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# Justification of the parameters of the life-test bench for railway wheelsets

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**Abstract.** One of the tasks that need to be solved in order to predict the life of wheelsets is to assess the contact strength and wear of their parts. Thus, the purpose of this work is to develop a scheme and justification of the parameters of the life-test bench for railway wheelsets in conditions close to real operating ones. This bench simulates a contact of “wheel-rail” pair as a contact of “wheel-roller” pair. To ensure compliance of the bench tests conditions with the real wheelsets operation conditions we took that the maximum contact pressure in the “wheel-roller” pair is equal to the maximum contact pressure in the “wheel-rail” pair. As results of this study we obtained the dependences of the values of the roller diameter and the working force of the bench on the characteristics of the tested wheelset and proved the mutual influence of these quantities. For railway wheelsets with an axle load of 230.5 kN and wheels with a tread diameter of 957 mm, we obtained the following values of the bench parameters: roller diameter is 75 mm; working force of the bench is 8,375 kN; wheel speed is 500 rpm; number of rollers is 8; test duration is about 220 h.

## 1. Introduction

Any product has an element that limits the duration of its use. For railway rolling stock, such an element is a wheelset (WS), the resource of which is an important characteristic of the vehicle.

Current regulations [1, 2] provide for a number of bench tests of wheelsets, among which are those that allow you to assess the resource of the WS – fatigue testing of WS parts. The works [3, 4, 5] deals various aspects of conducting such tests and the corresponding bench equipment.

One of the tasks that need to be solved in order to predict the life of wheelsets is to assess the contact strength and wear of WS parts, their individual surfaces. Thus, in [6] a bench was proposed for testing railway wheels for contact strength. However, the tests are simulated, i.e. the object of research is not a railway wheel, but an equivalent roller.

A number of works [7, 8, 9] are devoted to the problem of determining and increasing the life of wheelsets and their elements.

The purpose of this work is to develop a scheme and justification of the parameters of the life-test bench for railway wheelsets in conditions close to real operating conditions.

## 2. General description of the life-test bench

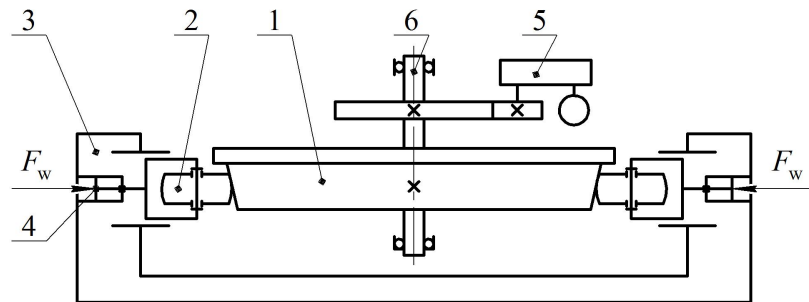
Since the life of wheelsets depends on the contact strength of the wheels, the object of the test is the railway wheel. The bench simulates the interaction of wheel with the rail under a load that corresponds to the wheelset axle load.



The general scheme of the life-test bench is given in figure 1.

Rollers 2 are installed on a frame 3 with the possibility of rotation relative to its own axes and radial movement relative to the test wheel 1. They contact with the test wheel 1 under pressure created by the loading device 4. Test wheel 1 is fixed on the axle 6 and together with it rotates by means of the bench drive 5.

Axle 6 can be both the axle for wheelsets (its fragment) and a separately manufactured part (technological axle). The bench drive 5 can be of any type and configuration. The working surface of the roller 2, in contact with the test wheel 1, corresponds to the shape of the rail head in cross section.



**Figure 1.** Scheme of the life-test bench:

1 – test wheel; 2 – roller; 3 – frame; 4 – loading device;

5 – bench drive; 6 – axle

( $F_w$  – working force of the bench)

The test wheel rotates for a certain period of time, which will be called the test duration. It corresponds (but does not equal) to the required life of wheelsets in terms of contact strength of the wheels.

