1. Introduction

Ensuring the efficient operation of railroads as a leading sector of the transportation network requires the commissioning of modern rolling stock. In this case, making it more competitive predetermines the strictest requirements not only to the technical and economic indicators of rolling stock but also to the possibility of adapting the structures to the appropriate operating conditions.

It is known that one of the most loaded nodes in the design of rolling stock is automatic coupling equipment. During operation, this node experiences significant longitudinal-dynamic loads whose numerical values can be up to 3.5 MN – collision at shunting. In addition, significant loads emerge when operating the rolling stock on main tracks – braking, pulling off, etc.

To ensure that wagons are coupled to each other and to a locomotive, that they are kept at a certain distance from each other, and that the longitudinal efforts are transferred, the trains that include cars with a gauge of 1520 mm use the standard automatic coupling device SA-3 (Fig. 1). At present, there are the modernized variants of a given automatic coupler as well.

It is important to note that one of the main shortcomings of a given design is a significant cost, due to the use of a large number of components, specifically a coupler with the absorbing device.

This necessitates the implementation of new variants of automatic coupling devices. In this case, they should ensure the possibility to perceive and damp impact loads while meeting the conditions of strength for the rolling stock bearing structures. This would improve the operating...
efficiency of rolling stock and reduce the cost of unscheduled repairs.

Paper [1] reports the results of mathematical modeling of train longitudinal dynamics. The authors gave the main theoretical positions, on the basis of which they developed an apparatus for determining the longitudinal efforts in a train. However, they failed to pay attention to devising measures to reduce the longitudinal efforts in a train.

The measures to improve the efficiency of rolling stock brakes operation were addressed in [2]. The author tested a graphic-analytical algorithm for the dual wear of the pads, which makes it possible to determine the magnitude and direction of the action of force factors. However, the author did not conduct research on the influence of these factors on the longitudinal dynamics of a train.

Work [3] studied the longitudinal-dynamic forces in a cargo train using different types of absorbing devices (Sh-1-T, Sh-2-T, and Sh-2-V). The longitudinal dynamics were simulated employing the software Universal Mechanism UM8.1. The authors analyzed the values of the longitudinal efforts in a train and defined the type of absorbing device that is the most optimal one to use.

The issue of improvement of the design of an automatic coupling device in order to reduce the longitudinal loading on cars in a train was not considered in the cited work.

The influence of the length of a train on the longitudinal forces at braking is examined in [4]. The study was conducted in the programming environment MATLAB. Based on their calculations, the authors found the longitudinal loads that act on the train composed of a different number of cars. However, no measures to reduce the longitudinal loads were proposed.

Paper [5] determined the longitudinal dynamics in the coupled trains. Wireless measuring equipment was developed, which allowed the authors to find indicators for the longitudinal dynamics of a train. However, they did not consider the task of determining the longitudinal loads in a train by mathematical modeling.

The distribution of the longitudinal-dynamic forces for various characteristics of the hysteretic buffer was reported in [6]. The paper takes into consideration that the time of filling the brake cylinders with compressed air and the parameters of the car buffers in a passenger train are the same. That is, the cars have the same braking force.

The development of the longitudinal-dynamic forces in a passenger train at braking is examined in [7]. The authors considered the influence of the number of cars in a train, the gross weight, and a deviation of ±20 % for length on the braking force.

However, the authors of [6, 7] did not consider the issue of distribution and reduction of the longitudinal-dynamic forces in cargo trains.

Work [8] analyzed the longitudinal dynamics of a train under operating modes. In this case, the longitudinal dynamics and the coupling force were modeled according to the results of experimental research. The model was solved in the MATLAB software suite. The cited work did not consider the improvement of the design of devices that couple cars and a locomotive to reduce the longitudinal-dynamic load on a train.

The simulation of vehicle dynamics under loading operating regimes is reported in [9]. The calculations were conducted in the Universal Mechanism programming environment. The authors took into consideration the different microgeometry of the track. The maximal values of the longitudinal efforts in a train were derived. However, no measures to reduce the longitudinal forces in a train during operating modes were proposed in the cited work.

The evolution of the simulation of longitudinal train dynamics was considered in [10]. The authors defined a series of potential research topics regarding the longitudinal dynamics of trains, which should be solved at the present stage of the development of the railroad industry in order to ensure the safety of motion. However, the issues of improving the coupling devices in order to reduce the longitudinal efforts in a train were not considered.

Our analysis of literary sources makes it possible to conclude that it is expedient to conduct research aimed at determining the longitudinal dynamics of a railroad train. A feature of this research is that the units of rolling stock interact between themselves through the new conceptual coupler. This would reduce the longitudinal loading on a train during operating modes.

