Methodology for estimating the resource of the friction vibration damper of a freight car trolley

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Abstract. The concept of digitalization of railway transport and the introduction of digital technologies provides for the creation of a "Digital Railway" based on a "Digital Twin", including a "Digital Twin" of a car, which implements a number of information and organizational measures aimed at assessing the current technical condition of a real car during its operation; reducing the cost of the life cycle of the car; increasing the reliability of assemblies and parts (increasing the overhaul life); reduced maintenance costs; creation of a service maintenance system for freight cars throughout the life cycle. However, the wear of vibration damper parts is the most important parameter that determines the turnaround time, the volume of repairs and the dynamic qualities of the car, which requires more detailed and reliable scientific substantiation.

The assessment of the wear of vibration damper parts (friction bar, friction wedge, bolster) is carried out in two ways - by direct examination of them in operation (in this case, wear is estimated by changing the linear dimensions, i.e., in mm over the service life) and according to the results of bench tests of models of vibration damper or testing samples on friction machines (in this case, wear is estimated by the mass of the worn-out material). The proposed method for predicting the wear of parts of frictional vibration dampers implements the Archard friction model, takes into account the variability of loads acting on the working surfaces, for which a method for determining the friction path under various driving conditions has been developed. The developed methodology makes it possible to evaluate their service interval at the design stage of the car's running gears.

1. Introduction

In 2017, the government of the Russian Federation adopted a bill [1], which provides for the development of digital and information technologies in many areas of the economy of the Russian Federation. In the same year, Russian Railways, relying on this law, adopted the concept of digitalization of railway transport and the introduction of digital technologies [2]. One of these areas was the creation of a “Digital Railway” based on the “Digital Twin”, including the “Digital Twin” of the car, which provided for a number of information and organizational measures, namely: monitoring and assessment of the current technical condition of a real car during its operation; reducing the cost of the life cycle of the car; increasing the reliability of assemblies and parts (increasing the overhaul life); reducing maintenance costs; creation of a service maintenance system for freight cars throughout the life cycle, etc. [3].
These approaches should fundamentally change the existing and used system of repairing freight cars. As a result, it should be expected that the “Digital Twin” technology will make it possible to switch from a planned preventive system of car repairs to repairs based on technical conditions.

In work [4], the author found that the repair system "on condition", in addition to railway transport, is actively used in many areas of technology, and each of the industries uses its own method of assessing the technical condition.

In the railway area, there were also prerequisites for the transition to a system for repairing a car according to the technical condition [4], and various methods were proposed for determining the state of units and parts. As a rule, these methods suggested assessing the actual resource of car assemblies and parts using the principle of determining the fatigue fracture durability [5, 6], or using their gamma resource to determine the fatigue resource [7].

Since the existing and currently used system for repairing units and parts of a car, according to [8], is based on the restoration of their linear geometric dimensions, which change in the process of wear of their contact friction surfaces, in our opinion, it is advisable to determine the actual resource of car parts from the point of view of the wear, and not their fatigue state [4].

The purpose of this work is to develop a technique for assessing the wear of parts of a friction vibration damper and its subsequent conversion in the "Universal Mechanism" software environment.

2. Materials and methods
In work [9], to determine the resource of car parts, the authors proposed to use the Archard wear model [10] based on the hypothesis of a linear relationship between volumetric wear by friction and normal force:

\[ W = k \frac{NS}{H}, \]  

(1)

where: 
- \( k \) – wear factor;
- \( H \) – material hardness;
- \( N \) – normal contact force;
- \( S \) – sliding distance (friction path).

This wear model is implemented in the "Universal Mechanism" software package, and is used to predict the wear of the profiles of railway wheels and rails [11].

As a mathematical model of the car, the model of the gondola car 12-132 was used, which was developed at the department of cars of the USUPS under the guidance of Professor A.E. Pavlyukov and implemented in the "Universal Mechanism" software package [12]. Figure 1 shows the structural graph of this model.

The structure of the graph was designed so that there are no closed kinematic chains in the model, which greatly simplifies the modelling process.

The main bodies of the model are interconnected by various kinds of connections: power (solid lines) - power elements (springs, dampers, contacts) connecting individual bodies; kinematic connections between bodies - dotted lines.

A characteristic feature of a wedge vibration damper is the presence of contact interactions "vertical surface of the friction wedge - side frame" and "inclined surface of the friction wedge - inclined surface of the bolster".

When analysing the car model, it was found that the interactions between the contact surfaces of the friction wedge, the inclined surface of the bolster and the friction bar are described by point-to-plane contact with four contact points on each surface (at the corners of the surfaces).

