



## **Wear and reliability of cylindrical vehicle joints**

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### **Abstract**

A method for calculating the wear of car suspension liners is presented. To determine the wear by a computational method, a system of equations for the spherical contact of mates of the friction unit of a car was obtained. An example of calculating wear for a given service life of a ball joint is given. The issue of determining the reliability indicators of a cylindrical hinge by the wear criterion is considered. An experimental study of the effectiveness of various greases for car hinge joints has been carried out.

**Key words:** ball joints, contact pressure, wear, reliability, tests, prediction.

### **Introduction**

The suspension is one of the most important parts of a car in terms of the comfort of transportation of goods and passengers and, most importantly, in terms of traffic safety. Wear of friction units is the main reason for the decrease in suspension reliability. The suspension structure of the front wheels of a car contains many important friction units. Among these supports, the most important are ball bearings. These bearings have different friction pairs: steel on steel, steel on polymers, steel on rubber, etc. The study of the support structure is the first mandatory stage of calculations and tests for wear and reliability of friction units. It is also necessary to understand how the construction of the assembly works, as well as to understand the entire subsequent calculation methodology. Analysis of the design of ball joints of different cars allows us to trace the tendencies of their improvement. Suspension ball joints are applied between control arms and wheel carriers or knuckles of the suspension that allow relative articulation and rotation between the mating components. Depending on its application, the ball joint can transmit longitudinal and lateral forces with only a small proportion of forces as well as the complete vehicle weight in vertical direction. Much attention is currently paid to the study of the tribological aspects of extending the service life of the movable joints of a car [1]. At the same time, a comprehensive study of analytical and experimental methods for predicting the resource of vehicle friction units is an actual problem.

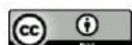
### **Calculation of wear and contact pressures in ball bearings of a vehicle suspension**

To determine the effectiveness of construction and lubricants used in transport, a calculation method is proposed for determining the wear of vehicle parts. The technique takes into account the properties of materials and the operating conditions of the friction unit and makes it possible to determine the durability of the friction unit by the wear criterion. The basis is the solution of the wear contact problem for a given contact pattern of parts.

In this case, the design of the ball joint of the car suspension is considered. In the ball joint, there is a contact interaction of a rigid spherical liner and a rigid cage according to the scheme in Fig. 1. A ball bearing of radius  $R$  rotates around its axis along the body and wears out. It is necessary to obtain the dependence of the wear of the mating of the axes of the main factors. The angle of the liner ranges from  $\varphi = \varphi_1$  to  $\varphi = \varphi_2$ .

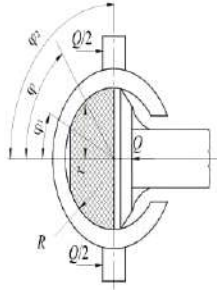
The mathematical formulation of the problem consists of an equilibrium condition, a geometric relationship, and a wear law.

The equilibrium condition for a spherical joint is:



$$Q = 2\pi R^2 \int_{\varphi_1}^{\varphi_2} \sigma(\varphi) \cos \varphi d\varphi, \quad (1)$$

where  $\sigma(\varphi)$  is the required pressure function.



**Fig. 1. Design scheme of the hinge**

The geometric dependence of displacements  $u_0$  on wear  $u_w(\varphi)$  is as follows:

$$u_w(\varphi) = u_0 \cos \varphi. \quad (2)$$

The wear law with two parameters  $k_w$  and  $m$  is taken as:

$$\frac{du_w}{dt} = k_w \sigma^m 2\pi n r, \quad (3)$$

where  $n$  is the number of vibrations of the hinge per unit time  $t$ ;  $r = R \cos \varphi$ .

