THE STRENGTH CALCULATION OF THE MODERNIZED BRAKE LEVER TRANSMISSION FOR A WAGON BOGIE

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Resume

The article highlights is determination of the main strength indicators of the modernized brake lever transmission for a wagon bogie. The special feature of this modernization is that the technological pin-hole in the strut is displaced by 112 ± 2 mm towards the bolster beam, i.e., in line with the pendulum joints. This helps to eliminate abnormal wear of the brake pads in operation. To substantiate the proposed modernization, the strength of the brake lever transmission was calculated. It was found that the maximum equivalent stresses were 124 MPa; they occurred in the tension member. The resulting stresses did not exceed the permissible values. Thus, the proposed modernization measures are appropriate.
This research could improve the safety of rail freight operations and profitability of the rail transport.

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1 Introduction

In today's competitive environment, sustainable development of the railway industry is aimed at the modernization of key elements of technical infrastructure [1-5]. Particular attention should be paid to the mechanical parts of braking systems of freight trains, which play a critical role in ensuring the traffic safety.

The braking process of a freight train is a complex multidisciplinary task, involving the interaction of various components and systems, and aimed at optimal deceleration and stopping of the rolling stock. The efficiency of braking systems and their reliable operation are crucial for reducing the risks during the transportation of goods, taking into account the high requirements for safety and stability of rail transport. Therefore, current trends in the railway transport industry include stricter requirements for braking systems, a need for their continuous modernization and improvement.

The imperfect design of the brake lever transmission (BLT) of wagon bogies has a significant impact on the braking efficiency. In the traction and idling modes, the brake pads are tilted and touch the rolling surfaces of the wheels with their upper ends. Under these conditions, unwanted friction is created, thus causing abnormal pad wear. This not only leads to an increase in operating costs for the wagon maintenance, but to an increase in traction resistance as well, which consumes additional energy resources.

Moreover, the absence of a clearance between the pad and the wheel in the upper part (Figure 1, a) leads to abrasion and a decrease in the working area, which reduces the train braking efficiency. Under this condition, the temperature of the brake pad/wheel tribotechnical surfaces increases, and this causes high-temperature damage to the wheels (Figure 1, b). Such damage can include microcracks, shelled treads and metal dislocations, which significantly shorten the service life of wheelsets and increase the cost of maintenance and replacement.

Thus, there is a need to modernize the lever transmission design of wagon brake systems to improve safety and optimize costs in the railway industry.

An analysis of scientific, technical, and advertising sources on the BLT performance of railcars has confirmed that the existing designs of brake systems

a) b)

***Figure 1** The elements of the brake pad/wheel tribotechnical assembly that are inoperable due to the BLT design features; a) without a clearance between the pad and a wheel when the brakes are released; b) high-temperature damage on the wheel surface*

of bogies described in [6-8] do not completely solve the problem of abnormal pad wear during the brake release. It is established that the developers of these systems mostly focused on auxiliary devices to counteract the forces, that affect the position of the pads according to the kinetostatic analysis of the BLT mechanism [9]. This indicates that an integrated approach is needed when designing and optimizing brake systems. The dynamics and the real operational aspects of their functioning should also be taken into account, as well as the forces that occur in the elements of the bogie's brake beams during braking.

In study [10] are analyzed the operational quality indicators of the cast iron brake pads and composite ones used for passenger and freight rolling stock. The authors considered a number of negative aspects of composite pads, in particular, their impact on the environment. In addition, the processes that cause damage to vehicle wheels due to the mechanical properties of the materials used for the pads are described. In this study the aim was to identify the cause-and-effect relationships between different types of brake pads. The study results show the impact of pad wear on environmental pollution, durability, and safety of railway transport.

In many scientific studies that were focused on the use of composite brake pads in the rolling stock the aspects related to traffic safety and their impact on the environment were emphasized. To reduce the operating costs in the railway industry, brake pads are often perceived as a commodity purchased at the lowest price while performing satisfactorily [11]. However, this approach may not ensure the lowest operating costs in the long run. The choice of material of the friction components of the pads has a direct impact on duration of the standard service life of the wheelset, the replacement of which usually significantly exceeds the cost of the BLT elements of the wagon. Thus, the economic benefit of the low initial price of pads can be considered unjustified due to the increased costs of further maintenance and repair of rolling stock.