In our opinion, such a number of contact points on the studied surfaces is not enough, since a limited number of contact points will not be able to fully describe the physical process of wear of each contact surface and obtain a profilogram of the wear of the studied contact surfaces.

In connection with the above, it was decided to refine the mathematical model of the friction damper unit of the bogie vibrations, which consisted in increasing the number of contact points on the studied surfaces.
To simulate the contact interaction between the surfaces "Vertical surface of the friction wedge - friction bar" and "Inclined surface of the friction wedge - bolster" a mathematical model of the force of the "point-plane" type was used [12].

When describing contact forces, the coordinates of the contact points of the first body, the coordinates of the point and the direction of the outer normal of the contact plane of the second body, the coefficients of sliding friction, rest, the coefficient of contact stiffness and dissipation were indicated.

When describing contact interactions "lateral frame - friction wedge" and "friction wedge - bolster" four contact surfaces of force interaction were created:

- IncPlane – thirteen points of contact of the wedge with the bolster, inclined contact, angle 45°.
- LeftPlane – four points of lateral contact of the wedge with the side cavity of the groove of the bolster, the points lie in the vertical plane.
- RightPlane – four points of the second lateral contact of the wedge with the side cavity of the bolster groove, the points lie in the vertical plane.
- Lateral – fifteen points of contact with the side frame, the points lie in an inclined surface with an inclination of 1°.

All power elements are described in the subsystem as external, that is, the second contact body is not specified. The power elements are of the close contact type without autodetection of the normal. Figure 2 shows a wedge showing contact points on its surfaces.

A distinctive feature is the "friction wedge - side frame" contact, where a restriction on the dimensions of the contact plane was introduced. The contact plane in this contact has dimensions corresponding to the dimensions of the friction plate. Thanks to this, it became possible to determine the real magnitude of the displacement of the contact point and the magnitude of the normal force arising...
in the case of physical contact between the friction wedge and the friction bar. The parameters of the wedge-beam and wedge-frame contacts were calculated using the formula [11]

\[ c = \frac{P_0}{f_{st}}, \quad \mu = 1.2 \sqrt{\frac{cm_{klin}}{P_0}}, \]

(2)

where \( P_0 \) is the static value of the force acting on the wedge from the side of the spring that presses it; \( c \) is the stiffness of the spring that presses the wedge; \( f_{st} \) is the value of the static deformation at a separate point of contact under the action of the force \( P_0 \); \( m_{klin} \) – the mass of the wedge; the damping of the contact is 0.6 times the critical value.

Figure 2. Location of contact points on the surface of the wedge.
assemblies change, external and internal forces (inertial forces, normal forces, stresses in the contact zone) change, i.e., there is continuous feedback.

When creating a methodology for modelling and calculating the wear process, the method of discrete integration was used. It is based on the assumption that for some small interval $\Delta L$ the dimensions of the rubbing parts remain unchanged. The interval $\Delta L$ can be taken as the distance travelled by a car with a length of one km. For this small gap with constant surface dimensions, there are normal forces acting in the contact zone. Next, the relative displacement of the rubbing surfaces relative to each other is found and the thickness of the worn metal layer is determined.

To determine the dimensions of the layer removed from the surface for the next interval $\Delta L$ or cycle, new linear dimensions of the parts involved in friction are found. Dimensions as input parameters for determining the loads at the next step of the simulation are determined by changing the calculated dimensions of the previous segment travelled by the carriage. Then the calculation is repeated, similar to the previous step of integration.

To speed up the calculation and simplify the model, the following assumptions were made:

- the geometry of the surfaces remains unchanged at a certain interval of the friction path $\Delta S$, which is introduced by the initial parameter into the computational system;
- the value of the worn layer at a distance $S$ is determined as quotient $h/\Delta h$ (where $\Delta h$ is the thickness of the removed metal during the time the car passes one km of the railway track $\Delta L$, $h$ is the maximum thickness of the removed metal for setting a unit or part for repair);
- then a new calculation cycle is performed at the next distance $\Delta L$ until the car has covered the entire path $L$ or reaches the wear limit $h$, specified as an initial parameter.

The above assumptions can significantly reduce the calculation time and make it possible to analyze the system in a relatively short period time.

Figure 3 shows an algorithm for modelling the wear process of tribopairs "friction wedge - friction bar", "friction wedge - bolster".

The whole complex of calculation consists of three main stages. Let's consider each of the stages of the modelling process in more detail.

At the first stage of the calculation, the data are entered, which are the initial data for the calculation. These parameters are:

- the track travelled by the car ($\Delta L$), on which the calculation must be made;
- the speed of the car on the rails;
- the speed of the car on the rails;

The irregularities of the track are taken as a disturbing factor in the movement of the car according to [13].