3. The aim and objectives of the study

The aim of this study is to determine patterns in the longitudinal loading on the rolling stock taking into consideration the application of the new conceptual couplers instead of the automatic coupling of the SA-3 type.

To accomplish the aim, the following tasks have been set:

- to simulate the longitudinal load on a railroad train equipped with the new conceptual couplers;
- to estimate the strength of the rod of the conceptual coupler.

4. Methods for determining the longitudinal load on a train at steady motion

It is known that the maximal longitudinal effort at steady motion along a track of a uniform profile emerges in an automatic coupling device between a locomotive and the train's cars (Fig. 2). The features of determining the longitudinal efforts in a train are described in [11].
In addition, the longitudinal effort in a train can be derived using the procedure given in [12]:

\[
N = \left[ \frac{T_i}{Q_i + Q} + \frac{w_{cp} - b_t}{Q} \right] \cdot \sum_{j=1}^{n} \left( Q_i: w_i + B_{Tj} \right).
\]

where \( T_i \) is the tangent traction force of a locomotive, kN; \( Q_i \) is the weight of a unit of a train with serial number \( i \), forward direction (at \( i=0 \) – the main locomotive, \( i=1, 2, 3, ..., n \) – cars), kN; \( w_{cp}, b_t \) are, respectively, the average resistance and average braking force per unit weight of the train, kN; \( n \) is the number of cars in the train; \( w_i \) is the specific motion resistance of a rolling stock unit, taking into consideration the track profile; \( B_{Tj} \) is the braking tangent forces of a locomotive and cars, kN.

The tangent traction force of a locomotive is determined from formula [13]:

\[
T_i = P_{max} \cdot \varphi_k \cdot \left( L \cdot m \right) \left( 0.18 + \frac{4}{22 + v} \right).
\]

where \( P_{max} \) is the maximum traction force of a locomotive, permitted by the conditions of adhesion between the wheels and rails, kN; \( \varphi_k \) is the estimated coefficient of adhesion between the wheels and rails; \( L \) is the load from the driving wheelset on rails, kN; \( m \) is the number of drive wheelsets of a locomotive equal to the number of traction engines; \( v \) is the estimated motion speed, km/h.

The resistance of locomotive motion is determined from [14]:

\[
w_i = 1.9 + 0.01 \cdot v + 0.0003 \cdot v^2.
\]

The resistance of four-axle cars can be found from:

\[
w_c = 0.7 + 3 + 0.1 \cdot v + 0.0025 \cdot \frac{v^2}{q}.
\]

where \( q \) is the axial load, kN/axle.

The average braking force per unit weight of the train is determined from:

\[
b_i = 1000 \cdot \varphi_i \cdot \varphi_{br}.
\]

where \( \varphi_i \) is the estimated brake coefficient of a train; \( \varphi_{br} \) is the estimated coefficient of friction of brake pads.

\[
\varphi_{br} = 0.36 \cdot \frac{v + 150}{2.6 \cdot v + 150}
\]

According to [15], the cargo car frame strength is calculated as follows:

1) at emergency braking by a pneumatic brake from any speed and at any condition of gaps in automatic coupling devices:
   - in a non-uniform train – 2.5 MN;
   - in a uniform train – 2.0 MN;
2) at complete service braking to stopping from any motion speed, as well as at controlled braking from a motion speed of 15 km/h and less:
   - in a non-uniform train – 2.0 MN;
   - in a uniform train – 1.5 MN;
3) at controlled braking from a motion speed over 15 km/h:
   - in a non-uniform train – 1.5 MN;
   - in a uniform train – 1.2 MN.

To ensure the strength of load-bearing structures in the rolling stock, it is important to study the longitudinal loading and devise measures to ensure traffic safety.

5. Modeling the longitudinal loading on a train equipped with new conceptual couplers

In order to reduce the longitudinal-dynamic efforts in a train under operating modes, including braking, it is proposed to use, instead of a standard automatic coupling device, a conceptual coupler (Fig. 3). In this case, the impact’s kinetic energy is damped by transforming it into the work of viscous resistance forces. This resistance is created by moving a viscous liquid through the throttle holes of the piston based on the principle of hydraulic damper operation. The system returns to its original state by using a release spring, which is mounted inside a telescopic element [16].

It is important to note that such a conceptual coupler can be implemented in the rolling stock whose girder beams have a closed cross-section. For example, such a technical solution could be used on cars whose load-bearing elements are made from round pipes.