Authors of study [12] presented the comparative indicators of the quality and performance characteristics of the cast iron brake pads as well as the composite ones. They highlighted several disadvantages of composite pads, in particular, their low thermal conductivity, which causes the thermal damage to the rolling surfaces of wheels. This, in turn, leads to an increase in operating costs associated with the repair of wheelsets. Another significant drawback is that the regulatory documents for manufacturing do not have the chemical composition of the composite mixture ingredients. This violates the current legislation of Ukraine and makes it difficult to control it. However, the authors do not mention the costs caused by the abnormal wear of composite brake pads that occurs when freight rolling stock operates without active braking.

In study [13] is presented an analysis of various braking systems, in particular those used in pad/wheel tribotechnical pairs. The authors presented the advantages of disc brakes and the feasibility of their use with conventional tribotechnical pairs. The main problem with composite brake pads used in wagons is their negative impact on the rolling surface of the wheelsets. Due to the low thermal conductivity of the pads, high temperatures occur in the contact zone with the wheels, which causes various defects on the wheel surface.

In study [14] was indicated that most international scientists focus on solving issues related to disc brakes. They analyse the strength of their parts, monitor operating parameters, and determine the temperature conditions of individual tribotechnical components of rolling stock brake systems. This indicates a global trend towards optimizing the brake systems to increase their efficiency and reliability, reduce wear, and improve the overall performance of rolling stock.

Authors of studies [15-16] have dealt with the use of modern materials for the development of new designs of tribotechnical systems. The effectiveness of these innovations in the context of modern rolling stock has been proven. Thanks to these innovations, it is possible to increase the speed and axle load, as well as improve the efficiency of the braking system of rolling stock. However, the above studies did not pay due attention to the problems associated with abnormal wear of brake pads in freight rolling stock. Therefore, it would be advisable to analyse the factors influencing the performance of tribotechnical systems in order to increase their wear resistance and take into account modern materials used for them.

The analysis of publications [6-16] has made it possible to conclude that the problem of abnormal wear of pads used in the brake systems of railcar bogies is very urgent. The solution of it is important for ensuring the reliability and safety of the rail transport, thus further research and development is necessary.

The objective of this study was to substantiate the feasibility of using a modernized brake lever transmission on the wagon bogie 18-100.

The following tasks have been set to achieve this objective:

- to develop measures for eliminating abnormal pad wear by upgrading the BLT elements; and
- to calculate the strength of the upgraded BLT of a wagon bogie 18-100.

2 Materials and methods

In their previous works the authors [17-19] established that the main reason for abnormal wear of brake pads is the inconsistency of the force factors acting in the break lever transmission of the bogie. It can be explained by the design of the brake system on the pendulum suspension, and the tribotechnical processes that occur between the pad and the wheel when the train is moving with the brakes released.

To eliminate the harmful torque to the brake lever transmission, it is proposed to relocate the joint of the vertical arm and the brake beam strut. Based on the measurements of a typical brake beam structure, it was determined that the pin-hole in the strut should be located at a distance of 112 mm (instead of the existing 224 mm) from the outer end of the brake beam. This rather simple, but theoretically justified structural change in the location of the pin-hole in the brake beam strut can solve the problem of abnormal wear of the pads (Figure 2).

This design change makes it possible to completely eliminate the harmful torque on the brake lever transmission and the modernized brake beam will restrain the brake pads from resting on the rolling surfaces of the wheelset.

Given that the modernized brake beam is held loosely on the pendulum suspension joints by friction, it is likely that intense vibrations of the wagon's running gears may periodically cause the brake beam to tilt in one direction or another. In this case, the brake pads may tilt and touch the rolling surface of the wheels with their upper or lower parts when the brake is released.

