*Удосконалено метод теоретичного визначення направляючої сили з урахуванням дії поперечних сил крипа та кута нахилу направляючої сили до вертикальної осі.*

*Встановлено, що при визначенні направляючої сили потрібно перевіряти зазор між гребенем колеса та головкою рейки, що важко здійснити без комп'ютерного моделювання.*

*При визначенні рамної сили на осі колісної пари застосований комплексний підхід, який враховує геометричні нерівності рейкової колії як у вертикальній, так і в горизонтальній площині; поздовжні та поперечні сили крипа в точці контакту «колесо-рейка»; вплив суміжних колісних пар вагона дизель-поїзда.*

*Отримано залежності рамної та направляючої сили від швидкості руху екіпажу та величини амплітуди горизонтальної нерівності рейкової колії. Встановлено, що під час руху в прямій ділянці колії збільшення швидкості руху від 0 м/с до 50 м/с призводить до зростання числового значення рамної та направляючої сили відповідно: 1-а колісна пара – до 8,3 кН, 2-га колісна пара – 19,37 кН; 1-а колісна пара – до 31,38 кН, 2-га колісна пара – до 46,83 кН. Збільшення амплітуди горизонтальної нерівності рейкової колії, яке є однією із першопричин появи вимушених коливань надресорної будови транспортного екіпажу, також призводить до зростання числових значень сил взаємодії рухомого складу з рейковою колією. Все це може призвести до підвищеного силового впливу колісної пари на рейкову колію та негативного впливу на основні критерії безпеки руху.*

*Досліджено вплив швидкості руху екіпажу на величину поперечних сил крипа. Встановлено, що при збільшенні швидкості руху екіпажу від 0 м/с до 50 м/с ці сили зростають в діапазоні: I колісна пара – від 0 до 15,75 кН; II колісна пара – від 0 до 29,22 кН. Це говорить про неможливість нехтуванням поперечними силами крипа при визначенні направляючої сили.*

*Виконано порівняння числових значень направляючої сили, визначених за різними методиками. Встановлено, що методика, яка використовується при проведенні судових залізнично-транспортних експертиз, дозволяє проводити нижню оцінку виконання умови сходу колеса з рейки. При цьому розрахунки за формулою, яка була удосконалена у даній роботі, дають можливість отримувати результати, найбільш наближені до реальних умов експлуатації.*

*Проведено порівняння експериментального та теоретичного розрахункового значення рамної сили на першій колісній парі вагона дизель-поїзда, а також показано їх практичне співпадіння. Відхилення порівнюваних значень рамної сили знаходиться у межах 7,2 %*

*Ключові слова:дизель-поїзд, рамна сила, колісна пара, направляюча сила, залізнична колія, нерівність*

UDC 625.032: 629.4.015: 629.424.25-833.6

DOI: 10.15587/1729-4061.2018.149838

# IMPROVING A METHODOLOGY OF THEORETICAL DETERMINATION OF THE FRAME AND DIRECTING FORCES IN MODERN DIESEL TRAINS

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## 1. Introduction

The cases of rolling stock derailing taking place on railroads of Ukraine and the world inflict considerable material losses. When investigating such cases, forensic experts encounter the problem of objective assessment of the causes of railway traffic accidents. The problem arises because of outdated branch regulatory base or even lack of relevant regulatory documents.

The main lines of development of rail transport in accordance with the Strategy of Development of Rail Transport Until 2020 include renewal of passenger rolling stock and gradual introduction of high-speed trains. Therefore, a modern diesel train DPKr-2 [1] was built by the Kriukiv railway car building plant (Kremenchuk, Ukraine). It was designed mainly for providing suburban passenger traffic on railway sections with small passenger streams having 1,520 mm wide tracks in Ukraine, CIS countries, Latvia, Lithuania and Estonia.

In diesel train operation, forces occur that can reach significant values and cause intensive oscillations. As a result, the railway track and the rolling stock may be subjected to serious damages. The nature of emerging oscillations is largely determined by the structure and properties of the rolling stock as a dynamic system: moments of inertia, height of the center of gravity, types of joints between the structure elements and so on. It should be noted that the dynamic system oscillations are influenced by the rail track state as well.

In order to formulate safety criteria for diesel train operation, it is essential to determine frame and directing forces that characterize wheel set loading and level of the wheel and track force interaction.

Hitherto, no theoretical studies of such forces taking into account geometric irregularity of the track and the wheel set positioning in it were carried out for the DPKr-2 diesel train cars. Determination of transverse creep forces arising on the wheel rolling surface of the rail and analysis of their influence on magnitude of the forces under study deserve particular attention. Therefore, improvement of the methodology of theoretical ascertaining of frame and directing forces for the DPKr-2 diesel train is an actual task for further establishment of derailing resistance criteria and corresponding rise of the diesel train operation safety.

