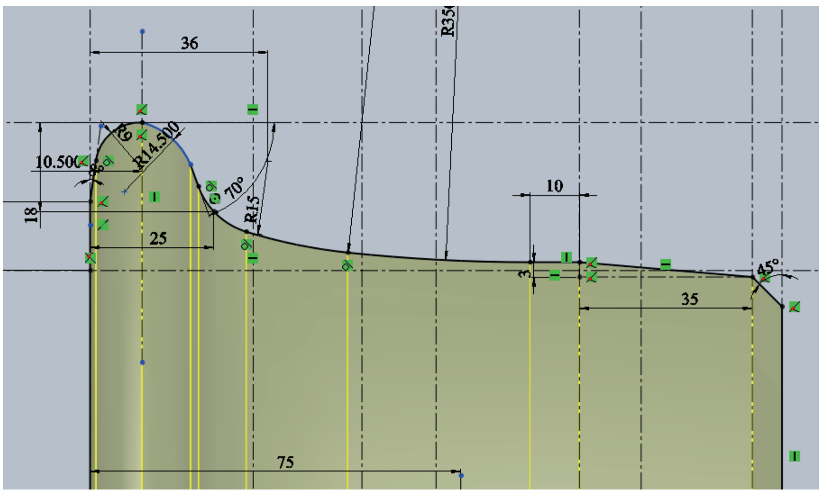

***Figure 4** The profile of the locomotive wheel TE33A with a ridge thickness of 25 mm*

To solve this issue, the following tasks have been set:

1) determination of the vertical dynamics coefficient of the first and second suspension stages;
2) identification of the coefficient of stability against wheel derailment from the railway track;
3) experimental evaluation of dynamic stress in the edges of the sole of the rail in curved sections with a radius of 290 m;
4) determination of the parameters of the frame (lateral) forces in straight and curved sections of the railway track.

2 Materials and methods

To measure the level of impact of the diesel locomotive on the track in the curved sections of the