Figure 2 *Modernized brake beam with the pin-hole in the strut moved in line with the pendulum joints*

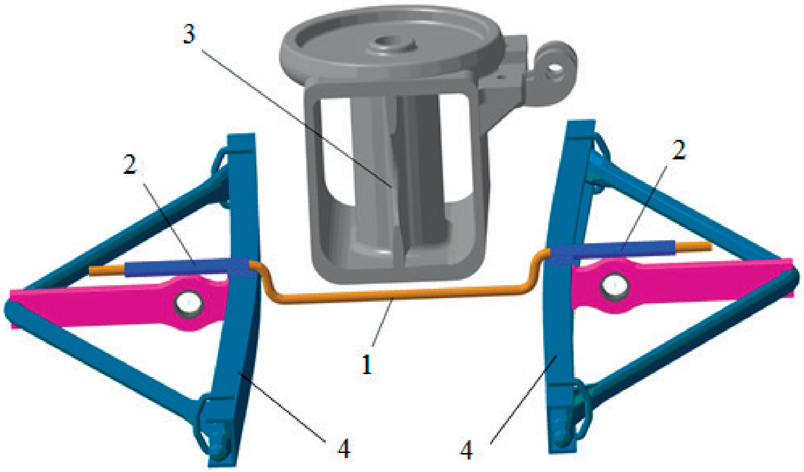


Figure 3 *The general view of the device for maintaining uniform clearances between the brake pads and the rolling surfaces of the wheelset*
1 - curved rod; 2 - cylindrical slides; 3 - bolster beam;
4 - modernized brake beams with a displaced pin-hole in the strut

To prevent the brake pads from tilting up to their partial contact with the rolling surface of the wheel, a device (Figure 3) was developed to absorb minor accidental forces from oscillations and tilts of the bogie when the train moves. This will ensure that the clearances between the pads and wheels are kept exactly even. This device has a curved guide rod 1 between the pair of modernised brake beam 4 in their middle part. The ends of this rod fit into the cylindrical slides 2, welded to the brake beam symmetrically with respect to the opening of its strut. The downward bending of the rod should be no less than the maximum possible displacement of the bolster beam 3 due to the loading of the railcar. The parts of the rod that are curved downward are located near the ends of the cylindrical slides [17-20].

When the random forces are applied to the brake beams due to intense oscillations and tilts of the wagon bogie when the train moves, the curved rod in the slides works. The forces that cause the brake beam to tilt relative to the pendulum suspensions are counteracted by the reactive forces generated between the rod and the slides. This ensures that the brake beams are constantly in balance and maintain uniform clearances between the pads and wheels when the brake is released. The bending of the rod provides space for the bolster to move downward due to the load of the wagon and simultaneously keeps it from longitudinal displacement and falling out.

3 Results and discussion

To determine the strength of the modernized brake beam with a displaced pin-hole in the strut, the corresponding calculations were performed. Due to bending moments at the end parts of the brake beam caused by the eccentric application of load at the joint of the tension member and the brake beam, the calculation was performed by the refined method including the bending deformation of the beam. The brake beam was calculated using the force method according to the diagram shown in Figure 4. The force in the tension member was taken as an extra unknown variable, i.e., the system was once statically indeterminate. Then, the canonical equation of the method of forces is as follows:

$$X_1\delta_{11} + \Delta_{1p} = 0\,, \tag{1}$$

where δ_{11} is a single displacement in the direction of the i-th link caused by the force X_1; Δ_{1F} is a displacement in the direction of the i-th link caused by the simultaneous action of the entire external load.

The basic brake beam system is shown in Figure 4. The bending moments and longitudinal forces from the force $X = 1$, as well as the load diagram, are shown in Figure 5.