## 2. Literature review and problem statement

It is noted in [2] that the directing force, *Y,* is acting transverse to the track axis and transmitted from the wheel flange to the rail head. The frame force, $Y_f$, is a resultant of the side forces taken by wheels of a one-wheel set (Fig. 1).

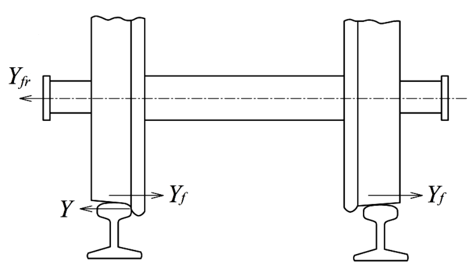


Fig. 1. Horizontal forces acting on the wheel set

Depending on direction of the transverse force acting on the wheel rolling surface of the rail, the frame force is determined from formula:

$$Y_{fr.} = Y_s + Y_f \tag{1}$$

or

$$Y_{fr.} = Y_s - Y_f, \tag{2}$$

where $Y_s$ is the side force; $Y_f$ is the transverse force acting on of the wheel rolling surface of the rail.

The side force determined in [2] is the total action of the directing force and the transverse force acting on the wheel rolling surface of the rail.

However, the frame and directing forces are determined without reflection of the dynamic processes occurring when the car passes vertical and horizontal track irregularities.

The above-mentioned methodology was widened and refined in [3]. These changes relate to the variable value of friction coefficient in various carriage inscribing schemes and the effect of longitudinal compressive forces arising during train braking on horizontal forces.

It was noted in [4] that sliding friction forces can be neglected in practical calculations when determining directing and frame forces. Besides, it was noted there that in a quiescent condition of the carriage when speed is zero, directing and frame forces should be zero which is not true as judging by the results obtained with the use of the above methodologies. In this connection, the issue of new interpretation of the above notions and search for analytical expressions determining numerical values of the frame and directing forces has become relevant. Estimation of values of emerging transverse forces is important for establishment of traffic safety criteria and conducting forensic examination of railroad accidents.

An object-oriented simulation method was proposed [5] that could be used to predict some aspects of dynamic rolling stock behavior. When modeling behavior of each object, a virtual prototype of the rolling stock is built and as a result of modeling, critical speed, forces arising in wheel set and track interaction, etc. can be determined. However, this method does not take into account accidental nature of the rail track irregularity which has a major impact on dynamic parameters of the rolling stock.

Studies of effect of track irregularity on dynamic forces in the wheel-rail contact point were conducted in [6]. To this end, two two-dimensional and two three-dimensional numerical models were built. They describe the process of interaction of the rolling stock with the railway track. Their disadvantage consists in that with increase in the difference between irregularity of the right and left rails, the results obtained in simulation significantly differ from experimental results.

A numerically efficient model was developed in [7]. It describes three-dimensional rolling stock dynamics. For simulation of rolling stock dynamics, Matlab-Simulink and Adams Rail graphical environments of simulation modeling were used. This model features estimation of local deformations in the wheel-rail contact area.

An experimental approach to determination of the forces acting in interaction of a wheel set with a rail track was presented in [8]. The method for determining the forces of interaction is based on analysis of bending deformation of the rail track in a transverse plane.

A method of inversion in the time domain was proposed in [9] for studying vertical and horizontal forces in a wheel-rail contact. It was established that the proposed mathematical model has higher accuracy in comparison with the results obtained in simulation in the Simpack software package. However, lack of comparison of theoretical results with experimental ones is a significant drawback of this method.

The study [10] considers a methodology of calculating distortion of the track geometry taking into account uneven accumulation of residual deformations along the track. A forecast of distortion of the track geometry for a given design of the railway track and the existing main rolling stock was presented based on the developed methodology.

It was established in [11] that design features of the railroad rolling stock have significant influence on distortion of the railway tracks and point frogs.

Based on analysis of design of the carriage part of the diesel train car and the selected mechanical models of its main parts conducted in [12], a spatial mechanical model of the DPKr-2 diesel train was built. To describe oscillations of the mechanical model of the diesel train, a mathematical model consisting of 38 differential equations of the second order [13, 14] was built.

As a result of integration of differential equations of motion with the numerical Runge-Kutta method in the Maple18 computation environment, values of all generalized coordinates and force systems acting between the bodies were found for each instant of time. This has made it possible to proceed to calculation of dynamic values of the frame and directing forces. However, this technique does not take into account influence of vertical and horizontal irregularity of the railway track on values of the frame and directing forces and position of the rolling stock relative to the track rails.