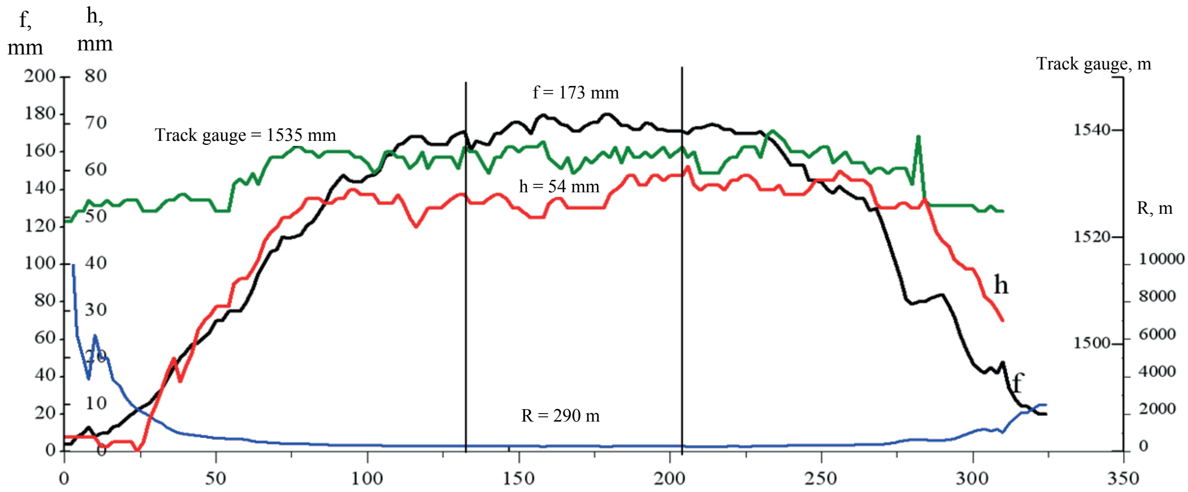

Figure 5 *Parameters of the curved section of the path of a radius of 290 m*

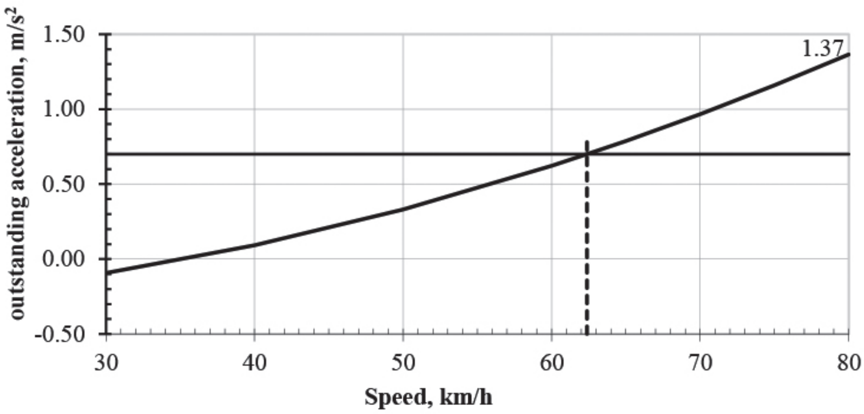

Figure 6 *Outstanding acceleration in 290 m radius curve*

track with a radius of 300-400 m, the track 3820-3821 km of the Chokpar- Alaaigyr section was selected. At the same site, the dynamic indicators of the locomotive were determined. In the section, the track is laid with rails R65 on wooden sleepers. Length of links 25 m. Ballast crushed stone [8-9]. Epura sleepers 2000 pcs/km. According to the certificate provided by the PCh-47 (track facilities), the average lateral wear of the rails is 8 mm. The number of unusable sleepers is 15 %. According to the design documents, the curve radius is 285 m. Immediately before the tests on the site, 6 unusable sleepers were replaced. According to the track measurement car, the state of the track corresponds to the “good” rating. Results of measurements of geometric dimensions of the track section are given in Figure 5.

Thus, the actual elevation of the outer rail is 54 mm, and the radius is 290 m. For this section [10-11], the dependence of the outstanding acceleration on the speed of movement is shown in Figure 6.

When testing in a 290 m radius curve, the ambient temperature was from 4 to 10 °C. The wind speed reached 14 m/s.

The races were performed at speeds of 30, 40, 50, 60 and 70 km/h.

The data for determining the estimated values of the frame forces, the vertical dynamics coefficients of the first and second suspension stages were processed using the same method as for the curve of radius 600 m [12].

The processing results are shown in Figures 7-9 and Tables 2-4.

The data shown in Figures 7-9 show that the dynamic parameters of the locomotive in the curve of radius 290 m are within the permissible limits [13-14].

Results of calculation of the safety factor against the wheel derailment from data measured in curve of radius 290 m are given in Figure 10. The minimum value of the coefficient is 1.7 at a speed of 70 km/h.

When drawing up a table of permissible speeds of movement of a diesel locomotive, an analytical expression is used in the calculations, describing the dependence of the coefficient of vertical dynamics of the first stage of suspension on the speed of movement. To form this expression, the vertical dynamics coefficients at forward travel were combined with the data at reverse travel [15]. The results of measurements in straight and curved sections of the path were included in the formed array. From the obtained variation series, the maximum probable values of the coefficient k_d were calculated for

