Stand for accelerated tests of rail vehicles wheelsets

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Abstract. Tests and research can be greatly accelerated by applying a multi-point load to the test wheel. The aim of this work is to substantiate rational parameters of the stand for accelerated life tests of railway wheelsets, in which a multi-point load applies to the rolling surface of the test wheel. On the basis of the laws of the theory of elasticity, we performed analytical calculations of the contact stresses created by the rolling bodies of the load mechanism of the test stand. Research has shown that the dependence of the contact stresses on the wheel rolling surface on the ball’s diameter is nonlinear. The decrease in stress is due to an increase in the contact area. We obtained the analytical and graphical dependencies, which make it possible to evaluate the influence of the parameters of the stand for accelerated life tests of railway wheelsets on contact stresses on the wheel rolling surface, taking into account the shape and dimensions of this surface. The authors have proposed a stand design for accelerated life tests of railway wheelsets. The use of a stand with the parameters justified in this work will reduce the test duration by 10-15 times.

1. Introduction
Wheelsets are one of the main assembly units that determine the reliability and safety of industrial, mine and railway transport. The technology of the manufacture of parts and assembly wheelsets continuously improved. In this regard, it becomes necessary to check the performance characteristics of wheelsets, such as geometric dimensions, stresses in the wheel-axle connection, static and dynamic strength, imbalance, wear resistance and electrical characteristics. Obtaining and registration of these parameters is performed out at special stands. Tolerances for geometric dimensions, electrical characteristics [1], values of cyclic loads, schemes for their application and test methods for axles are regulated by sectoral state documents [2, 3]. Axle fatigue tests are described in articles [4, 5]. The results of fatigue tests and the probability of destruction of the wheelsets axles, depending on the duration of operation, are due to many random factors [6].

2. Purpose and problem statement
The aim of this work is to substantiate rational parameters of the stand for accelerated life tests of railway wheelsets, in which a multi-point load applies to the rolling surface of the test wheel.

The interaction of the railway wheel with the rail forms a highest kinematic pair with an initial contact along the line. During operation, the wheel rolling surface is subjected to pulsating loads with large amplitude, which cause significant contact stresses. Over time, the wheels working surfaces begin to wear out and collapse; the wheels profile along the rolling circle changes. The greatest depth of the wheel surface wear is observed along the rolling circle placed on distance of 70 mm from the wheel side
surface (Figure 1). Such a feature of changing the wheels profile should be taken into account when designing a test bench.

The load on the wheelsets can be realized using the stand rollers, simulating an endless rail. This is the easiest way to select the method and place of load application to the wheelset. The disadvantage of this test method is their considerable duration and the need to use high pressure hydraulic equipment.

The initial profile of the wheel rolling surface has a tapered shape (Figure 2). This feature allows designing the load mechanism of stand resembling to a ball or roller bearing, in which the wheel working surface plays the role of the raceway of inner ring.

It should be established how the contact stresses on the rolling surface change depending on the shape and number of rolling elements of the test stand.

3. Designs of stands for wheelsets endurance test
The stand (Figure 3) with a pair of rollers simulating an infinite rail can be used to obtain the characteristics of endurance and wear resistance [7]. This stand contains two rollers 3 interacting with the wheels rolling surfaces, a device for rotation and braking of the wheelset and a loading device. The loading device includes independent oil stations 6 with hydraulic distributors, hydraulic cylinders with rods 5, at the ends of which wedge stops 4 are installed, resting on the wedge surfaces of the roller holders 3. The wheelset rotation device 2 includes axle boxes 1 and a coupling with an engine 11. The braking device consists of an oil station 15 with hydraulic distributors, a hydraulic cylinder 14 and brake shoes 13. The imitation of the operational misalignment of the axle box unit relative to the wheelset is carried out by a drive consisting of an engine 10, a gearbox 9 and a gear 8.
To test wheel steels for wear and contact strength, friction machine and small-sized samples made of the investigated wheel steels are used [8]. Analysis of the characteristics of small-sized samples using similarity criteria is much cheaper and easier. The test scheme assumes contact interaction of two rollers (Figure 4). One roller is cylindrical in wheel steel and the other is toroidal in carbide WC-Co. When assessing the wear, the Archald model was used, which assumes wear is proportional to the work of friction forces. This is a classic method, which gives an idea of the properties of materials; however, it does not take into account the peculiarities of the interaction between the wheel and the rail.

4. **Force calculation of the interaction of a wheelset and rolling bodies of the test bench**

The stand for accelerated tests of wheelsets proposed by the authors (Figure 5) contains an electric motor 1, a gear coupling 2, bearing assemblies 4 and a load mechanism 5. Wheels of the tested wheelset 3 play the role of inner bearing rings in units 4.
The calculation scheme of the stand for calculating the forces and contact stresses on the wheels rolling surfaces is similar to the calculation scheme of a ball bearing. We assume that the rolling bodies are of the same diameter and are equally loaded. For such a case, the dependence of the wheel normal reaction $R_N$ on the magnitude of the force $F$ developed by the load mechanism:

$$R_N = \frac{F}{n \cdot \sin \alpha}, \quad (1)$$

where $F$ is the axial force from the loading mechanism; $n$ is the number of rolling bodies; $\alpha$ is a half of the angle at the top of the generatrix of the wheel working surface.

Under the action of $R_N$ forces, contact stresses should be created not lower than the maximum stresses on contact surface of "wheel-rail" pair during the operation of railway rolling stock.