Given the design of the bench, we highlight its parameters:

- roller diameter;
- working force of the bench;
- wheel speed;
- number of rollers;
- test duration.

Other parameters of the bench within this work will not be considered.

### 3. Study of contact strength of wheels

Within this work, the purpose of the study of the contact strength of wheels is to establish the relationship between the individual parameters of the bench and the wheelsets characteristics. The research results are based on the use of Hertz's elastic contact theory [10].

According to the provisions of the contact mechanics [10], in the general case, each of the bodies in contact has a surface curved in two mutually perpendicular directions. Such surfaces are characterized by relative radii of curvature, which are denoted: for the first body –  $R'_1$  and  $R''_1$ ; for the second body –  $R'_2$  and  $R''_2$  (in some cases, these values can be infinite – the surface is flat in one or both directions).

The railway wheel and the roller are bodies of uncoordinated shape with external point contact. The relative radii of curvature of each of the bodies are different ( $R'_1 \neq R''_1$ ;  $R'_2 \neq R''_2$ ), therefore, due to the application of the load, an elliptical contact area will be formed. This also applies to the contact of the wheel with the rail in real operating conditions.

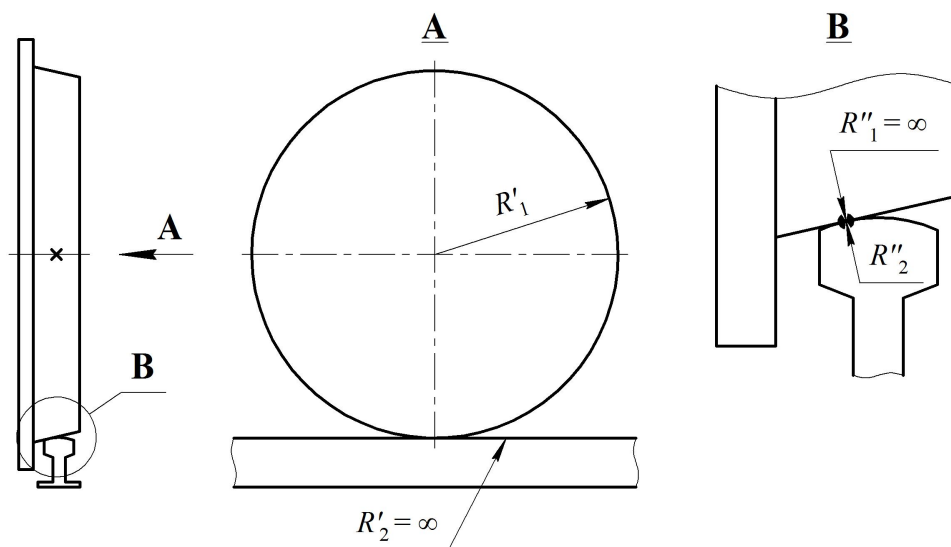
In further research, we believe that the values with the lower index “1” (except  $F_1$ ) characterize the railway wheel, and the values with the lower index “2” characterize the part with which the wheel is in contact (rail or roller).

We note that the life-test bench must reproduce the real operating conditions of the wheelset. This can be ensured if the characteristics of the contact of the pairs “wheel-rail” and “wheel-roller” is equal. Let us consider each of the options for interaction in more detail.

### 3.1. Contact of the “wheel-rail” pair

The contact of the “wheel-rail” pair is schematically presented in Figure 2.

As you can see, the contact surfaces are flat in different, mutually perpendicular planes, so  $R''_1 = \infty$  and  $R'_2 = \infty$ .