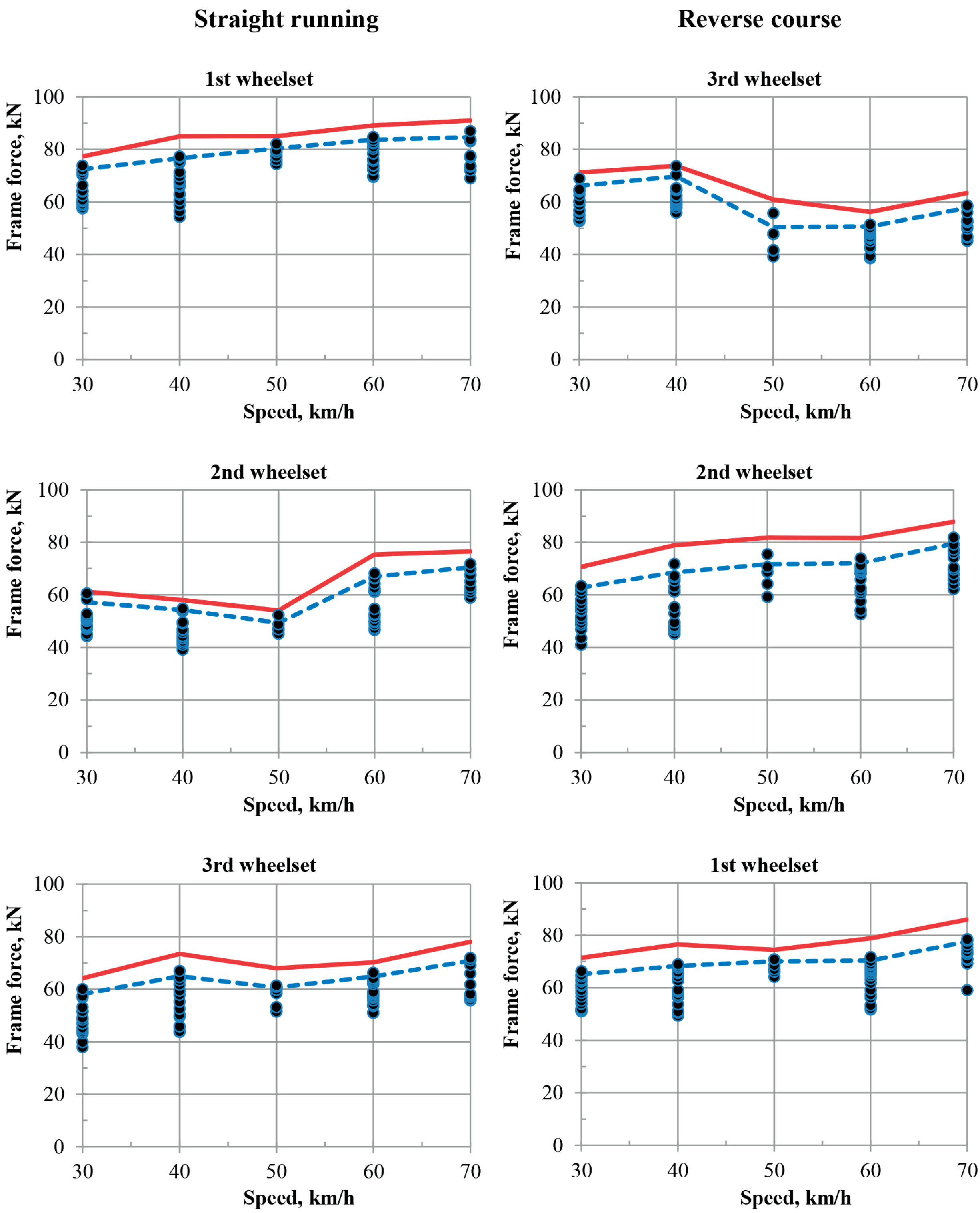

Figure 7 *Frame forces during the diesel locomotive movement TE33A in a 290 m radius curve*

each realized speed of movement. Based on least-squares data processing, this relationship can be described by the empirical formula:

$$k_d = 0.06667 + 0.00211V, \qquad (1)$$

where V - speed, km/h.

Before the tests and after the completion of the tests, the diesel locomotive was inspected. As a result of visual inspection of the crew of the locomotive, it was established that before the tests and after them there was no touch of the nodes and elements in places not provided for by the design [16].