As a result of differentiating condition (1) and equating (3), we obtain:

$$k_w \sigma^m 2\pi n R^2 \cos \varphi = \frac{du_0}{dt} \cos \varphi, \quad (4)$$

from here we have:

$$\sigma = \left( \frac{du_w}{dt} \right)^{\frac{1}{m}} \left( \frac{1}{k_w 2\pi n R^2} \right)^{\frac{1}{m}}. \quad (5)$$

As a result of mutual substitutions, we obtain the resolving integral equation of the problem:

$$Q = 2R \left( \frac{du_w}{dt} \right)^{\frac{1}{m}} \left( \frac{1}{k_w 2\pi n R^2} \right)^{\frac{1}{m}} \int_{\varphi_1}^{\varphi_2} \cos \varphi \sin \varphi d\varphi. \quad (6)$$

After integration, we obtain the dependence for the maximum wear:

$$u_w(t) = \left( \frac{Q}{\pi R^2} \right)^m \frac{k_w 2\pi R n}{(\sin^2 \varphi_2 - \sin^2 \varphi_1)^m} t. \quad (7)$$

If we take the friction path for one cycle  $s_1$ , then for the time  $t$  the total friction path is:  $s = s_1 n t$  or  $n t = \frac{s}{s_1}$ . As a result, for the friction path we get:

$$u_w(s) = \left( \frac{Q}{\pi R^2} \right)^m \frac{k_w 2\pi R^2 s}{\sin^2 \varphi_2 - \sin^2 \varphi_1}. \quad (8)$$

The formula for determining the contact pressure is obtained by substituting (8) in (5):

$$\sigma = \left( \frac{Q}{\pi R^2} \right) \frac{(k_w 2\pi R^2 n)^{\frac{1}{m}}}{\sin^2 \varphi_2 - \sin^2 \varphi_1} \left( \frac{1}{k_w 2\pi n R^2} \right)^{\frac{1}{m}}. \quad (9)$$

After the transformations, we finally get:

$$\sigma = \frac{Q}{\pi R^2 (\sin^2 \varphi_2 - \sin^2 \varphi_1)}. \quad (10)$$

An example of calculating the wear of the lower support in the contact of a hardened steel shaft and a polyurethane polymer liner.

Initial data. Hinge load 2,000N. The total path of friction in the joint for 10,000 km of run. Ball pin radius 16 mm. The parameters of the wear model for polyurethane are:  $m = 2.04$ ;  $k_w = 0.5 \cdot 10^{-8}$  (MPa)<sup>-m</sup>. Liner contact angles:  $\varphi_1 = 35^\circ$ ;  $\varphi_2 = 85^\circ$ .

The calculation of the wear of the liner is performed according to dependence (7) with the selected initial data per 10,000 km of run:

$$u_w = \left( \frac{Q}{\pi R^2} \right)^m \frac{k_w 2\pi R s}{\sin^2 \varphi_2 - \sin^2 \varphi_1} = \left( \frac{2000}{\pi 15^2} \right)^{2.04} \frac{0.5 \cdot 10^{-8} \cdot 2\pi \cdot 16^2 \cdot 3 \cdot 10^4}{\sin^2 85^\circ - \sin^2 35^\circ} = 0.233 \text{ mm}.$$

The obtained dependencies for calculations of wear are valid for all ball bearings of the front suspension. The difference lies in the different values of the contact angles  $\varphi_1$  and  $\varphi_2$ . The formula for calculating contact pressures is also common to all mates in ball joints. This calculates the wear due to the frictional path caused by the axial rotation of the ball pin during the rotation of the steering knuckle when turning from the steering. The friction path from lowering and raising the wheels while driving must be added to the total friction path in accordance with the suspension kinematics.

The maximum initial contact pressure in the hinge is determined by the dependence (10):

$$\sigma = \frac{Q}{\pi R^2 (\sin^2 \varphi_2 - \sin^2 \varphi_1)} = \frac{2,000}{\pi 15^2 (\sin^2 85^\circ - \sin^2 35^\circ)} = 3.95.$$

It has been established that this value of contact pressure does not change with wear of the liner.

Contact pressure is the main, basic characteristic of the friction unit. To determine pressures, it is necessary to have a solution corresponding to the contact problem. The upper and lower supports each have two types of spherical couplings, which differ both in the sign of curvature and in the properties of the contacting bodies. However, all mates can be represented as one generalized mate: the convex spherical surface of the finger and the concave spherical surface of the support body (or vice versa). In all cases, contact starts at some angle  $\varphi_1 > 0$  and ends at an angle  $\varphi_2 \leq 90^\circ$ .