Determining the displacement:

a) from the force $X_1 = 1$

$$E\delta_{11} = \frac{2l_1 a^2 \sin^2\beta}{3I_y} + \frac{2l_1\cos^2(\alpha+\beta)}{F_b} + \frac{2l_2}{F_c} + \\ + \frac{4\cos^2\gamma\cdot n}{F_p}\,, \tag{2}$$

where I_y is the moment of inertia of the beam relative to the vertical axis; F_b is the cross-section of the beam; F_p is the cross-section of the strut; F_c is the cross-section of the tension member; E is modulus of elasticity of the triangular component (the triangle); a is a half-length of the triangle the end joints; l_1 is length of the triangle between end joint and middle joint; l_2 is a string length; α is a tilting angle of the triangle; β is a tilting angle of the string; γ is angle formed by the spacer and the string; n is a distance from the center hinge to the point

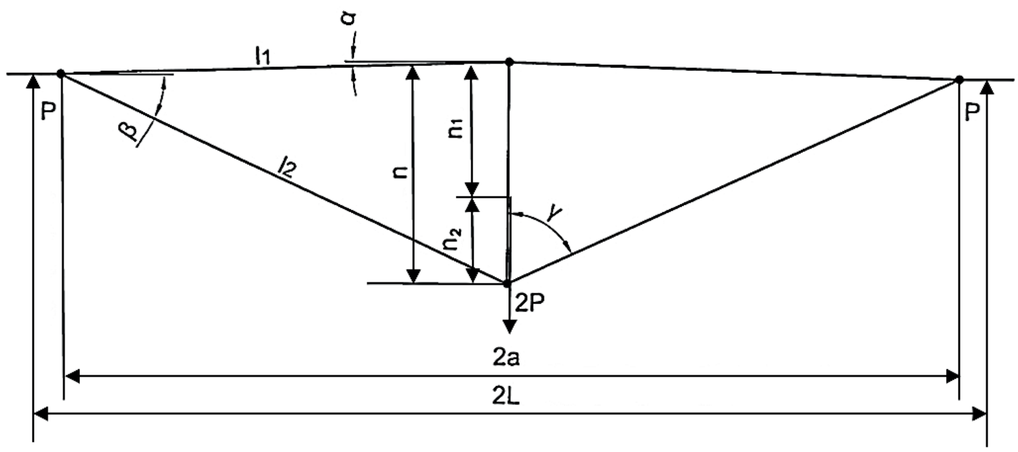


Figure 4 *The design diagram of the brake beam*

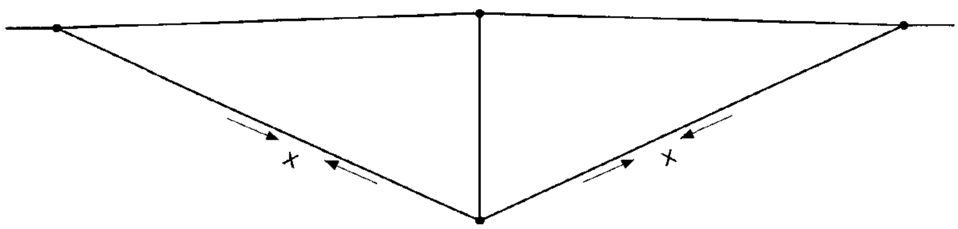


Figure 5 *The basic system of the brake beam*

of interaction of the string arms; n_1 and n_2 are segments forming the distance from the middle hinge to the point of interaction of the string arms; $2L$ is the triangle length; $2P$ is the force acting on the triangle.

b) from the load 2 P in the direction of the extra unknown value

$$E\Delta_{1p} = -\frac{a\sin\beta \cdot l_1 P}{3I_y}(3L-a) - \frac{4Ph_1\cos\gamma}{F_p}. \quad (3)$$

The force X_1 (Figure 4) is determined from Equation (1)

$$X_1 = -\frac{\Delta_{1p}}{\delta_{11}}. \quad (4)$$

The values included in calculation Equations (2) and (3) were taken from the design drawings of a typical brake beam:

$2L$ = 1,607 mm;	h = 375.3 mm;	$I_y = 20.9 \cdot 10^4$mm^4;
2 a = 1.517 mm;	h_1 = 208 mm;	F_b = 1185 mm^2;
l_1 = 758.9 mm;	cos α = 0.999	F_c = 855 mm^2;
l_2 = 845 mm;	cos γ = 0.444	F_p = 1500 mm^2.