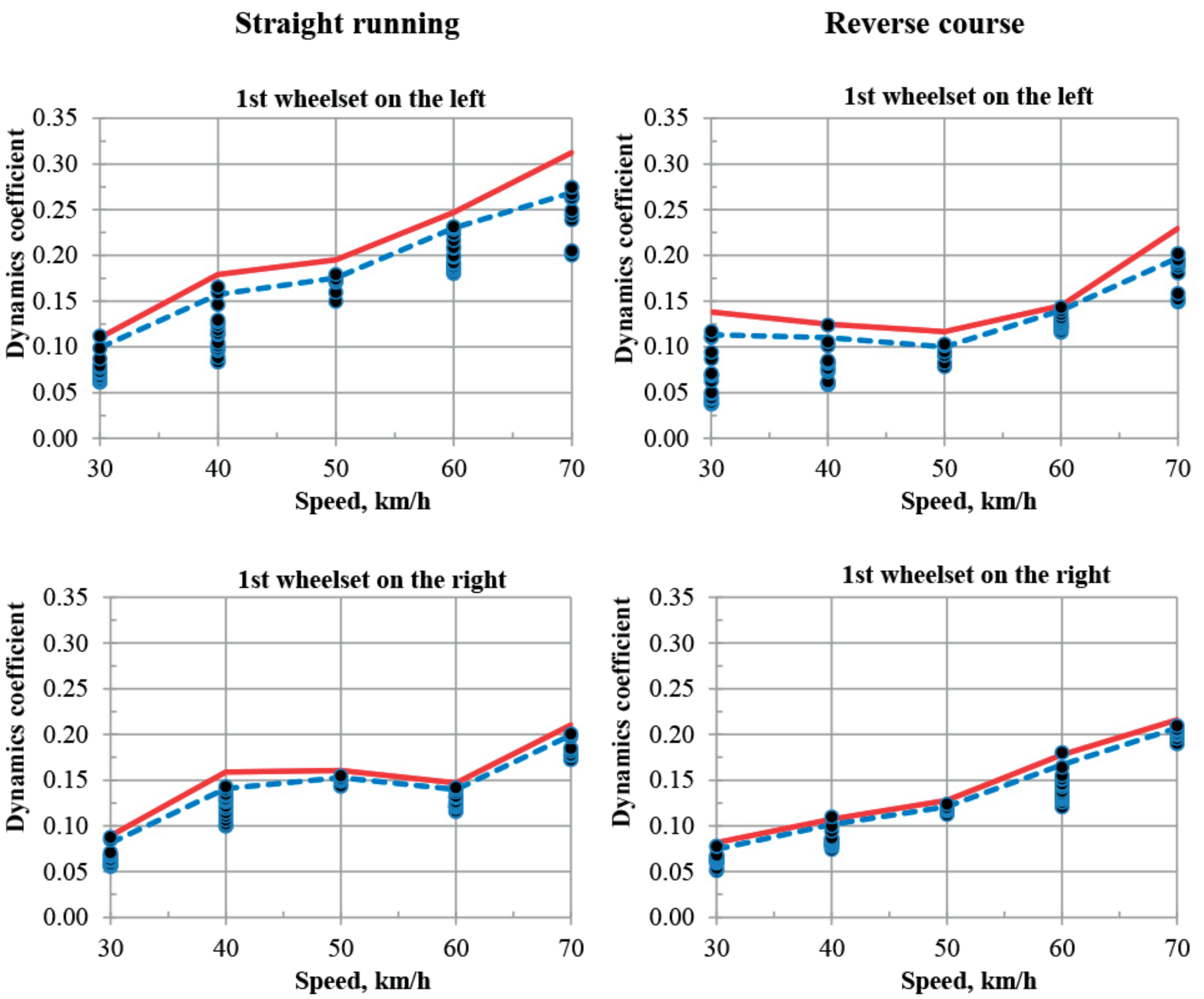

Figure 8 *Coefficient of vertical dynamics of the first stage of suspension at movement of a diesel locomotive TE33A-0023 in curve of radius 290 m*