### Reliability calculations of ball bearings

Let us assume that the current wear of the suspension ball bearings is distributed according to the normal law. In this case, the quantile up of the reliability function is determined [3] by the dependence:

$$u_p = - \frac{n-1}{(n^2 V_{w^*}^2 + V_w^2)^{\frac{1}{2}}} (\varphi), \quad (11)$$

where  $V_{w^*}, V_w$  is variation coefficients of limit and current wear;  
where  $n$  is safety factor for wear:

$$n = \frac{u_w^*}{u_w}. \quad (12)$$

The limit wear value can be taken constant  $u_w^* = 0$ , then the variation coefficient  $V_{w^*} = 0$ . In this case, from (12) we have:

$$u_p = - \frac{n-1}{V_w}. \quad (13)$$

By quantile, the reliability function is determined by the Laplace function:  $\Phi(u_p)$ :

$$P(u_w < u_w^*) = 0.5 + \Phi(u_p). \quad (14)$$

The determination of the general variation coefficient is performed through the partial coefficients of variation, taking into account the form of the main dependence (7).

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In this case, the load, the wear rate factor and the friction path are random values. The variation coefficients for these quantities  $V_Q, V_{k_w}, V_s$  are assumed to be known.

In accordance with the general methodology [2] determining the coefficient of variation of a function by the variation coefficients of the arguments, we have:

$$V_w = (m^2 V_Q^2 + V_{k_w}^2 + V_s^2)^{\frac{1}{2}}. \quad (15)$$

Let us determine the probability of failure-free operation of the ball joint of the car during 10,000 km of run. Let's use the initial data from the previous example. Let us take the coefficients of variation:  $V_Q = 0.4$ ,  $V_s = 0.4$ ,  $V_{k_w} = 0.35$ .

The general coefficient of variation of wear is determined by the dependence (15):

$$V_{u_w} = (m^2 V_Q^2 + V_{k_w}^2 + V_s^2)^{\frac{1}{2}} = (2.04^2 \cdot 0.4^2 + 0.35^2 + 0.4^2)^{\frac{1}{2}} = 0.98;$$

Average wear is determined by the formula (8). According to the accepted data, the wear of the ball joint was obtained for 10,000 km.  $u_w = 0.233$  mm.

According to [3], if, when checking the wear in the upper ball joint, the total indicators of the indicator exceed 0.8 mm, then the hinge must be repaired. Thus, dividing this value into two joints (lower and upper) we obtain the allowable wear value in the ball joint of the front suspension:  $u_w^* = 0.3$  mm.

$$\text{Safety factor: } n = \frac{u_w^*}{u_w} = \frac{0.3}{0.233} = 1.85.$$

The quantile of the probability of no-failure operation is determined by the dependence (11):

$$u_p = - \frac{n-1}{V_w} = - \frac{1.85-1}{0.98} = -0.867.$$

According to the tables of the Laplace function:  $\Phi(-0.867) = 0.3078$

Finally, reliability function according to (14):  $P = 0.5 + \Phi(u_p) = 0.5 + 0.3078 = 0.8078$ .

The given example is methodological in nature. For a more reliable assessment of reliability, it is necessary to perform an experimental determination of both the average values and their coefficients of variation.

### Determination of wear parameters of ball bearings

In the calculation formulas, the wear parameters  $m$  and  $k_w$  were taken as given. The problem of determining the parameters of the friction pair wear model is one of the stages of the computational and experimental evaluation of the wear of the ball joints. The problem of increasing the wear resistance of this unit can be solved in two directions: improving the wear-resistant properties of the liners and choosing new types of lubricants. Currently, great progress has been made in the development of lubricants with additives. The task is to assess the effectiveness of lubricants for use in ball joints. The wear characteristics of the materials were determined from the results of wear tests. The test scheme is shown in Fig. 2. An automobile ball joint polyurethane liner was used as a sample. The outer surface of the insert was tested with loading with a sphere diameter  $R = 28$  mm (Fig. 2).