Then, based on the calculations, it was obtained:

$$E\delta_{11} = 2777.13,$$

$$E\Delta_{1p} = -6736.01P.$$

From here, follows:

$$X_1 = \frac{6736.01P}{2777.13} \cong 2.43P.$$

After determining the force X_1, the total bending moment diagram was constructed by superimposing the diagrams shown in Figure 6. In this case, the first of them is multiplied by the resulting value X_1 before o superimposing.

Based on the total bending moment diagram (Figure 7), the stresses in the brake beam elements were determined.

In the calculations it was assumed that the maximum load on the brake beam was $2P$ = 77 kN.

The tensile stress in the tension member of the brake beam is

$$\sigma = \frac{2.43P}{F_c} = \frac{2.43 \cdot 3850\mathrm{N}}{855\,\mathrm{mm}^2} = 11\,\mathrm{MPa}. \quad (5)$$

Determining the stress in the brake beam (at the end of the tension member reinforcement, i.e., at a distance of 245 mm from the support).

The slenderness of the beam was determined from the expression

$$\lambda = \frac{l_1}{\sqrt{\frac{I_{\min}}{F}}} = \frac{75.9}{\sqrt{\frac{20.9}{11.85}}} = 56.5[-]. \quad (6)$$

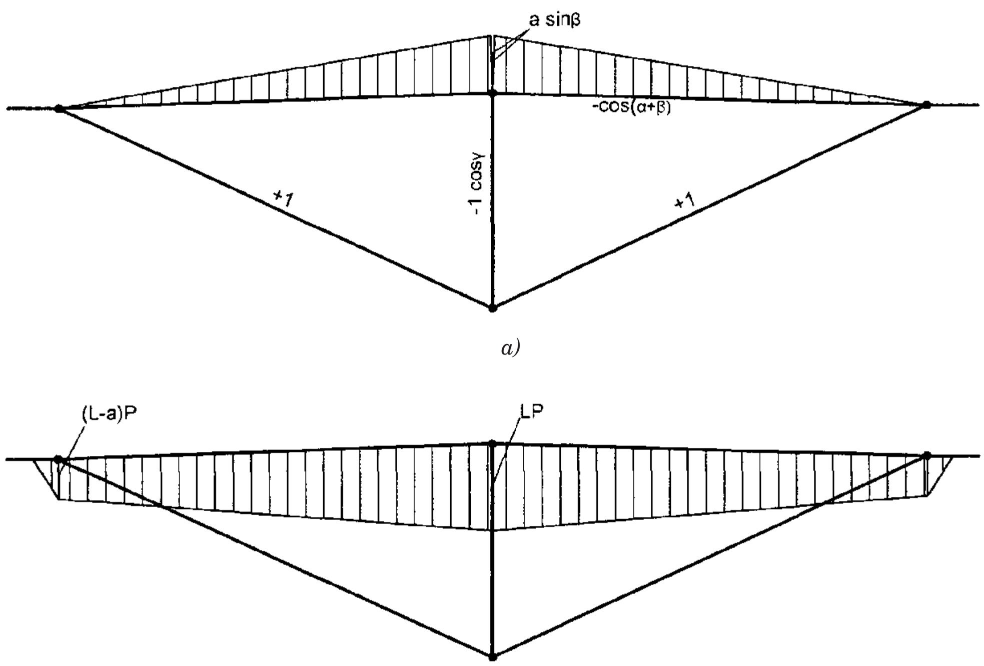
The coefficient of longitudinal bending with the resulting slenderness is equal to φ = 0.87.

The stress in the rod is

$$\sigma = \frac{2.7 \cdot 3850\mathrm{N}}{1375\mathrm{mm}^2} + \frac{2.14 \cdot 3850\mathrm{N}}{1185\mathrm{mm}^2 \cdot 0.87} = 15.7\,\mathrm{MPa}.$$

The results of the analytical calculations showed that the proposed modernization did not change the stresses in the dangerous areas of the brake beam and, therefore, did not affect its strength.