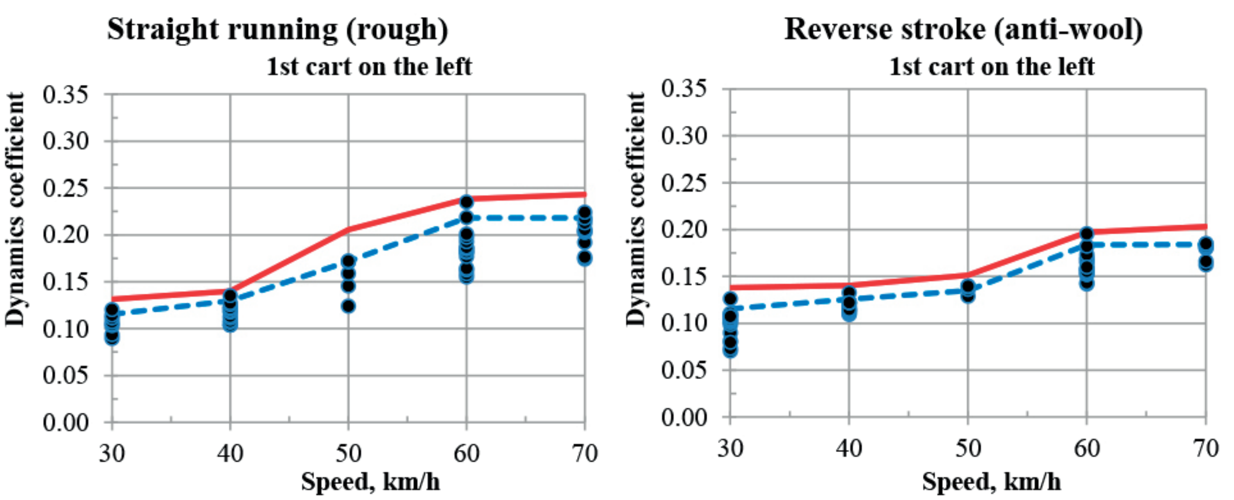

Figure 9 *Coefficient of vertical dynamics of the second stage of suspension at movement of a diesel locomotive TE33A-0023 along curve of radius 290 m*

__Table 2__ Frame forces during the movement of a diesel locomotive TE33A in a curve of radius 290 m

Speed, km/h	Maximum probable value						Maximum observed value					
	Straight running			Reverse course			Straight running			Reverse course		
	Wheelset number											
	1	2	3	1	2	3	1	2	3	1	2	3
30	77.3	70.7	64.2	71.4	61.2	71.2	72.4	62.8	58.1	65.2	57.3	66.1
40	85.0	78.9	73.4	76.5	58.0	73.8	76.7	68.6	64.9	68.3	54.3	69.8
50	85.1	81.8	68.0	74.4	54.1	60.9	80.3	71.6	60.7	70.0	49.4	50.5
60	89.2	81.6	70.2	78.9	75.4	56.3	83.7	72.1	64.8	70.4	67.0	50.6
70	91.0	87.8	78.1	86.0	76.5	63.4	84.6	79.4	70.8	77.6	70.5	57.8

__Table 3__ Vertical dynamics coefficient of the first stage suspension during movement of the diesel locomotive TE33A in the curve of radius 290 m

Meaning	Speed, km/h	Straight running		Reverse course	
		1st wheelset		1st wheelset	
		on the left	on the right	on the left	on the right
maximum probable	30	0.11	0.09	0.14	0.08
	40	0.18	0.16	0.12	0.11
	50	0.20	0.16	0.12	0.13
	60	0.25	0.15	0.14	0.18
	70	0.31	0.21	0.23	0.22
maximum observed	30	0.10	0.08	0.11	0.07
	40	0.16	0.14	0.11	0.10
	50	0.18	0.15	0.10	0.12
	60	0.23	0.14	0.14	0.17
	70	0.27	0.20	0.20	0.21

__Table 4__ Coefficient of vertical dynamics of the second suspension stage during the movement of the diesel locomotive TE33A in the curve of radius 290 m

Speed, km/h	Maximum probable value		Maximum observed value	
	Straight running on the left	Reverse course on the right	Straight running on the left	Reverse course on the right
30	0.13	0.14	0.12	0.12
40	0.14	0.14	0.13	0.13
50	0.21	0.15	0.17	0.14
60	0.24	0.20	0.22	0.18
70	0.24	0.20	0.22	0.18

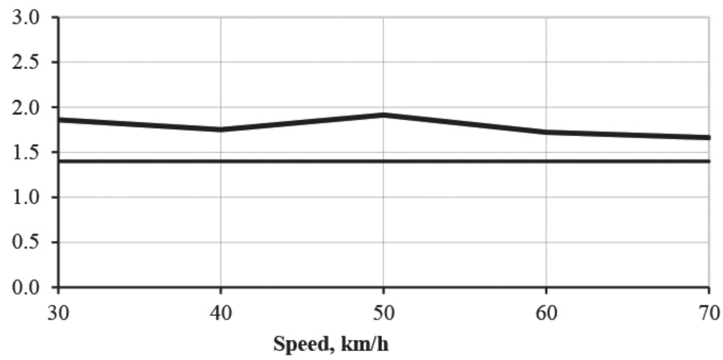

__Figure 10__ Stability margin factor against the wheel derailment in 290 m radius curve

3 Results

Allowable speeds of the diesel locomotive circulation according to the conditions of bending strength of rails are determined based on the theoretical calculations of the path for strength using the experimental data obtained during the present tests: loads from the wheel on the rail based on the results of weighing the diesel locomotive, the coefficient of vertical dynamics, characteristics of the diesel locomotive, track parameters, stress levels in the edges of the rail foot in the test areas. Calculations were performed according to the method presented in [6, 17]. When drawing up the table of permissible velocities, the requirements set forth in [18] are taken into account. The calculations used the values of the dynamic maximum vertical load from the wheel to the rail from the vertical vibrations of the overspring structure, calculated using the formula:

$$P_p^{\max} = k_d(P_w - q), \tag{2}$$

where k_d - coefficient of vertical dynamics, taken from experimental data,
P_w - load from the wheel on the rail according to the results of weighing, kN,
q - weight of unsprang parts per wheel, kN.