![Fig. 3. A conceptual coupler for the automatic coupling: a – structural components; b – in the interaction between cars; 1 – automatic coupling body; 2 – adapter; 3 – wedge; 4 – girder beam, made from round-section pipe; 5 – bottom; 6 – spring; 7 – telescopic element](image-url)
the strength of the coupling device through the imaginary separation of a train into two parts. The calculation scheme is shown in Fig. 2.

Suppose that it is required, for the train consisting of 40 cars of model 12-7023, to determine the magnitude and a sign of the reaction between a locomotive and the cars. The locomotive’s accepted model is 2TE10V. The speed of the train is 60 km/h.

On the basis of Newton’s Law III, the reaction \( R \) can be represented in the form of two forces with different signs.

Then the motion equation for the left side of the train takes the form:

\[
M_{gx} \ddot{x} = \sum K_i \varphi_i - N - \sum \beta \cdot \dot{x},
\]

(7)

for the right part:

\[
M_{gx} \ddot{x} = \sum K_i \varphi_i + N - \sum \beta \cdot \dot{x},
\]

(8)

where \( \sum K_i, \sum K_2 \) are, accordingly, the sum of the forces of pressing the brake pads in the left and right parts of the train, \( kN; \beta \) is the coefficient of viscous resistance, which is created by the conceptual coupler, \( kN \cdot s/m \).

The sum of forces of pressing the brake pads can be found from formula [11]:

\[
\sum K = M_{gx} \cdot g \cdot \delta,
\]

(9)

where \( \delta \) is the normative value for a coefficient of brake pressing force.

The differential equations (7), (8) were solved by the Runge-Kutta method in the Mathcad programming suite [17–20].

For the left part of the train:

\[
\begin{aligned}
Q_1 &= \begin{bmatrix}
\dot{x} \\
\sum K_i \varphi_i - N - \sum \beta \cdot \dot{x}
\end{bmatrix}, \\
X_1 &= \text{rkfixed}(Y0, tn, tk, n', Q_1).
\end{aligned}
\]

(10)

For the right part of the train:

\[
\begin{aligned}
Q_2 &= \begin{bmatrix}
\dot{x} \\
\sum K_i \varphi_i + N - \sum \beta \cdot \dot{x}
\end{bmatrix}, \\
X_2 &= \text{rkfixed}(Y0, tn, tk, n', Q_2).
\end{aligned}
\]

(11)

The transition from the systems of second-order differential equations to the first-order differential equations was performed in order to apply the standard algorithms for system solving using the Mathcad rkfixed function.

To solve equations (7) and (8), we used the standard function \( \text{rkfixed}(Y0, tn, tk, n', Q) \). The \( Y0 \) vector contains the initial conditions (12). The magnitudes \( tn \) and \( tk \) determine the starting and final variable of the integration, \( n' \) is the fixed number of steps, \( Q \) is a symbol vector [21–25]. The models (7) and (8) do not take into consideration the rigidity of release springs, which would also resist the movement of the pistons (Fig. 3).

The estimated scheme of a rod will be considered in the form of a pivot system, as a beam with a cantilever clamping (Fig. 5). It is taken into account that the rod receives the load from the piston. A second piston is considered in the form of a rigid clamping. That is, the initial displacement and speed are equal to zero.

The sum of the forces of pressing the brake pads can be found from formula [11]:

\[
\sum K = M_{gx} \cdot g \cdot \delta,
\]

(13)

where \( \delta \) is the normative value for a coefficient of brake pressing force.

Under a standard scheme of interaction between a locomotive and the cars, the acceleration operating on the cars was about 1.2 m/s² (Fig. 4). Taking into consideration the coefficient of viscous resistance, created by the conceptual coupler, the acceleration was about 0.8 m/s². The results were derived for the estimated scheme shown in Fig. 2 and under the input parameters specified above.

In other words, the use of a conceptual coupler makes it possible to reduce the longitudinal loading on a train by almost 30 % compared to the interaction scheme involving the automatic coupling devices SA-3.

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### 6. Calculation of the coupling rod strength

Given that the rod under operating modes perceives significant loads, we calculate its strength.

It should be noted that a given conceptual coupler is proposed instead of the automatic coupling SA-3, whose main component is the absorbing device placed in a traction clamp. It is known that the absorbing device is designed to perceive and mitigate the impacts and jerks in the transmission of compressing and stretching efforts through a car frame (the longitudinal loads) [26]. Therefore, when calculating the rod of the proposed device, we considered the longitudinal loading that occurs under operational modes.

The estimated scheme of a rod will be considered in the form of a pivot system, as a beam with a cantilever clamping (Fig. 5). It is taken into account that the rod receives the load from the piston. A second piston is considered in the form of a rigid clamping. That is, the limitation of a given problem is the absence of piston movement when the rod is loaded. Such a scheme may occur in the case when the rigidity of the release spring of the device exceeds the magnitude of the longitudinal load acting on the rod. In addition, this...
scheme is valid for the case when the adapter is in an extreme position (in the region of the bottom).