Analysis of the published studies shows a discrepancy among the methods of calculating the force interaction between the rolling stock and track. The considered methodologies take into account separate factors of influence on the value of frame and directing forces without simultaneous accounting for geometric irregularity of the rail track in vertical and horizontal planes, transverse and longitudinal creep forces acting in the wheel-rail contact zone.

Solution of this problem can be achieved by applying an improved spatial mathematical model of rolling stock which includes all of the above factors and allows one to take into account influence of adjacent wheel sets of the DPKr-2 diesel train.

## 3. The aim and objectives of the study

The study objective was to improve the methodology of theoretical determination of frame and directing forces and their analysis during diesel train movement along the straight section of the railway track. This will ensure further development of criteria of traffic safety assessment taking into account geometric irregularity of the rail track in vertical and horizontal planes and position of the wheel set relative to the rails.

To achieve this objective, the following tasks must be solved:

– to analyze creep forces, frame and directing forces depending on geometrical state of the track and speed of the diesel train;

– to compare the results of calculation of the directing force with application of the improved and existing methodologies;

– to conduct experimental study of the frame force and estimate convergence of values of the frame force obtained by application of the improved methodology and experimentally.

## 4. Analytical determination of the frame force

The frame force is understood as a horizontal force transmitted to the axle of the wheel set and acting on the carriage frame sides in a direction transverse to the track axis.

Using the mathematical model presented in [13], the frame force is determined by the formula:

$$Y_p = 2s_{tran.} \cdot \left( \left( y_{trol.j} \pm a_1 \psi_{trol.j} + a_7 \theta_{trol.j} \right) - \left( y_{w.p.i} \right) \right), \quad (3)$$

where $s_{\mathrm{tran.}}$ is the transverse stiffness of the spring of an axle box spring suspension; $y_{\mathrm{trol}.j}$, $\psi_{\mathrm{trol}.j}$, $\theta_{\mathrm{trol}.j}$, $y_{\mathrm{w.p}.i}$ are generalized coordinates of side offset, wobbling and lateral swaying of the carriage and wheel set obtained in calculations of the above spatial mathematical model; $a_1$, $a_7$ are linear dimensions.

The upper signs "+" and "–" in formula (3) are used for the first- and second-wheel set of the carriage, respectively.

Geometric irregularity of the left and right rails is taken as perturbation both in vertical and horizontal planes:

$$\eta_{\mathrm{ver}} = H_{\mathrm{ver}} \sin \frac{2\pi}{L_{\mathrm{ver}}} \upsilon t, \quad (4)$$

$$\eta_{\mathrm{hor}} = H_{\mathrm{hor}} \sin \frac{2\pi}{L_{\mathrm{hor}}} \upsilon t, \quad (5)$$

where $H_{\mathrm{ver}}$, $H_{\mathrm{hor}}$ is the amplitude of vertical and horizontal irregularity; $L_{\mathrm{ver}}$, $L_{\mathrm{hor}}$ is the length of the vertical and horizontal irregularity; $\upsilon$ is the speed of movement.

The following parameters of the DPKr-2 diesel train car were taken in calculations: weight of the body, carriage frame and wheel set, respectively: $w_{\mathrm{b.}}$=39.65 t, $w_{\mathrm{trol.}}$=4.6 t, $w_{\mathrm{w.p.}}$=1.52 t; the moments of inertia of the body and the car carriage of the DPKr-2 diesel train relative to the axes $X$, $Y$, $Z$: $J_{\mathrm{x\ b.}}$=11.43 t·m$^2$, $J_{\mathrm{x\ trol.}}$=5 t·m$^2$, $J_{\mathrm{y\ b.}}$=1,570.21 t·m$^2$, $J_{\mathrm{y\ trol.}}$=6.6 t·m$^2$, $J_{\mathrm{z\ b.}}$=1,570.21 t·m$^2$, $J_{\mathrm{z\ trol.}}$=10 t·m$^2$; vertical and transverse stiffnesses of the axle box suspension springs $s_{\mathrm{ver.}}$=1,136.6 kN/m, $s_{\mathrm{tran.}}$=1,403.5 kN/m; the moments of inertia of the wheel set relative to the $X$ and $Z$ axes: $J_{\mathrm{x\ w.p.}}$= =2.3 t·m$^2$, $J_{\mathrm{z\ w.p.}}$=1.2 t·m$^2$.

The calculated maximum values of the frame force acting in the first- and second-wheel sets corresponding to different speeds of the carriage are shown in Fig. 2.