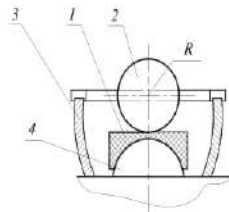


Fig. 2. Scheme of testing the polyurethane liner for wear:

- 1 – polyurethane hinge liner;
- 2 – steel counter sample HRC 50;
- 3 – container with grease;
- 4 – loading arm

The tests were carried out at a rotational speed  $n = 830 \text{ rev}^{-1}$  and a load of 10N. For lubrication, we used: ShRB-4 grease, Mobil grease, Shell grease. Three samples of each variant were tested. The contact patch was measured in two directions. The average test results are presented in table 1.

Table 1

### Test results of polyurethane with different types of lubricants

Friction path $s \cdot 10^6$ , mm	$a$ , mm			
	without lubrication	Shrb-4	Mobil	Shell
1	1,44	1,211	1,201	1,240
3	1,52	1,46	1,33	1,335
5	1,665	1,511	1,432	1,38
8	1,71	1,58	1,49	1,50
10	1,802	1,715	1,575	1,585

The test results were approximated by a power function of the dependence of wear on the friction path in the form:  $a = cS^\beta$ . The power-law approximation parameters were determined using the Excel program (Table 2).

Table 2

### Results of determining wear parameters for various lubricants

Parameters	without lubrication	ShRB-4	Mobil	Shell
$\beta$	0.105	0.14	0.118	0.14
$c$	0.33	0.25	0.269	0.295
$m$	3.65	3.15	3.52	3.34
$k_w$	$13.4 \cdot 10^{-6}$	$0.425 \cdot 10^{-6}$	$0.38 \cdot 10^{-6}$	$0.425 \cdot 10^{-6}$
$I(7\text{MPa}) \cdot 10^{-6}$	2.175	0.0827	0.064	0.0745
$I(2\text{MPa}) \cdot 10^{-6}$	0.0378	0.0029	0.00156	0.00175

The formulas for determining the wear parameters were determined by solving the inverse wear-contact problem [4]. Relationships (1 - 3) were used as the basic equations. As a result, we got the formulas:

$$m = \frac{1 - 2\beta}{2\beta}. \quad (16)$$

$$k_w = \frac{c^{2m+2}}{(2m+2)(Q/\pi)^m R}. \quad (17)$$

The results of calculating the wear parameters are shown in table. 2. Comparison of the effectiveness of lubricants is carried out in terms of the wear rate at two pressures of 7MPa and 2MPa. The calculation of the intensity is carried out according to the main dependence:

$$I = \frac{du_w}{ds} = k_w \sigma^m. \quad (18)$$

The results of calculating the wear rate for various lubricants are shown in Table 2. From the results of comparison with the base grease ShRB-4 it follows: the use of Mobil grease reduces wear at 7 MPa by 1.2 times; at 2 MPa by 1.79 times; the use of Shell lubricant reduces wear at 7 MPa by 1.15 times; at 2 MPa, 1.70 times.

### Conclusion

1. To estimate the wear of the ball joints of the car suspension, the wear-contact problem was solved. The solution is based on: the equilibrium equation of the tribosystem, the law of wear, the geometric equation of continuity in contact. As a result, closed formulas were obtained for determining the amount of wear and contact pressure.

2. A method is proposed for calculating the reliability of ball joints of a car half-suspension by the criterion of wear. A method of sequential calculation of the coefficient of variation of wear is considered by the coefficients of variation of random parameters that are included in the formula for calculating wear.

3. Based on the solution of the inverse wear-contact problem, formulas are proposed for calculating the parameters of the wear law. A comparative assessment of the tribological properties of lubricants for ball joints of the car suspension is carried out.

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**Диха О., Падгурскас Ю., Кукурудзяк Ю., Бабак О.** Знос та надійність циліндричних шарнірів автомобіля.

Представлена методика розрахунку зносу вкладишів підвіски автомобіля. Для визначення зносу розрахунковим способом отримана система рівнянь для сферичного контакту сполучень вузла тертя автомобіля. Наведено приклад розрахунку зносу для заданого ресурсу роботи кульової опори. Розглянуто питання визначення показників надійності циліндричного шарніра за критерієм зносу. Проведено експериментальне дослідження ефективності різних консистентних мастил для шарнірних з'єднань автомобіля.

**Ключові слова:** шарові шарніри, контактний тиск, знос, надійність, випробування, прогнозування.