**Figure 2.** The scheme of contact of the “wheel-rail” pair

According to Hertz's elastic contact theory [10], we determine the following quantities:

- large relative radius of curvature

$$R' = \left( \frac{1}{R'_1} + \frac{1}{R'_2} \right)^{-1} = R'_1; \quad (1)$$

- small relative radius of curvature ( $R'' < R'$ )

$$R'' = \left( \frac{1}{R''_1} + \frac{1}{R''_2} \right)^{-1} = R''_2; \quad (2)$$

- equivalent radius of curvature

$$R_c = \sqrt{R'R''} = \sqrt{R'_1 R''_2}; \quad (3)$$

- maximum contact pressure

$$p_0 = \left( \frac{6PE^{*2}}{\pi^3 R_c^2} \right)^{1/3} \times [F_1(R'/R'')]^{-2/3}. \quad (4)$$

In formula (4), the following designations are adopted:  $P$  is the normal force in the contact zone – a component of the load in the contact zone, which acts along the common normal to the contact surfaces at the point of contact;  $F_1(R'/R'')$  is the coefficient that takes into account the eccentricity of the actual contact surface of bodies (ellipse), it is a function of the ratio of large ( $R'$ ) and small ( $R''$ )

relative radii of curvature, can be determined by recommendations [10];  $E^*$  is the reduced modulus of elasticity.

Normal force in the contact zone

$$P = 0.5F \cos \beta, \quad (5)$$

where  $F$  is the wheelset axle load;  $\beta$  is the wheel tread slope (the angle  $\beta = 2.86^\circ$  corresponds to a taper of 1:10, which is specified in GOST 10791).

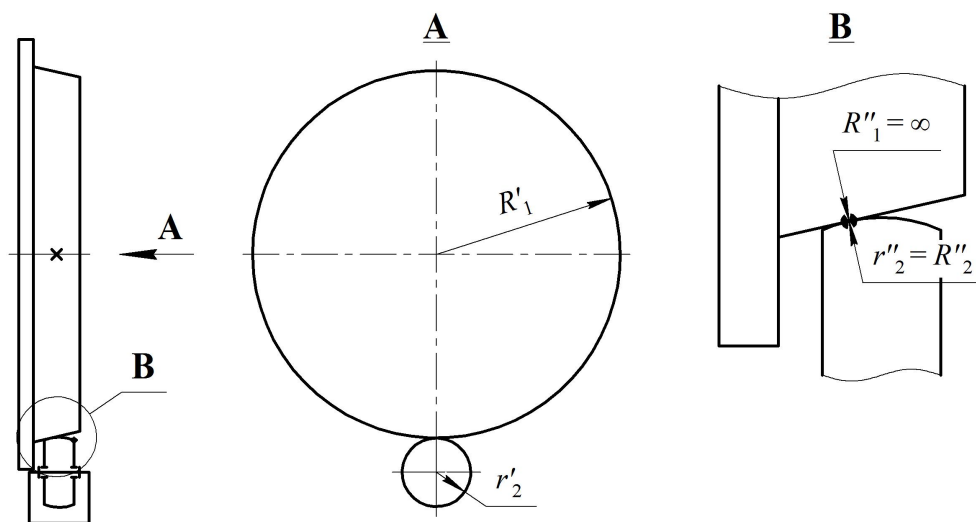
Reduced modulus of elasticity

$$E^* = \left( \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)^{-1}, \quad (6)$$

where  $E_1$  and  $\nu_1$  are the Young's modulus and the Poisson's ratio of the wheel material;  $E_2$  and  $\nu_2$  are the same values for the rail material.

### 3.2. Contact of the "wheel-roller" pair

The contact of the "wheel-roller" pair is schematically presented in figure 3.



**Figure 3.** The scheme of contact of the "wheel-roller" pair

The roller curvature  $r''_2$  in view B (figure 3) is equal to the rail curvature:  $r''_2 = R''_2$ .