To determine the stress distribution fields in the modernized brake lever transmission, a strength calculation was performed using the finite element method, implemented in SolidWorks Simulation [21-23]. The graphic works on creation of the brake lever transmission was reproduced in SolidWorks. Figure 8 shows a graphical model of the brake lever transmission. Since the calculations were performed as quasi-static,



a)

b)

Figure 6 *The bending moment diagrams, a) from the single force X_1; b) from the external load 2P*

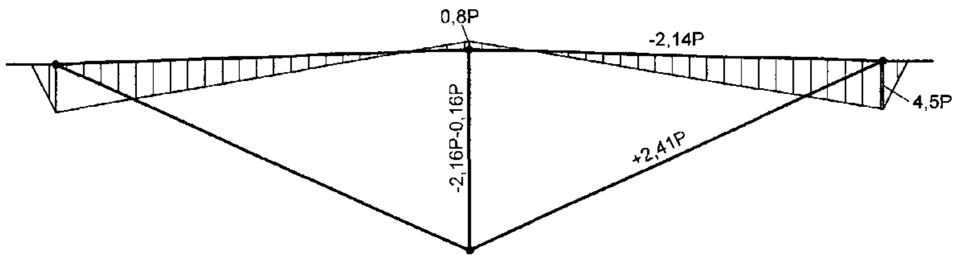


Figure 7 *The total bending moment diagram*

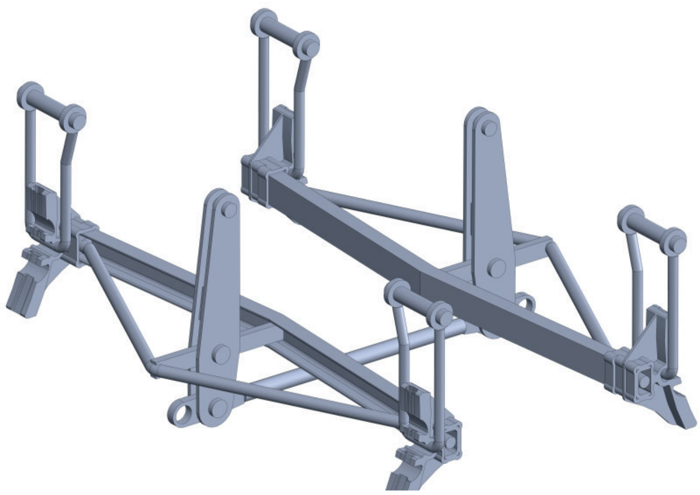


Figure 8 *The graphical model of the brake lever transmission*

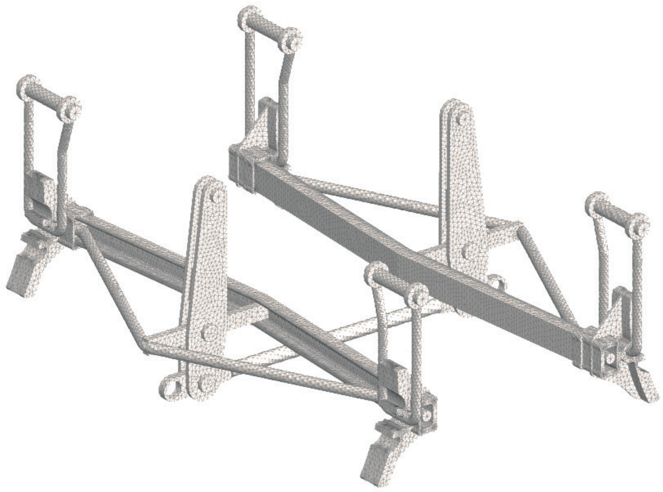


Figure 9 *The finite element model of the brake lever transmission*

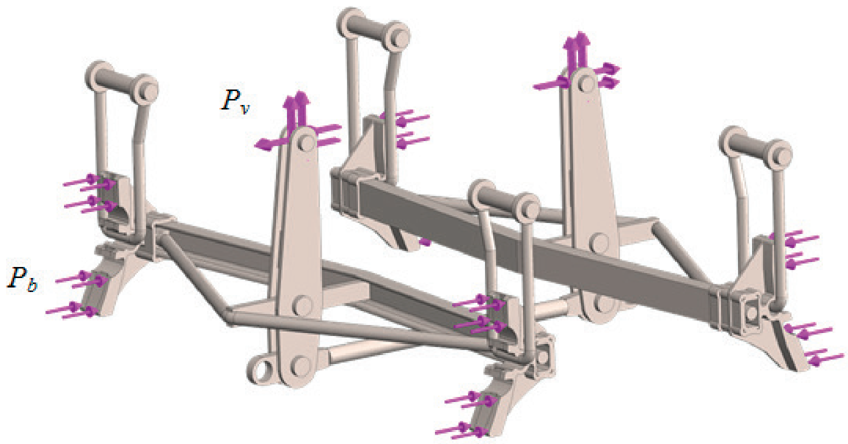


Figure 10 *The design diagram of the brake lever transmission*

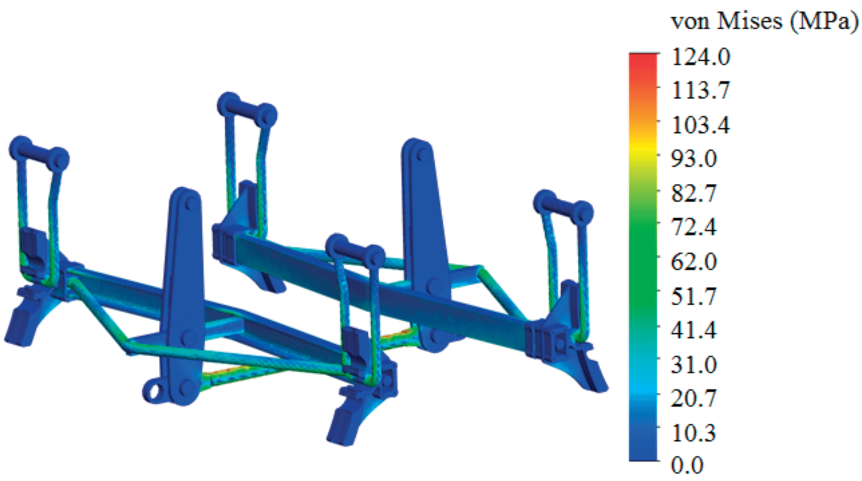


Figure 11 *The stressed state of the brake lever transmission*

the guide device was not taken into account.

The finite element model was formed by tetrahedra (Figure 9).

Their number was calculated graphically and analytically [24]. The grid had 288,115 elements and 76,296 nodes. The model was fixed by the suspension. The load was applied to the shoes P_b, as well as to the joints of the vertical arms P_v, which was decomposed into two elements taking into account the angle of application (Figure 10).

Based on the calculations performed with the von Mises criterion, the stresses in the elements of the brake lever transmission were obtained (Figure 11).

The maximum stresses, recorded in the tensile member, were 124 MPa. Those stresses are lower than the permissible value [25], so the strength of the brake transmission is ensured. Thus, the solutions proposed in the study are appropriate.

The scientific novelty of this research is the included derived model of the strength of the brake lever transmission of the freight wagon bogie, which allows to determine the main indicators of the strength of its components under various external loads. This model can be used within the framework of the educational process of students studying in the discipline "Autobrakes of rolling stock". This model can also be used in the conditions of wagon-building enterprises when designing and modernizing the brake lever transmissions of modern rolling stock.

4 Conclusions

1. The measures aimed at eliminating abnormal pad wear by modernizing the brake lever transmission elements are proposed. The technological pin-hole in the strut is displaced by 112 ±2mm towards the bolster beam, i.e., in line with the joints of the pendulum suspensions. In addition, it is recommended to use the innovative BLT guide device, which ensures uniform clearances between the pads and wheels. Such measures to modernize the brake lever transmission elements help to balance the weight of the parts relative to the pendulum suspension and neutralize the harmful torque to the brake beams.
2. The strength of the modernized brake lever transmission of the wagon bogie 18-100 is calculated. The maximum equivalent stresses recorded in the tension member of the brake beam are 124 MPa, which is lower than the permissible value. The research shows that the proposed modernization measures are appropriate.

The research results could help in improving the safety of freight trains and increasing the profitability of the rail transport.

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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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