Study of the stress-strain state of the wheel pair of a freight car during braking

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Abstract. Objective: study of the stress-strain state of the wheel pair of a freight car in the process of braking. Methods: to determine the stress-strain state of the wheel pairs of a freight car. Results: A volumetric finite element model of a wheel pair with rail sections was created using a finite element of the ten knot tetrahedron type, and maximum shear stresses and maximum equivalent stresses were determined according to the Mises and Dang Wan theory. Practical significance: It is shown that the maximum tangential stresses are observed at a point located at a depth of 4.5-5.3 mm below the rolling surface of the wheel. In case of emergency (short) braking, maximum stresses occur on the rolling surface of the wheel. During prolonged braking (movement of the train along a prolonged descent), maximum stresses occur at the point of transition from the disk to the rim on the inner side of the wheel, and the magnitude of these stresses is 2.5 times higher than in the emergency braking mode.

1. Introduction

In the second half of the 20th century, a series of comprehensive measures were taken on the USSR railways to improve the structure of the rolling stock and the superstructure. This has led to a change of wear of the wheel-rail pair. However, to date there is no complete clarity in the physics of the process of interaction between the wheel and the rail. At the same time domestic experts emit more than 30 factors, and foreign (Canadian National Research Center) - more than 60 factors, which, in their opinion, affect the wear of the wheel and rail. These reasons include the totality of such factors as:

- narrowing the gauge to 1520 mm and changing the norms of the broadening of the gauge in curves;
- the establishment of the elevation of the outer rails in the curve in accordance with the maximum speeds;
  - increase the hardness of the rails;
  - an increase in static axle load;
  - increase the mass and length of the freight train;
  - transition of rolling stock to roller box bearings;
  - integration of new wheel profiles, etc.

There are three main modes of train movement: connection rod, slowing-down and braking. In the braking mode, when the pads are pressed against the wheel, there is a sharp heating of the rolling
surface of the wheel. The article conducted a study of the stress-strain state of the wheel pair of a freight car in the process of emergency (short) and long (service) braking.

2. Formulation of the problem
When modeling the “car-track” system, the vertical forces acting in the “wheel-rail” contact were determined, the load values from the maximum to the minimum are divided into 6 ranges. Forces in contact “wheel - rail” are determined in of period of time equal to 0.05 s. The values of forces, obtained during the simulation were divide over the ranges, as well as the calculation of the number fell into this range the ratio of the number of forces in this range to their total number gives the frequency of occurrence of force from the corresponding range, and then for each range is determined by the average value of the current load [1].

The values of the vertical loads acting in the contact “wheel - rail” and the frequency of their appearance for a freight car under various operating conditions, obtained as a result of numerical simulation of the “car-track” system are given in Table 1.

| № | Load | Value, kN | Frequency of occurrence | Max. contact stress, MPa |
|---|------|----------|-------------------------|-------------------------|
| P1 | 221,60 | 0,005 | 1341,5 |
| P2 | 192,96 | 0,040 | 1281,0 |
| P3 | 164,32 | 0,146 | 1214,2 |
| P4 | 135,68 | 0,556 | 1139,1 |
| P5 | 107,04 | 0,195 | 1052,6 |
| P6 | 78,40  | 0,058 | 948,8  |

According to the loads in Table 1, the stresses occurring in the contact zone between the wheel and the rail can be determined. To calculate stresses, the Hertz theory can be used [2, 3]. The disadvantage of Hertz's theory is that it considers deformations only in the contact zone, the real shape and dimensions of the contacting bodies are not taken into account. The contacting bodies in this theory are represented as elastic semi-infinite spaces. The assumptions determined by the theory of Hertz, are justified by the fact that the contact zone has dimensions much smaller than the dimensions of the contacting bodies [3].

3. Theory
When calculating, the stress-strain state in the contact of the wheel pair RU1-950 with the profile of the rolling surface according to GOST 10791-2011 [4] with the rails R65 GOST R 51685-2013 [5] was estimated.

Calculation of the wheel pair for thermal load was made by the finite element method [6]. The calculation is performed in two stages: at the first stage, the problem of determining temperature fields arising in the wheel during braking is solved, at the second stage, the stress-strain state is calculated using previously determined temperature fields. The created finite element model contains 261381 nodes and 172785 finite elements (Fig. 1). When solving the problem of determining the temperature field on the rolling surface, the heat flux was set equal to the power of friction forces acting on one wheel during braking (boundary conditions of the second kind), but the other surfaces of the wheel were set to exchange heat with the environment by heat transfer (boundary conditions of the third kind) determined by heat transfer coefficient 30 W / m2. The temperature of the wheel pair before the braking process was assumed to be equal to 0 °C.