According to the obtained data, the maximum probable value of the maximum dynamic load from the wheel on the rail is calculated. In this case, it is assumed that the distribution of values of the maximum dynamic load from the wheel to the rail is subject to the normal law. The probability of non-exceeding the maximum dynamic load from the wheel to the rail of the obtained values is taken 0.994. Under these conditions, the maximum dynamic load from the wheel to the rail is calculated using the formula [16]:

$$P_{din}^{\max} = P_{av} + 2.5S\text{, kN}, \tag{3}$$

where P_{av} - average value of vertical load from the wheel to the rail, kN,
S - mean square deviation of dynamic vertical load from the wheel to the rail, kN.

Value S is defined as composition of average square deviations of dynamic load from wheel to the rail from oscillations of overspring structure, from inertia forces of unsprang masses when wheel passes isolated path irregularity, from inertia forces of unsprang masses arising due to continuous irregularities on surface of wheels rolling and due to presence of smooth isolated irregularities on surface of wheels [18].

The action of horizontal transverse forces and torques, created by eccentric application of vertical forces on the rail, is taken into account by introducing a transition factor from axial stresses at the bottom of the rails to edge *f*. To determine the coefficient, the maximum probable stress values in the edges of the bottom of the rails, calculated from measurements in a straight line and a curve with a radius of 600 m at a speed of 100 km/h and in a curve with a radius of 290 m at a speed of 60 km/h, are used [19].

Maximum probable values of experimental axial stresses in the rail foot are calculated as maximum probable values of the stress semi-summes in the edges of the rail foot.

The maximum equivalent load for calculating stresses in rails from bending and torsion, replacing a system of concentrated wheel loads, is formed taking into account vertical loads from both wheelsets of the bogie:

$$P_{eq}^{I} = P_{din}^{\max} + P_{av2} \cdot \mu_2, \tag{4}$$

where P_{av2} - average value of the vertical load from the wheel of the second wheel pair on the rail;
μ_2 - ordinate of the line of influence of bending moments of the rail in the section of the track located under the wheel of the second wheel pair.

Maximum design axial stresses (in MPa) in the rail bottom are calculated using the formula

$$\sigma_o^p = \frac{P_{equ}^{I}}{4kW}, \tag{5}$$

where k - coefficient of relative stiffness of rail base and rail, cm^{-1},
W - moment of rail resistance along the bottom of the sole, cm^3.
k and W- values for different topsides structures are shown in Table 2 [20].

The coefficient *f* in the areas under consideration is calculated using the formula

$$f = \frac{\sigma_k^e + \Delta\sigma_o}{\sigma_o^p}, \tag{6}$$

where σ_k^e - maximum probable value of edge stress in the rail base, MPa.
$\Delta\sigma_o = \sigma_o^p - \sigma_o^e$ - difference between the design and experimental axial stresses for the section of the track where the tests were carried out.

Based on the obtained values, an analytical expression is compiled describing the dependence of the coefficient *f* on the radius of the curve, from which its value is calculated for sections of the path with a different radii of curvature. Results of calculation *f* and corresponding normalized value of axial stress, determined by the formula:

$$\left[\sigma_o = \frac{[\sigma_k]}{f}\right], \tag{7}$$

are summarized in Table 5, where $[\sigma k]$ = 240 MPa - allowable stresses in the edges of the rail foot.