In this case, the maximum stress that operates on the rod would be determined as the sum of stresses arising from effort \( P_1 \) (the compression deformation) and \( P_2 \) (the bending deformation) [27–29].

The diagram of the transverse force at compression deformation is shown in Fig. 6, \( a \); and the diagram of the bending moment at bending deformation is shown in Fig. 6, \( b \).

![](image)

Then, the maximum stresses operating on the rod will be determined from:

\[
\sigma_{\text{max}} = \frac{P_1}{A} + \frac{P_2}{W} \cos \alpha, \tag{14}
\]

where \( A \) is the cross-sectional area of the rod, m\(^2\); \( l \) is the rod length, m; \( W \) is the resistance momentum of the rod cross-section, m\(^2\).

Fig. 5. The rod estimation scheme under the action of angular loads on it

Fig. 6. Results of calculating the rod of a conceptual coupler: 
\( a \) — diagram of transverse force; 
\( b \) — diagram of bending momentum

Since the component \( P_1 \) determines resistance against the longitudinal bending, we checked the following:

\[
\sigma_{\text{max}} = \frac{P_1}{A} \leq \sigma_n, \tag{15}
\]

where \( \sigma_n \) is the critical load, MPa; \( \sigma_n \) is the resistance reserve.

The piston with the rod was made from steel of brand 09G2S. The value of the yield strength is \( \sigma_y = 345 \text{ MPa} \) and durability limit is \( \sigma_e = 490 \text{ MPa} \).

It was established that at value \( P_1 = 1.2 \text{ MN} \), the rod diameter of 80 mm, the value \( \sigma_y = 345 \text{ MPa} \) and \( n = 1 \), the condition (15) is met.

The rod strength condition in this case takes the form [30, 31]:

\[
\sigma_{\text{max}} \leq \sigma, \tag{16}
\]

where \( \sigma \) are the permissible stresses, MPa.

The following formula is used to determine the transverse angle of the inclination of the automatic coupling axis from the car axle.

\[
\alpha = \beta + \gamma, \tag{17}
\]

where

\[
\beta = \arctg \frac{\ell + c_1 + a_1}{R}, \tag{18}
\]

\[
\beta' = \arctg \frac{\ell' + c'_1 + a'_1}{R}, \tag{19}
\]

\[
\gamma = \arcsin \frac{b + b' + \xi}{2a}, \tag{20}
\]

where \( \ell, \ell' \) are the semi-bases of the coupled cars, m; \( c_1, c'_1 \) is the length of the consoles of the coupled cars from the axis of the hoop to the hinge axis of the automatic coupling shank, m; \( a_1 \) is the length of the automatic coupling body from the center of the hinge shank to the coupling axis, m.

\[
b = \frac{(2\ell + c_1) - c_1^2 - a_1^2}{2R}, \tag{21}
\]

\[
b' = \frac{(2\ell' + c'_1) - (c'_1)^2 - a'_1^2}{2R}, \tag{22}
\]

where \( 2\ell_1, 2\ell'_1 \) are the bases of bogies of the coupled cars, m; \( \xi \) is the additional reciprocal deviation of the automatic coupling hinges in the transverse direction, mm, determined according to [30, 31]; \( R \) is the estimated curve radius, m.

It is taken into consideration that a car fits the curve with a radius of \( R \geq 60 \text{ m} \). In this case, we derived the value of \( \beta = 6.9 \text{ degrees} \) and \( \gamma = 12.7 \text{ degrees} \). Hence, \( \alpha = 19.6 \text{ degrees} \).

Taking into consideration the above calculations, we obtained \( \sigma_{\text{max}} = 308.1 \text{ MPa} \) (a rod diameter of 80 mm and its length is 360 mm), that is the resulting stresses are smaller than those acceptable by 10 %, which are indicated for the operating conditions of cars on various railroad networks for cars’ metallic structures [30–32]. Consequently, the strength of the rod is ensured.

We also determined the strength of the rod of our conceptual coupler using a finite element method. The spatial model of the adapter was constructed, whose part is the rod (Fig. 7, \( a \)). The graphical study was carried out in the SolidWorks software package. The calculation was implemented in the CosmosWorks programming environment. The finite element model of the adapter is shown in Fig. 7, \( b \).