When carrying out numerical experiments, the following modelling conditions and assumptions were set:

1. The car was moving along a straight section of one km long track with regulated irregularities of the railroad bed.
2. Numerical experiments were carried out at 12 different speeds from 10 to 120 km / h at two different degrees of car load (empty, loaded).

At the second stage of the calculation, a multicyclic parametric calculation is started, during which the following are determined:

- values of normal forces $N$ at the contact points of the wedge;
- friction paths of contact points. The contact point friction path was determined using an expression of the form:

$$\Delta S = \text{sign} \cdot N \int \sqrt{v_x^2 + v_y^2 + v_z^2}, \quad (3)$$

where $v_x$, $v_y$, $v_z$ are the projections of the speed of the friction wedge relative to the side frame;
$N$ is the value of the normal force when contact occurs between the point and the surface of the friction bar at the current time.

Wear $\Delta W$ in the section $\Delta L$ was calculated by the following formula (1).

At the third stage, the analysis of the results was carried out. The obtained data on wear make it possible to change the linear and geometric dimensions of the tribo pair for further research on the behaviour of this friction pair.

The calculation result by the above method is to obtain the dependence of the change in the dimensions of the wedge, friction bar and bolster on the operating time of the tribo pair.

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3. Results and discussion

To speed up the calculations and simplify the analysis of the results obtained for all speeds when carrying out numerical modelling in accordance with the developed methodology, it was decided to take as the analysed data the numerical values of the resulting vectors arising during the movement of the car of normal forces acting at the contact points, and the friction path of each of the contact points.

![Figure 3. Algorithm for predicting the service life of a part based on modelling the process of its wear.](image-url)
As an illustration of the obtained in the course of modelling the movement of a car at a speed of 60 km / h along a straight section of track with irregularities in values, figures 4 and 5 show graphs of changes in the resulting vectors of friction paths and normal forces, and a graph of changes in the vertical displacement of the bolster.

\[ a \] - friction path of the tribocontact "friction wedge-friction bar"; \[ b \] - friction path of the tribocontact "friction wedge-bolster"; \[ c \] - normal force of the tribocontact "friction wedge-friction bar"; \[ d \] - normal force of the tribocontact "friction wedge-bolster".

**Figure 4.** Graph of changes in the resulting vectors of the friction path and normal force during the simulation.

**Figure 5.** Graph of changes in the vertical displacement of the bolster when the car is empty at a speed of 60 km / h.
Based on the condition of the problem, the results of the required quantities (normal force $N$, friction path $S$ for the resulting vectors) were obtained in a similar way, which are shown in table 1.

**Table 1.** Results of numerical experiments to determine the normal force $N$, the friction path $S$ at different degrees of car loading.

| Speed, km / h | Friction wedge vertical surface - friction bar | Friction path $L_{fr}$, m | Inclined surface of friction wedge - inclined surface of bolster | Friction path $L_{fr}$, m |
|--------------|-----------------------------------------------|--------------------------|-------------------------------------------------------------|--------------------------|
|              | Normal force $N$, H                           | Friction path $L_{fr}$, m | Normal force $N$, H                                         | Friction path $L_{fr}$, m |
| 10           | 0.451                                         | 0.030                    |                                                             |                          |
| 20           | 0.483                                         | 0.032                    |                                                             |                          |
| 30           | 0.516                                         | 0.034                    |                                                             |                          |
| 40           | 0.552                                         | 0.037                    | 5173.72                                                     | 0.030                    |
| 50           | 0.591                                         | 0.039                    |                                                             |                          |
| 60           | 0.633                                         | 0.042                    |                                                             |                          |
| 70           | 0.677                                         | 0.045                    |                                                             |                          |
| 80           | 0.724                                         | 0.048                    |                                                             |                          |
| 90           | 0.775                                         | 0.052                    |                                                             |                          |
| 100          | 0.829                                         | 0.055                    |                                                             |                          |
| 110          | 0.887                                         | 0.059                    |                                                             |                          |
| 120          | 0.949                                         | 0.063                    |                                                             |                          |
| 10           | 0.451                                         | 0.027                    |                                                             |                          |
| 20           | 0.465                                         | 0.027                    |                                                             |                          |
| 30           | 0.478                                         | 0.028                    |                                                             |                          |
| 40           | 0.493                                         | 0.029                    |                                                             |                          |
| 50           | 0.508                                         | 0.030                    | 27951.74                                                    | 0.027                    |
| 60           | 0.523                                         | 0.031                    |                                                             |                          |
| 70           | 0.539                                         | 0.032                    |                                                             |                          |
| 80           | 0.555                                         | 0.033                    |                                                             |                          |
| 90           | 0.571                                         | 0.034                    |                                                             |                          |
According to [14], it is known that the wear coefficient for a friction bar made of steel 30HGSA is 0.367 and for a friction wedge made of VCh-120 cast iron is 0.3, and for an inclined surface of a bolster made of steel 20GL according to [15] the value of the wear factor is 1.2.