Figures 11 and 12 show graphs of the design axial stresses at the bottom of rails for track with different topsides structures versus speed. Lines corresponding to the level of the normalized value of axial stresses for paths with different radii of curvature are applied to the

Table 5 *Values of coefficient f and corresponding normalized values of axial stress*

Indicator	Direct	Radii of curves, m							
		1000	900	800	700	600	500	400	300
f	1.23	1.53	1.56	1.60	1.65	1.72	1.82	1.97	2.22
$[\sigma_o]$, MPa	195	157	154	150	145	140	132	122	108

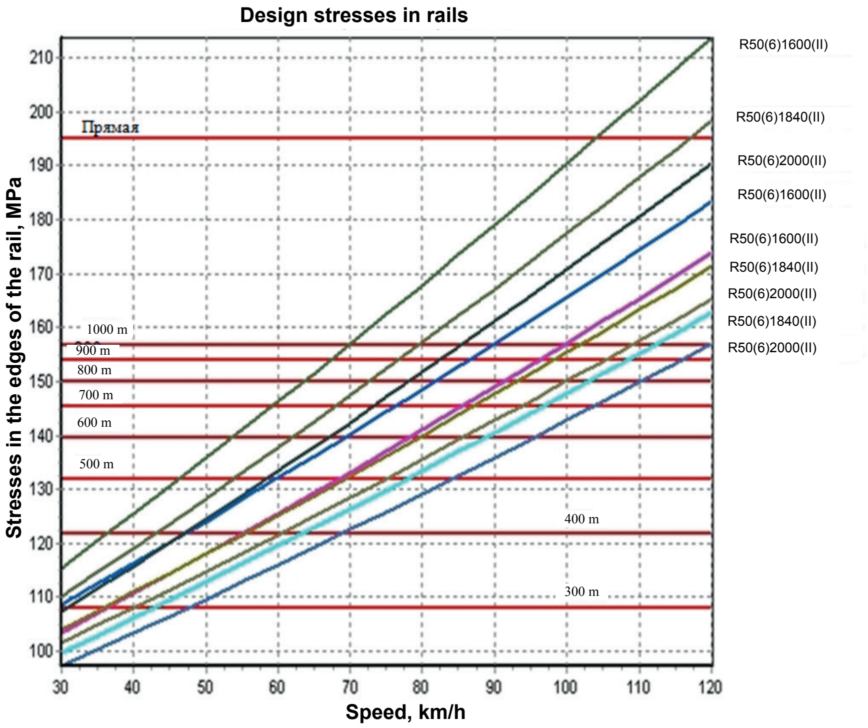

Figure 11 *Design axial stresses in rails under the influence of a diesel locomotive TE33A on the track with rails R 50*

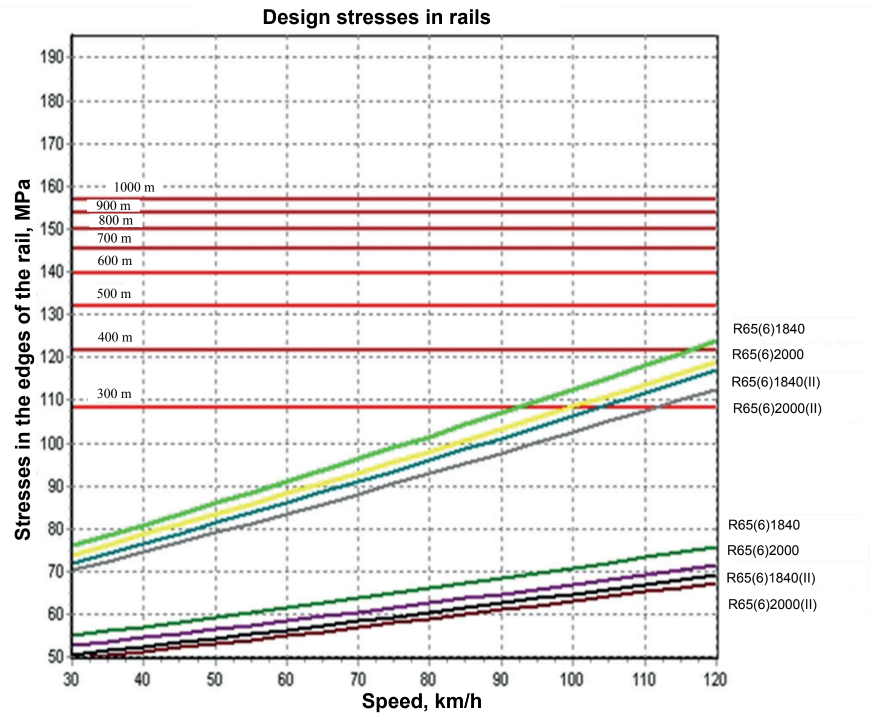

Figure 12 *Design axial stresses in rails under the influence of a diesel locomotive TE33A on the path of structure R 65*