During research the interaction of a wheel with a rail, the authors of work [9] established the range of values of contact stresses on the rolling surface of railway freight cars wheels: 1300–1700 MPa. And with the appearance of trough wear, the stress values can reach 6000 MPa. These data are the starting point for determining the load mechanism characteristics and the rolling elements dimensions.

Contact areas of the rolling bodies and the wheel rolling surface are ellipses with semi-axes [10]:

$$a = \alpha \sqrt{\frac{3R_N (1 - \mu^2)}{\frac{1}{\rho_{11}} + \frac{1}{\rho_{12}} + \frac{1}{\rho_{21}} + \frac{1}{\rho_{22}}}}, \quad (2)$$

$$b = \beta \sqrt{\frac{3R_N (1 - \mu^2)}{\frac{1}{\rho_{11}} + \frac{1}{\rho_{12}} + \frac{1}{\rho_{21}} + \frac{1}{\rho_{22}}}}, \quad (3)$$

where $\rho_{11} = \rho_{12}$ are the radii of the balls; $\rho_{21} = 0.479$ m and $\rho_{22} = \infty$ are the radii of curvature of the wheel working surface; $\mu = 0.3$ is the Poisson's ratio; $E = 210$ GPa is the elastic modulus of steel; $\alpha$, $\beta$ are coefficients determined from the table as functions of the auxiliary angle $\psi$:

$$\cos \psi = \pm \sqrt{\frac{\left(\frac{1}{\rho_{11}} + \frac{1}{\rho_{12}}\right)^2 + \left(\frac{1}{\rho_{21}} + \frac{1}{\rho_{12}}\right)^2 + 2 \left(\frac{1}{\rho_{11}} + \frac{1}{\rho_{12}}\right) \left(\frac{1}{\rho_{21}} + \frac{1}{\rho_{22}}\right)}{\frac{1}{\rho_{11}} + \frac{1}{\rho_{12}} + \frac{1}{\rho_{21}} + \frac{1}{\rho_{22}}}} \cos 2\varphi, \quad (4)$$

where $\varphi = 0$ is the angle between the main planes of curvature of the ball and the wheel, in which there are smaller radii $\rho_{11}$ and $\rho_{21}$; the sign of the numerator is determined so that $\cos \psi > 0$. 

![Figure 5. Scheme of wheelset load.](imageurl)
The maximum contact stresses are determined by the formula:

$$\sigma_{\text{max}} = 1.5 \frac{R_N}{\pi ab}$$  \hspace{1cm} (5)$$

In order to determine the relationship of contact stresses, the wheel-ball interaction forces, the ball diameter, according to the formulas (2)-(5), variant calculations of contact stresses were performed. Studies have shown that the dependence of contact stresses on the wheel rolling surface on the ball diameter is nonlinear (Figure 6). The decrease in stress is due to an increase in the contact area.

![Figure 6. Dependence of contact stresses on the wheel rolling surface on the ball diameter and the load on the contact area.](image)

Having given the value of contact stresses, according to the graphs, you can select the balls diameter and determine the load on them. Analysis of the graphs allows us to recommend a rational range of balls diameter under load:

- 5 kN – from 110 to 160 mm;
- 10 kN – from 160 to 180 mm.

The radius of the arc along which the balls can contact each other is several millimeters less than the radius of the arc passing through the ball’s centers (Figure 7). The lengths of these arcs differ slightly from the ball diameter. Therefore, the length of these arcs can be approximately taken equal to the ball diameter.

The number of balls in the load mechanism of the stand, taking into account the size of the separator, can be determined by the formula:

$$n = k \pi \left( \frac{D}{d} + 1 \right)$$  \hspace{1cm} (6)$$

where $k$ is coefficient taking into account the dimensions required to install the separator (in the first approximation we take $k = 0.5$); $D$ is wheel rolling circle diameter (we take $D = 957$ mm for new railway wheel); $d$ is ball diameter.
Figure 7. Geometric characteristics of balls with a diameter of 50, 100 and 150 mm in the load mechanism of the stand.

The fractional part of the number of balls obtained from (5) is removed, so the graph of the dependence of the balls number on their diameter takes on a stepped form (Figure 8). From formula (1) we determine the value of the axial force of the load mechanism of the stand.

Figure 8. Dependence of the number of balls in the load mechanism of the stand on their diameter.

The test stand allows realizing contact stresses of 1300-1700 MPa with 10 balls in the load mechanism with a force of 10 kN on each ball. With a force of 5 kN per ball, there should be 11-15 balls in the load mechanism.

Thus, the authors have proposed a stand design for accelerated life tests of railway wheelsets. The use of a stand with the parameters justified in this work will reduce the test duration by 10-15 times.

5. Conclusions
1) The authors performed a review and analysis of existing designs of stands for testing axles and methods of applying cyclic loads to the wheels of wheelsets of rail vehicles.
2) The dependence of the contact stresses on the wheel rolling surface on the ball’s diameter and the load on the contact area was obtained.
3) An approximate formula for calculating the number of balls in the load mechanism of the test stand is proposed.
4) We recommend a rational range of parameters of the load mechanism of the stand: at a load of 5 kN, the diameter of the balls is from 110 to 160 mm; at a load of 10 kN, diameter of the balls is from 160 to 180 mm.
Due to the large number of balls in the load mechanism of the stand, the tests will be significantly accelerated. The use of a stand with the parameters justified in this work will reduce the test duration by 10-15 times.

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