The characteristics of contact are as follows:

- large relative radius of curvature

$$r' = \left( \frac{1}{R'_1} + \frac{1}{r'_2} \right)^{-1}; \quad (7)$$

- small relative radius of curvature ( $r'' < r'$ )

$$r'' = \left( \frac{1}{R''_1} + \frac{1}{r''_2} \right)^{-1} = R''_2; \quad (8)$$

- equivalent radius of curvature

$$r_c = \sqrt{r' r''} = (R_2'')^{1/2} \cdot \left( \frac{1}{R_1'} + \frac{1}{r_2'} \right)^{-1/2}; \quad (9)$$

- maximum contact pressure

$$p_0 = \left( \frac{6P_b E^{*2}}{\pi^3 r_c^2} \right)^{1/3} \times [F_1(r'/r'')]^{-2/3}. \quad (10)$$

In the formula (10) the normal force in the contact zone “wheel-roller” is

$$P_b = F_w \cos \beta, \quad (11)$$

where  $F_w$  is the working force of the bench – the force applied by the loading device to the roller.

### 3.3. Ensuring the adequacy of the bench tests conditions

To ensure compliance of the bench tests conditions with the real operation conditions of wheelsets, it is necessary that the maximum contact pressure in the “wheel-roller” pair is equal to the maximum contact pressure in the “wheel-rail” pair. Equating the right-hand sides of equations (4) and (10), we obtain:

$$\left( \frac{6PE^{*2}}{\pi^3 R_c^2} \right)^{1/3} \times [F_1(R'/R'')]^{-2/3} = \left( \frac{6P_c E^{*2}}{\pi^3 r_c^2} \right)^{1/3} \times [F_1(r'/r'')]^{-2/3}. \quad (12)$$

After performing mathematical transformations we have:

$$\frac{P}{P_b} = \left( 1 + \frac{R_1'}{r_2'} \right) \cdot \frac{F_1(R'/R'')^2}{F_1(r'/r'')^2}. \quad (13)$$

In [10] it was noted that in the first approximation the functions  $F_1$  can be taken as equal to one. So that

$$\frac{P}{P_b} = 1 + \frac{R_1'}{r_2'} \Rightarrow P_b(r_2') = P \cdot \left( 1 + \frac{R_1'}{r_2'} \right)^{-1} \quad (14)$$

The dependence analysis showed that the choice of rollers of smaller diameter ( $r_2'$  is the roller radius) allows to reduce the force  $P_b$  and, consequently, the working force of the bench.

Note that the life-test bench can be universal: the obtained dependence (14) in combination with formulas (5) and (11) allows determining the working force of the bench for different wheelset axle load, including the constant roller diameter, and the ability to move the rollers in radial direction relative to the wheel allows you to test wheels of different diameters.

It also follows from formula (14) that in the first approximation the roller profile ( $r_2'' = R_2''$ ) does not affect the adequacy of the test conditions. Studies conducted within this work have shown that at the stage of refinement, the influence of the radius  $r_2'' = R_2''$  is reflected in the coefficient  $F_1$  and is insignificant. Therefore, there is no need to use different sets of rollers to simulate the impact of different rails, which further indicates the versatility of our life-test bench.

### 3.4. Determining the roller diameter and the working force of the bench

When determining the roller diameter, we will start from the roller axle diameter, which is determined from the condition of bending strength; after transformations it takes the form

$$\sigma = \frac{8F_w L}{\pi d_a^3} \leq [\sigma], \quad (15)$$

where  $L$  is the distance between the axle supports;  $d_a$  is the roller axle diameter;  $[\sigma]$  is the permissible axle bending stresses.

Combining formulas (5), (11), (14), counting  $R'_1 = 0.5D_w$  ( $D_w$  is the test wheel diameter),  $r'_2 = 0.5d_r$  ( $d_r$  is the roller diameter) and assuming  $d_r = 3d_a$ , we obtain

$$F_w = 0.5F \left( 1 + \frac{D_w}{3d_a} \right)^{-1}. \quad (16)$$

Substituting (16) into (15) and performing mathematical transformations, we obtain the equation for determining the roller axle diameter:

$$ad_a^3 + bd_a^2 + d = 0, \quad (17)$$

where  $a = \pi$ ;  $b = \frac{\pi D_w}{3}$ ;  $d = -\frac{4FL}{[\sigma]}$ .