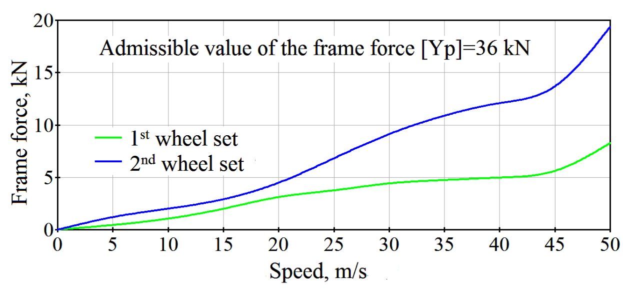


Fig. 2. Dependence of the frame force on motion speed at $L$=15 m, $H$=0.006 m

An analysis of the frame force dependence on the speed of motion shows that with an increase in speed to 50 m/s, the frame force monotonically increases up to 8.3 kN for the first wheel set, and up to 19.37 kN for the second wheel set.

Irregularity of the track rails is the source of forced fluctuations in the structure of the transport carriage above springs which eventually result in dynamic loads on the elements of the rolling stock structure and the railway track. Let us construct dependence of the frame force on the amplitude of horizontal irregularity of the rail track at a steady speed.

The maximum values of the frame force obtained for the first- and second-wheel sets corresponding to different values of amplitude of the horizontal irregularity of the rail track are given in Fig. 3.

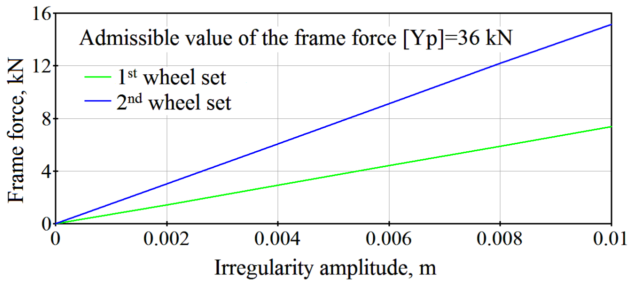


Fig. 3. Dependence of the frame force on the amplitude of horizontal irregularity at $L$=15 m, $V$=30 m/s

Calculation results show that when irregularity amplitude increases from 0 m to 0.01 m, the frame force increases in the range from 0 kN to 7.36 kN for the first wheel set and from 0 kN to 15.16 kN for the second wheel set.

## 5. Analytical determination of the directing force

Let us turn to determination of numerical value of the directing force that occurs in the straight track section. It was assumed in simulation of the wheel-rail contact that reactions in the contact zone between the wheel and the rail act in longitudinal, $F_x$, and transverse, $F_y$, directions and are described by the nonlinear hypothesis of creep [15].

Dependences of these reactions on relative slip speed are determined from formulas:

$$F_x = K\frac{\upsilon_x}{\upsilon}, \tag{6}$$

$$F_y = K\frac{\upsilon_y}{\upsilon}, \tag{7}$$

where $K$ is the coefficient of creep; $\upsilon_x$ is the speed of longitudinal slip of the wheel on the rail; $\upsilon_y$ is the speed of transverse slip of the wheel on the rail; $\upsilon$ is the carriage speed.

Longitudinal slip speeds for the wheels of the first carriage are determined as:

$$\upsilon_{x1} = \upsilon\frac{\Delta r_1}{r_1} + \ddot{\psi}_{\text{w.p.1}} S_1, \tag{8}$$

$$\upsilon_{x2} = \upsilon\frac{\Delta r_2}{r_2} - \ddot{\psi}_{\text{w.p.1}} S_2, \tag{9}$$

$$\upsilon_{x3} = \upsilon\frac{\Delta r_3}{r_3} + \ddot{\psi}_{\text{w.p.2}} S_3, \tag{10}$$

$$\upsilon_{x4} = \upsilon\frac{\Delta r_4}{r_4} - \ddot{\psi}_{\text{w.p.2}} S_4. \tag{11}$$

The following notations were taken in expressions (8) to (11): $\Delta r_{1\text{-}4}$ is growth of radii of wheels of the first carriage during movement of the diesel train car; $r_{1\text{-}4}$ are mean radii of the first carriage wheels.

Transverse slip speeds for the wheels of the first carriage are determined from formulas:

$$\upsilon_{y1} = \upsilon_{y2} = \dot{y}_{\text{w.p.1}} - \upsilon\psi_{\text{w.p.1}}, \tag{12}$$

$$\upsilon_{y3} = \upsilon_{y4} = \dot{y}_{\text{w.p.2}} - \upsilon\psi_{\text{w.p.2}}. \tag{13}$$

Similarly, slip speeds for the wheels of the second carriage are determined.

The increase in the wheel radii depends on the following: transverse displacement of the wheel set; horizontal irregularity; numerical value of conicity of the wheel profile at the point of contact with the rail; nominal gaps between the wheel flange and the inside edge of the rail head.