Table 6 *Allowable diesel locomotive speeds TE33A*

Rail Type	Radius of curves, m								
	Direct	1000	900	800	700	600	500	400	300
R65(6)1840	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R65(6)2000	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R65(6)1840	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R65(6)2000	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R65(6)1840	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R65(6)2000(II)	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R65(6)1840(II)	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R65(6)2000(II)	K-120	K-120	K-120	K-120	K-120	N-115	N-105	N-95	N-80
R50(6)1600(II)	K-120	95	95	90	80	75	65	50	35
R50(6)1840(II)	K-120	110	105	100	95	85	75	60	40
R50(6)2000(II)	K-120	120	115	105	100	90	80	65	45
R50(6)1600(II)	C-100	85	80	80	70	65	55	45	25
R50(6)1840(II)	C-100	100	95	90	85	75	65	50	35
R50(6)2000(II)	C-100	C-100	100	95	90	80	70	55	35
R50(6)1600(II)	100	65	65	60	55	50	45	35	20
R50(6)1840(II)	C-100	75	75	70	65	60	50	40	25
R50(6)2000(II)	C-100	80	80	75	70	65	55	45	30

Note 1: The letter “K” before the speed digit means the structural speed of the diesel locomotive.

Note 2: The letter “N” before the speed digit means the speed limit in the curved sections of the track by the value of the accepted allowable outstanding acceleration of 0.7 m/s2 at the elevation of the outer rail of 150 mm.

Note 3: The letter “C” after the speed digit means the speed limit by order No. 41 of the Ministry of Railways of the Russian Federation [22].

Note 4: Numerals of speed of movement without letters mean limitation of speed of movement according to conditions of the rail strength.

same figures. The abscissa of the point of intersection of these lines with the graph of the dependence of the design axial stress in the bottom of the rails on the speed corresponds to the permissible speed of the locomotive on the tracks of this type of topsides with the specified radius of curvature [13-14].

When drawing up the table of circulation rates, the inadmissibility of exceeding the outstanding acceleration in curves 0.7 m/s^2, the limitation on the ratio of lateral and vertical forces from the rail to the sleeper are taken into account. In addition, the speed of circulation of the locomotive should not exceed the structural speed provided for by the technical specifications and, according to the order of the Ministry of Transport and Communication of the Republic of Kazakhstan No. 41Ts of 12.11.01 [21], with rails R50 on gravel and sand ballast of 100 km/h. The results of studies to determine the speeds of movement of the diesel locomotive TE33A according to the criteria of the stability of the rail-sleeper grid by ballast shear, the strength of the bearing elements of the upper structure of the track and the non-exceedance of the outstanding acceleration of 0.7 m/s^2 in curves with an elevation of the outer rail of 150 mm are shown in Table 6.

4 Conclusions

According to the objectives of the study, the following results were obtained:

1) The coefficient of vertical dynamics of the first stage of suspension of the TE33A diesel locomotive in all considered modes fully complies with the established standards. The coefficient of vertical dynamics of the second stage of suspension of the TE33A locomotive in curves with a radius of 290 m to speeds with outstanding acceleration does not exceed 0.7 m/s^2, that is, it is within the permissible value.
2) The coefficient of stability margin against wheel derailment meets the requirements of NB ZhT TsT 02-98 in all considered modes. The minimum value of the coefficient was 1.7 at a speed of 70 km/h.
3) An experimental evaluation of dynamic stresses in the edges of the sole of the rail on curved sections with a radius of 290 m was performed, and the following values were obtained:

- in curved sections with a radius of 290 m with an outstanding acceleration of 0.7 m/s^2 - 198 MPa;
- with an outstanding acceleration of 1.0 m/s^2 - 221

MPa.
The values obtained correspond to acceptable standards.

4) The parameters of the frame (lateral) forces on straight and curved sections of the railway track are determined. The values of lateral forces in straight sections of the path up to the structural speed and in curved sections up to speeds with an outstanding acceleration of 0.7 m/s^2 are within the acceptable limits. However, in curves with a radius of 290 m, the values of lateral forces at all speeds are in the maximum permissible values.

Grants and funding

The authors received no financial support for the research, authorship and/or publication of this article.

Conflicts of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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