By equation (17) we determine the roller axle diameter  $d_a$  and then we determine the roller diameter as  $d_r = 3d_a$ .

As a special case, we determine the roller diameter and the working force of the life-test bench for railway wheelsets with wheelset axle load  $F = 230.5$  kN, which contain wheels type A1 according to GOST 10791 with a tread diameter  $D_w = 957$  mm.

Given the width of the rail P65 according to DSTU 4344 (75 mm) we take the distance  $L = 80$  mm. Permissible bending stresses for the roller axle made of medium carbon steel  $[\sigma] = 120$  MPa.

Equation (17) shows the minimum required value of the roller axle diameter is 24 mm. Given that the roller will be mounted on the axle using rolling bearings, we assume  $d_a = 25$  mm. So that the roller diameter

$$d_r = 3d_a = 3 \cdot 25 = 75 \text{ mm.}$$

By formula (16) we determine the working force of the bench:

$$F_w = 0.5F \left( 1 + \frac{D_w}{3d_a} \right)^{-1} = 0.5 \cdot 230.5 \cdot \left( 1 + \frac{957}{3 \cdot 25} \right)^{-1} = 8.375 \text{ kN.}$$

#### 4. Determination of test duration, wheel speed and number of rollers

The test duration ( $T$ ) depends on several factors:

- required wheel life on contact strength in kilometers ( $L_w$ );
- number of rollers ( $k$ );
- wheel speed in revolutions per minute ( $n$ ).

These values are related:

$$T = \frac{60L_w \cdot 10^3}{\pi n D_w k}. \quad (18)$$

When it comes to wheel life ( $L_w$ ), there is somewhat conflicting data. Biserikan notes [8] that about a third of the wheels of the car fleet have a mileage between rebounds, less than 160

thousand km. The maximum overhaul mileage is 300 thousand km. The results of Vorobyov's research [6] are presented more widely with their division according to the conditions of rolling stock operation and wheeled steel grades: for freight cars – 76 to 108 thousand km, for passenger cars – 196 to 436 thousand km.

To perform approximate calculations, we take the life of wheels  $L_w = 160$  thousand km. The test wheel diameter is  $D_w = 957$  mm.

Let the wheel speed is  $n = 500$  rpm, the number of rollers is  $k = 8$ . Then the test duration will be

$$T = \frac{60 \cdot 160000 \cdot 10^3}{3.14 \cdot 500 \cdot 0.957 \cdot 8} = 800 \cdot 10^3 \text{ s} \approx 220 \text{ h.}$$

The obtained dependence allows setting the value of the wheel speed and the number of rollers to obtain acceptable test duration, taking into account the required life of a wheel (wheelset).

## 5. Conclusions

We developed the scheme of the life-test bench for railway wheelsets. This bench simulates a contact of “wheel-rail” pair as a contact of “wheel-roller” pair. To ensure compliance of the bench tests conditions with the real operation conditions of wheelsets we took that the maximum contact pressure in the “wheel-roller” pair is equal to the maximum contact pressure in the “wheel-rail” pair.

Based on the provisions of the contact mechanics, in particular, the Hertz's elastic contact theory, we obtained the dependences of the values of the roller diameter and the working force of the bench on the characteristics of the tested wheelset. The mutual influence of these quantities is proved.

A functional relationship has been established that relates the test duration, the wheel speed and the number of rollers; it also determines the influence of the wheelset characteristics on these values.

For railway wheelsets with an axle load of 230.5 kN and wheels with a tread diameter of 957 mm, we obtained the following values of the bench parameters: roller diameter is 75 mm; working force of the bench is 8,375 kN; wheel speed is 500 rpm; number of rollers is 8; test duration is about 220 h.

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