The growth of radii of the first carriage wheels is found from formulas:

$$\Delta r_1 = n_2^{\text{I}} \cdot y_{\text{w.p.1}} - n_2^{\text{I}} \cdot \eta_{\text{hor.1}} + \delta_{\text{1nom.}} \cdot \left(n_1 - n_2^{\text{I}}\right), \tag{14}$$

$$\Delta r_2 = -n_2^{\text{II}} \cdot y_{\text{w.p.1}} + n_2^{\text{II}} \cdot \eta_{\text{hor.2}} + \delta_{\text{2nom.}} \cdot \left(n_1 - n_2^{\text{II}}\right), \tag{15}$$

$$\Delta r_3 = n_2^{\text{III}} \cdot y_{\text{w.p.2}} - n_2^{\text{III}} \cdot \eta_{\text{hor.3}} + \delta_{\text{3nom.}} \cdot \left(n_1 - n_2^{\text{III}}\right), \tag{16}$$

$$\Delta r_4 = -n_2^{\text{IV}} \cdot y_{\text{w.p.2}} + n_2 \cdot \eta_{\text{hor.4}} + \delta_{\text{4nom.}}^{\text{IV}} \cdot \left(n_1 - n_2\right). \tag{17}$$

Similarly, the increase in radii is determined for the second carriage wheels.

Creep coefficient, $K$, is determined from formula:

$$K = \gamma\sqrt{R_k r}, \tag{18}$$

where $\gamma$ is empirical coefficient; $R_k$ is the vertical wheel-rail reaction.

In models that describe movement of the carriage in straight sections of the track, it is recommended to take $\gamma$=800÷1,600 [16].

Present effect of the carriage speed on magnitude of the greatest transverse creep forces in the first- and second-wheel sets of the diesel train car (Fig. 4) using the spatial mathematical model of the diesel train car.

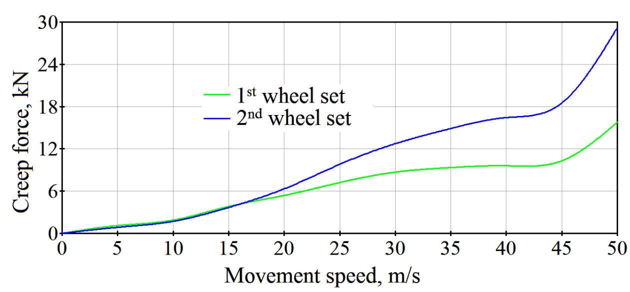


Fig. 4. Dependences of transverse creep forces on the carriage speed at $L$=15 m, $H$=0.006 m

Analysis of the above-mentioned dependences gives grounds to conclude that with an increase in speed from 0 m/s to 50 m/s, transverse creep forces increase in the range from 0 kN to 15.75 kN for the first wheel and from 0 kN to 29.22 kN for the second wheel.

It was mentioned in [4] that for practical calculations when determining directing force, the forces of slip friction can be neglected. This is explained by the fact that the component of speed of the wheel center in transportation motion of the carriage is considerably greater than the component of speed of the same wheel center in relative motion of the carriage. It was concluded on the basis of this fact that the slip friction force acting along the rail track is much greater than that of the same wheel acting across the rail track.

However, the results of our calculations show that the transverse creep forces under operating conditions can reach quite significant values, therefore their neglect in determining the directing force can lead to false results.

Analytical expression for determining the directing force, $N_d$, is found taking into account the effect of transverse creep forces and the angle of inclination of the directing force to the vertical axis (Fig. 5).

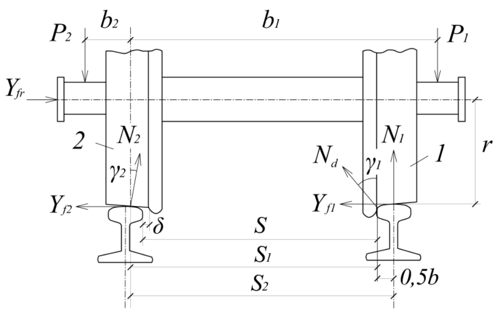


Fig. 5. The forces acting on a wheel set at its interaction with a rail track

In this case, the directing force is considered to be a resultant of component creep forces and the frame force and is determined from the following expression:

$$N_d = \frac{Y_{fr} - Y_{f1} - Y_{f2}}{\sin\gamma_1}, \tag{19}$$

where $Y_{fr}$ is the frame force; $Y_{f1,2}$ are the creep forces acting on the wheels of the first and second wheels, respectively; $\gamma_1$ is conicity of the wheel set wheel flange.

It should be noted that the directing force occurs when the gap δ in the rail track becomes zero.

In the case where clearance between the wheel flange and the rail head is in the range from 0 to the maximum value, there is no directing force.