Let us determine the wear for each speed and degree of load according to the previously given dependence (1).

Table 2 shows the results of numerical experiments to determine the wear W of the inclined surface of the side frame, the inclined surface of the friction wedge, the vertical surface of the friction wedge, and the surface of the friction bar.

Table 2. Results of numerical experiments to determine wear at different degrees of car loading.

| Speed, km/h | Wear of the inclined surface of the bolster | Wear on the inclined surface of the friction wedge | Wear on the vertical surface of the friction wedge | Friction plate wear |
|-------------|--------------------------------------------|-----------------------------------------------|--------------------------------------------------|-------------------|
| Empty car   |                                            |                                               |                                                  |                   |
| 10          | 1.10E-06                                   | 2.19E-07                                      | 2.32E-06                                         | 4.28E-06          |
| 20          | 1.18E-06                                   | 2.34E-07                                      | 2.48E-06                                         | 4.58E-06          |
| 30          | 1.26E-06                                   | 2.50E-07                                      | 2.65E-06                                         | 4.90E-06          |
| 40          | 1.35E-06                                   | 2.68E-07                                      | 2.84E-06                                         | 5.25E-06          |
| 50          | 1.44E-06                                   | 2.86E-07                                      | 3.04E-06                                         | 5.61E-06          |
| 60          | 1.54E-06                                   | 3.06E-07                                      | 3.25E-06                                         | 6.01E-06          |
| 70          | 1.65E-06                                   | 3.28E-07                                      | 3.48E-06                                         | 6.43E-06          |
| 80          | 1.77E-06                                   | 3.51E-07                                      | 3.72E-06                                         | 6.88E-06          |
| 90          | 1.89E-06                                   | 3.75E-07                                      | 3.98E-06                                         | 7.36E-06          |
| 100         | 2.02E-06                                   | 4.02E-07                                      | 4.26E-06                                         | 7.87E-06          |
| 110         | 2.16E-06                                   | 4.30E-07                                      | 4.56E-06                                         | 8.42E-06          |
| 120         | 2.31E-06                                   | 4.60E-07                                      | 4.88E-06                                         | 9.01E-06          |
| Loaded car  |                                            |                                               |                                                  |                   |
| 10          | 5.94E-06                                   | 1.18E-06                                      | 1.25E-05                                         | 2.31E-05          |
| 20          | 5.40E-06                                   | 1.07E-06                                      | 1.29E-05                                         | 2.38E-05          |
According to [8], the maximum wear of the friction strip is 1.5 mm, the total wear of the friction wedge is 3 mm (no more than 2 mm per side). Since the wear of the inclined surface is controlled only when the car enters the scheduled repair, the thickness of the worn layer of the inclined surface is not regulated in any way. When repairing a bolster, its inclined surfaces are fused with material to a wall thickness of at least 7 mm. Based on the condition of wear and the requirements for the content of the friction wedge and since the wear of the bolster is not regulated, we will conventionally assume that the limit wear of the inclined surface of the bolster is equal to the maximum wear of the inclined surface of the friction wedge and will be equal to 1 mm. Based on the calculated experimental data obtained and given earlier, we calculate the service life (km) of each tribo unit to the limit state at which it is necessary to repair or replace a part. The resulting service life is shown in table 3.

Table 3. Estimated service life of parts.

| Car loading degree | Inclined surface of bolster | Friction wedge | Inclined surface of friction wedge | Vertical surface of friction wedge | Friction plate |
|--------------------|-----------------------------|----------------|-----------------------------------|-----------------------------------|---------------|
| Empty              | 609960                      | 793437         | 3070130                           | 578821                           | 235009        |
| Loaded             | 159779                      | 186873         | 804222                            | 135041                           | 58980         |
| Average life of assemblies | 384869          | 490155         | 1937176                           | 356931                           | 146994        |

The analysis of the results of the studies carried out makes it possible to establish the average mileage of the car before replacing the studied units. Based on the data obtained, it can be argued that the bolster must be delivered for repair after 385 thousand km, the friction wedge must be replaced after 357 thousand km, and the friction bar after 147 thousand km.
4. Conclusions
The developed technique can be used to determine the wear of various tribo-couplings of a freight car, depending on its mileage, speed on a rail track, load factor and many other parameters.

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