The process of emergency (short) and long braking (movement on a long descent) for a freight car was considered. During emergency braking, it was assumed that the heat flux on the rolling surface is equal to the maximum allowable power per block during emergency braking, which according to the “Standards for calculating and designing cars” [7] when using composite pads is 70 kW. The time of
emergency braking of a freight car with a maximum allowable speed of 90 km / h [8] before the train stopped was assumed to be 60 s.

With prolonged braking, which may occur when the train is moving along a long descent, it was assumed that the braking power is 37 kW per wheel, the deceleration time is 25 minutes (1500 seconds). This mode is provided for by GOST 33783-2016 for determining the wheel strength indexes [9].

Figure 1. Finite element model created to study the strength of the wheels in the heating process when braking

4. Experimental results

The boundary conditions for solving the problem of determining the temperature fields arising during emergency (short) braking are shown in Figure 2. The temperature fields arising in a wheel pair at the end of emergency braking are shown in Figure 3, the corresponding equivalent stress fields are shown in Figure 4.

Figure 2. Boundary conditions during external braking
The boundary conditions for solving the problem of determining the temperature field arising during prolonged braking are shown in Figure 5. The temperature field arising in the wheel pair at the end of the long braking is shown in Figure 6.
Figure 6. The temperature field arising in the wheel pair at the end of long braking.

Figure 7. The distribution of equivalent stresses according to the Mises theory at the end of long braking (view from the inside).

Figure 8. The distribution of equivalent stresses according to the Mises theory at the end of long deceleration (outside view).
5. Discussion of results

For this calculation, a volumetric finite element model of a wheel pair was created using an isoparametric finite element of the ten knot tetrahedron type. Thanks to the quadratic functions of the form, this element makes it possible to well describe the stress state in the places of geometry change and accurately approximate complex curvilinear surfaces of the wheel. When determining the temperature fields, this element has one degree of freedom in the node (node temperature), and when solving the problem of determining the stress-strain state there are three degrees of freedom in the node (linear displacements along the X, Y, Z axes).

The finite element model of the wheel set created for the calculation (Fig. 1) assumes that the finite element mesh on the rolling surface and in places where the wheel geometry changes is thickened, which makes it possible to more accurately describe the stress-strain state. A larger mesh of finite elements is applied to the wheel axle and hub, which allows reducing the resources required for the calculation.

As can be seen from Figure 3, the maximum temperature during emergency braking occurs on the rolling surface and is 204.7 °C. From figure 4 it can be seen that during emergency (short) braking, the maximum equivalent stresses according to the Mises theory arise on the rolling surface, their value is 387.7 MPa, in other wheel parts there are no significant stresses under this mode.

From Figure 6 it follows that the maximum temperatures at the end of the continuous braking mode occur on the wheel rolling surface and are 473 °C, the maximum equivalent stresses according to the Mises theory in this mode occur at the point of transition from the disk to the rim from the inside of the wheel 5 MPa (Fig. 7 and Fig. 8) (the magnitude of these stresses is 2.5 times higher than with the emergency braking mode). Due to the high stress at the point of transition from the disk to the rim with this settlement mode, it can be argued that the long braking mode is much more dangerous than the emergency braking mode.

6. Conclusions

1. At short breakings, the maximum stresses occur on the wheel rolling surface; during long braking, the maximum equivalent stresses occur at the point of transition from the disk to the rim from the inside of the wheel.

2. The maximum temperature during emergency braking occurs on the rolling surface and is 204.7 °C. During emergency (short) braking, the maximum equivalent stresses according to the Mises theory arise on the rolling surface, their value is 387.7 MPa, in other parts of the wheel there are no significant stresses in this mode.

3. The maximum temperatures at the end of the continuous braking mode occur on the wheel rolling surface and are 473 °C, the maximum equivalent stresses according to the von Mises theory in this mode occur at the point of transition from the disk to the rim on the inner side of the wheel, the stress value reaches 936.5 MPa.

4. In case of emergency (short) braking, maximum stresses occur on the rolling surface of the wheel. During prolonged braking (movement of the train along a prolonged descent), maximum stresses occur at the point of transition from the disk to the rim on the inner side of the wheel, and the magnitude of these stresses is 2.5 times higher than in the emergency braking mode.

7. References

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