The following designations are used in Fig. 5: $P_1$, $P_2$ are vertical forces which are transferred from the carriage frame to the wheel set wheels 1 and 2; $N_1$, $N_2$ are vertical reactions acting from the rail sides on wheels 1 and 2, respectively; $N_d$ is the reaction that occurs at the point of interaction of the wheel 1 flange with the working face of the rail and is called the directing force; $b_1$, $b_2$ is the distance between the lines of action of $P_1$ and $P_2$ forces and the point of force application, $N_2$; $b_3$ is the distance between the $y$ axis and the line of force action, $Y_{fr}$; $S_2$ is the distance between the points of application of forces $N_1$ and $N_2$; $\gamma_1$, $\gamma_2$ is conicity of flange and rim of the first and second wheel, respectively; δ is clearance in the rail track; $S$ is the track width; $S_1$ is the distance between the points of application of forces $N_2$ and $N_d$.

Using the adopted diesel train model and the formula for determining the directing force, calculate and analyze dependence of the directing force on the speed of motion (Fig. 6).

With an increase in speed from 0 m/s to 50 m/s, the highest value of the directing force increases to 31.38 kN in the first wheel set and 46.83 kN in the second wheel set.

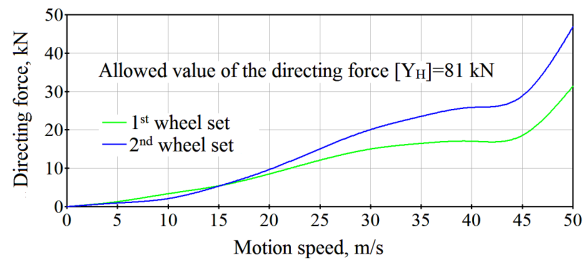


Fig. 6. Dependence of the directing force on the carriage speed at $L$=15 m, $H$=0.006 m

Similar results can be obtained when conducting calculations for any wheel of a of a car carriage wheel set.

Compare the calculated value of the directing force with the forces found using the methodologies of studies [2, 4].

The author of study [4] proposes to determine the directing force in practical calculations by formula:

$$N_d = \frac{Y_{fr} + (P_1 + P_2)\mathrm{tg}\gamma_2 - \left[\left(P_1 b_1 - P_2 b_2 + Y_{fr} b_3\right)\mathrm{tg}\gamma_2\right] \div S_2}{\sin\gamma_1 + (1-\xi)\cos\gamma_1 \cdot \mathrm{tg}\gamma_2}. \tag{20}$$

Unlike the accepted formula (19), it does not take into account transverse creep forces. Formula (21) which does not take into account conicity of the not running-on wheel ($\gamma_2$=0) is the partial case of formula (20):

$$N_d = \frac{Y_{fr}}{\sin\gamma_1}. \tag{21}$$

Formulas (20) and (21) are mainly used in the practice of forensic examinations.

The frame force is considered in [2] as a function of the directing force and the friction forces acting transverse to the track axis. However, this formula does not take into account the fact that the directing force is not horizontal but lies at an angle $\gamma_1$ to the vertical.

Proceeding from the above, the directing force is found by the formula:

$$N_d = Y_{fr} - Y_{f1} - Y_{f2}. \tag{22}$$

Comparison of the results obtained in calculation of the directing forces using the improved formula (19) and formulas (21), (22) is shown in Fig. 7.

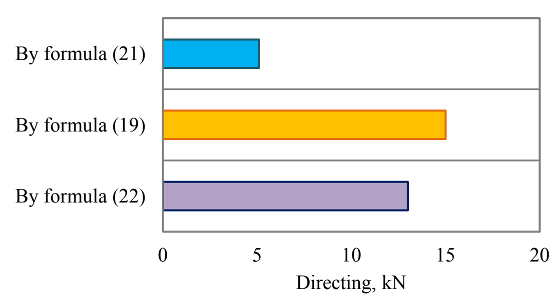


Fig. 7. Comparison of the directing force acting in the first wheel set obtained by different methodologies at train speed of 30 m/s

According to the EU standard BS EN 14363:2005 [17], the rolling stock is considered to be resistant to derailing if the following condition is fulfilled at each test stage:

$$\left(\frac{Y_d}{Q}\right)_{\max} \leq \left(\frac{Y_d}{Q}\right)_{\lim}, \tag{23}$$

where $Y_d$ is horizontal directing force; $Q$ is vertical force applied by the wheel to the rail.