![](image)

The isoparametric tetrahedrons were used as finite elements. The number of nodes of the finite element model was 3,792, elements – 15,554. The maximum size of the element in the model is 30.8 mm, the minimum is 6.16 mm. The percentage of elements with a side ratio of less than three is 96.9, exceeding ten – 0.03.

The load applied to the adapter was taken equal to \( N = 1.2 \text{ MN} \) (Fig. 7, \( b \)). The structure’s material is steel, grade 09G2S. The calculation results are shown in Fig. 8.
It was established that the use of the conceptual coupler makes it possible to reduce the longitudinal loading on a train by almost 30% compared to the interaction scheme involving the automatic coupling devices SA-3 (Fig. 4). This is explained by the fact that the proposed concept generates a viscous resistance to the movement, rather than the elastic-friction as is the case with the automatic coupling SA-3.

The proposed technical solutions would reduce the dynamic loading of cars under load operating modes. This could decrease the amount of damage to cars in operation. In addition, our research results would contribute to the construction of modern rolling stock structures.

To ensure the rod strength of the coupling, we performed the calculation described in the main part of this paper. This takes into consideration the longitudinal load, which acts on the rod (Fig. 5), as the principal one. It was determined that given the accepted geometric dimensions and numerical values of loads, the rod strength is provided (Fig. 8).

It is important to note that modeling the longitudinal dynamics and rod strength of the coupling device was carried out for the case of steady train motion and taking into consideration that the components of the coupling device were made from steel of grade 09G2S. It was taken into consideration that the longitudinal load that a coupling is exposed to is 1.2 MN, which is valid for a uniform train.

In the future, it is necessary to take into consideration the case when a train moves along a broken profile, as well as the transient processes in a train that pulls off and under the action of impact loads. As regards further studies, it also important to determine the influence of the longitudinal dynamics of tank cars with bulk cargo on the operation of a coupling device.

9. Conclusions

1. A mathematical model has been constructed to determine the longitudinal load on a railroad train equipped with the new conceptual couplers. We have determined the maximum loads operating on a train under operational modes. A given model could be used to model the longitudinal dynamics of non-uniform freight trains.

It was established that the proposed device makes it possible to reduce the longitudinal load on a train by almost 30% compared to the interaction scheme involving the automatic coupling devices SA-3.

2. The rod strength of the coupling has been estimated. The rod estimation scheme was adopted in the form of a pivot system. In this case, the maximum stress, which acts on the rod, was determined by taking into consideration the effect exerted on it by the longitudinal load of 1.2 MN.
Taking into consideration our calculations, at the angle of the lateral deviation of the longitudinal axis of the automatic coupling from the axis of the car $\alpha = 19.6$ degrees, we obtained $\sigma_{\text{max}} = 308.1$ MPa. That is, the resulting stresses are smaller than the permissible ones by 10%. It was taken into consideration that the structure's material was the steel of grade 09G2S.

We have also determined the strength of the coupling device's rod using the method of finite elements. It was taken into account that the adapter, whose part is the rod, is made of steel of grade 09G2S. The maximum equivalent stresses in the adapter were 269.2 MPa, the displacement is 0.5 mm. The maximum deformations are $1.16 \times 10^{-3}$.

Acknowledgments

The present study was conducted within the framework of the scientific topic of young scientists “Innovative principles of creation of resource-saving structures of railroad cars by taking into account the refined dynamic loads and functional-adaptive flash concepts”, which is funded by the state budget of Ukraine in 2020.

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The propeller, as a thruster, has been known for a very long time and has been the only possible mover in aviation for many years. The increase in flight speed, the emergence of jet, and then turbofan gas turbine engines significantly reduced the use of propellers. However, it should be noted that today the propeller has no competitors in terms of profitability at moderate flight speeds ($M \leq 0.6$), and in the case of coaxial counter-rotating fans, $M \leq 0.8$.

**A NUMERICAL STUDY OF PERFORMANCE OF THE SMALL-SIZE UAV PUSHING TANDEM PROPELLER WITH JOINED BLADES**

M. Kulyk
Doctor of Technical Sciences, Professor, Head of Department

F. Kirchu
PhD, Associate Professor
E-mail: fkirchu@gmail.com

Hussein Hanesh
Postgraduate Student*

**UDC 629.7.035.7**

DOI: 10.15587/1729-4061.2020.199486

**1. Introduction**

The propeller, as a thruster, has been known for a very long time and has been the only possible mover in aviation for many years. The increase in flight speed, the emergence of jet, and then turbofan gas turbine engines significantly reduced the use of propellers. However, it should be noted that today the propeller has no competitors in terms of profitability at moderate flight speeds ($M \leq 0.6$), and in the case of coaxial counter-rotating fans, $M \leq 0.8$.

Also, propellers...