It follows from this condition that opposed to formula (19), the use of formula (21) underestimates the condition of derailing. After analyzing values of the directing forces obtained by different methodologies (Fig. 7), it is recommended to use an improved formula (19) when assessing criteria of the diesel train safety,

## 6. Experimental determination of the frame force

Experimental values of the frame force acting on a straight section of the track for the first wheel set at speed of the DPKr-2 diesel train car of 30 m/s were obtained by the Branch Research Laboratory of Rolling Stock Dynamics and Strength of the Academician V. Lazarian National University of Railway Transport, Dnipro, Ukraine.

Experimental studies of the frame force were carried out at the rolling stock testing site of Ukrainian railroads (Novomoskovsk-Balivka and Diyivka-Verkhovtsevo sections of the Prydniprovska Railroad).

The upper track structure included the following:

– R65 rails,
– metal rail bearing plates and
– reinforced concrete sleepers.

Special polymer gaskets between the sleepers and the rail bearing plates were used for soundproofing.

The tests were carried out using strain gauges of KF5-P1-10-200 model mounted on the driver of the box-to-carriage frame connector (Fig. 8).

The main characteristics of the strain gauge:

– maximum measured deformation: ±3,000 μm/m;
– sensitivity to deformations: 1.9 to 2.3;
– relative measurement error: ±2 %.

All measurement data were transmitted to the corresponding software modules using a tensor amplifier of TMA-32 model with a filter unit and an analog-to-digital converter. The software modules were developed at the Academician V. Lazarian National University of Railway Transport, Dnipro, Ukraine. The frame forces were determined during operation of the PPKr-2 diesel train by means of this equipment and the developed software modules.


Fig. 8. Schematic of location of strain gauges of KF5-P1-10-200 type in the DPKr-2 train car

Experimental and theoretical values of the frame force of the diesel train car moving with speed of 30 m/s in the straight railway track section are shown in Fig. 9.

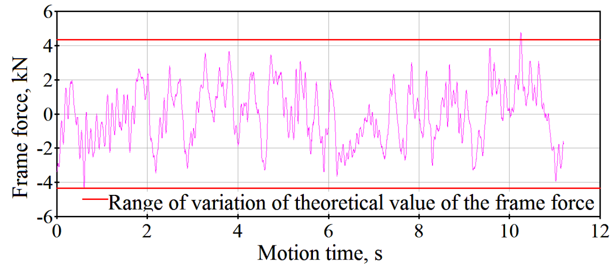


Fig. 9. Comparison of experimental values of the frame force acting in the diesel train car at a speed of 30 m/s in a straight railway track section with the values obtained in theoretical calculations

As can be seen from Fig. 9, the experimental values of the frame force fall into the theoretical range of values from −4.36 kN to +4.36 kN obtained from formula (3).

Comparison of the maximum values of the frame force obtained theoretically with the maximum experimental values shows high convergence of the results. Deviation of the compared force values was within 7.2 %.

Consequently, an improved formula (19) for determining the directing force can be used in solving the problems of assessing the diesel train safety.

## 7. Discussion of results obtained in studying the frame and directing forces

Analysis of dependence of the frame force on the speed of the diesel train has shown that an increase in speed from 0 m/s to 50 m/s results in an increase in the value of the frame force in the range: from 0 kN to 8.3 kN for the first wheel set and from 0 kN to 19.37 kN for the second wheel set.

Dependences of the frame force on the amplitude of horizontal irregularity of the rail track at steady speeds were constructed. It was established that with an increase in the amplitude of irregularity from 0 m to 0.01 m, the maximum frame force increased as follows. The first wheel set: from 0 kN to 7.36 kN; the 2$^{nd}$ wheel set: from 0 kN to 15.16 kN.

It was shown that the increase in speed of the diesel train from 0 m/s to 50 m/s results in a growth of the transverse creep forces from 0 to 15.75 kN for the first wheel set and up to 29.22 kN for the second wheel set.

When determining the directing force, it is necessary to check value of the gap between the wheel flange and the rail head because the wheel and the rail do not interact in the presence of a gap and the force is zero. In the process of numerical integration of the differential equations of the diesel train car motion, this test is simplest to perform.

Having analyzed the graphs of dependence of the directing force on the carriage speed, it is evident that when speed increased from 0 m/s to 50 m/s, value of the directing force increased in the range from 0 kN to 31.38 kN for the first wheel set and from 0 kN to 46. 83 kN for the second wheel set.

Similar dependences can be obtained when conducting calculations for any wheel of the car carriage wheel set.

The use of formula (21) results in underestimation of the derailing condition as opposed to formula (19). Therefore, taking in consideration the European Union standard BS EN 14363:2005, it is recommended that an improved theoretical formula (19) be used when assessing the diesel train safety criterion. This formula makes it possible to calculate value of the directing force while taking into account all main factors, such as geometric irregularity of the track in horizontal and vertical planes and the impact of creep forces and adjacent wheel sets.

One of the drawbacks of this study is that the improved methodology for theoretical determination of the frame and directing forces acting in the PPKr-2 diesel train and the values of corresponding forces obtained on its basis can be used only for direct sections of the railway track. Therefore, further studies of the force interaction of the rolling stock with the rail track will be carried out in curved railway sections.

Complexity of the task is determined by the necessity of simulating the characteristics of the rail track in curved sections (with increased level of the external rail, changing radius of curvature, etc.). But obtaining of such results will enable assessment of the wheel stability indicators in terms of derailing which are used on railroads of Ukraine and Europe under various operating conditions and ensure objectivity of examination of railway accidents.

## 8. Conclusions

1. It was established that with an increase in speed from 0 m/s to 50 m/s, transverse creep forces grow in the range from 0 kN to 15.75 kN for the 1st wheel set and from 0 kN to 29.22 kN for the 2nd wheel set. In comparison with the directing forces emerging at the corresponding speeds, it makes 50 to 60 %.

The carriage speed growth from 0 m/s to 50 m/s results in an increase in the frame force reaching 8.3 kN in the first and 19.37 kN in the second wheel set at a speed of 50 m/s. Deterioration of the track state, namely increase in irregularity amplitude from 0 m to 0.01 m at a steady speed of 30 m/s also results in a growth of frame force to the level of 7.36 kN for the first and 15.16 kN for the second wheel set. Increase in the speed of movement and the amplitude of the horizontal irregularity of the rail track results in an increased power influence of the rolling stock on the track which in the end result can bring about derailing of the rolling stock.

It has been established that when the speed of motion increased from 0 m/s to 50 m/s, the value of the directing force increased in the range from 0 kN to 31.38 kN for the first and from 0 kN to 46.83 kN for the second wheel set.

2. In the course of comparison of the results of calculation of the directing force by improved and existing methodologies, it was established that the use of the formula of the directing force without taking into account transverse creep forces may result in an underestimated estimate of fulfillment of the derailing condition as opposed to the formula which was improved in this work. When assessing safety criteria for diesel trains, it is recommended to use the improved formula of the directing force determination.

3. When comparing experimental and theoretical values of the maximum frame force in the first wheel set of the diesel train car, their practical coincidence is observed which indicates adequacy of the mathematical model and performed calculations since discrepancy makes only 7.2 %.

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UDC 629.5.081.326
DOI: 10.15587/1729-4061.2018.150346

# IMPROVEMENT OF THE STRUCTURE OF FLOATING DOCKS BASED ON THE STUDY INTO THE STRESSED-DEFORMED STATE OF PONTOON

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***Проведено дослідження напружено-деформованого стану залізобетонних плит понтона композитного дока зі зменшеною кількістю набору. Використана уточнена розрахункова схема, при розрахунках згину плит стапель-палуби і днища понтона, яка враховує роботу арматури обох напрямків. Врахування роботи арматури обох напрямків дозволяє точно оцінити міцність конструкції і надати рекомендації щодо проектування конструкцій понтону з точки зору матеріалоємності і оптимального розміру. При моделюванні роботи бетону враховано, що бетон при розтягу має меншу жорсткість на розтяг, ніж на стиск.***

***Показано, що розроблена конструкція та технологія побудови композитного доку зі зменшеною кількістю набору у понтоні дозволяє розширити технологічні можливості побудови доків. Отримані результати розрахунків зведених напружень в плитах стапель-палуби понтона дока показали, що фактичні коефіцієнти запасу задовольняють вимогам міцності. Враховуючі що конструкція бетонних перекриттів сприймає в декілька разів більший момент спротиву ніж сталь можна збільшити проліт перекриття і рідше розставляти опори-перебірки. Внаслідок цього зменшуються витрати на матеріали та знижуються трудомісткість робіт при побудові доку.***

***Запропоновано конструкцію та технологію побудови композитного доку зі зменшеною кількістю набору у понтоні. Показано, що встановлення поперечних переборок між внутрішніми бортами через 4 шпації, тобто через 3 метри, а в бетонних баштах відсутність шпангоутів, флорів та бімсів дозволяє зменшити кількість матеріалів, а також знизити трудомісткість побудови доку. Наведені особливості вибору суднобудівного бетону орієнтованого на екстремальні умови роботи морських залізобетонних споруд***

***Ключові слова: плавучий композитний док, технологія побудови доків, залізобетонні секції, понтон, міцність залізобетонних плит***

## 1. Introduction

Permanent needs of world shipping for ship repairs, inspection, and control over the condition of vessels, maintenance work for the underwater part of vessels, predetermine the elevated demand for floating docks. Construction of docks is a profitable business, constituting one of the important directions to promote domestic products in the