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Issues of Developing Equal-Strength Two-Layer Spherical Rubber-Metal Hinges

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ABSTRACT

Development of equal-strength two-layer spherical rubber-metal hinges (RMH), described in the paper, is associated with the problem that with an equal thickness of the layers of rubber bushings and an equal opening angle, there is a significant difference in their radial stiffness and relative deformation of the rubber. With hinge dimensions corresponding to those of the hinges used in the undercarriage of locomotives, there is a difference in relative deformation of inner and outer bushings by about 1,5 times. As a result, it is proposed to determine the load capacity of spherical two-layer RMH by the value of relative deformation of the rubber of the most loaded bushing. Also, studies have been carried out on the possibilities of creating a uniformly deformable design of a spherical two-layer RMH.

To determine the characteristics of a spherical rubber-metal hinge, applied digital computer modelling based on the finite element method. A proposed parametrised geometric model of a spherical two-layer RMH and a finite element model of an elastic bushing offer the ratio of radial stiffnesses of outer and inner bushings, which is close to the preliminarily determined one, based on the equations of the theory of elasticity in displacements in a spherical coordinate system.

It has been established that to achieve uniform elasticity by changing the opening angle, the opening angle of the outer reinforcement of RMH should be approximately 1,5 times less than the opening angle of the inner one. This makes it possible to reduce the width of outer reinforcement of RMH by 25 % but raises the problem of strength and rigidity of outer edges of intermediate reinforcement. Also, equal elasticity of hinge bushings can be achieved due to their different thicknesses, while to achieve non-uniform stiffness of bushings within ± 5 %, it is required to ensure that deviation of the intermediate cage diameter during hinge manufacture is less than ± 0,1 %.

The obtained research results prove the practical possibility of creating an equally strong (with equal bushing rigidity) spherical two-layer RMH. The issue of searching for a compromise design of RMS, acceptable from the point of view of loading of the intermediate cage and of the requirements for manufacturing accuracy, requires further study.

Keywords: locomotive traction drive, spherical rubber-metal hinge, reliability, design.

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INTRODUCTION

Object of the Study and Description of the Problem

Spherical rubber-metal hinges (RMH), which do not require maintenance during operation, are widely used in the nodes of transport vehicles in domestic and foreign practices, in particular, in the nodes of the undercarriage of railway vehicles (2ES6, 2TE25A, EP20 locomotives, etc.). This determines the relevance of development of methods for calculation and design of RMH, as well as of the search for new designs of spherical RMH with a higher load capacity compared to world counterparts.

Previously, the authors of [1], based on the research using mathematical modelling, found that rigidity of two-layer spherical rubber-metal hinges (RMH) in the radial direction can be 6,3 times higher than that of single-layer ones, all other conditions being equal. This means that at a rubber hardness of 70–80 units according to Shore and with the same dimensions, as of the hinge of the suspension of the 2ES6 traction electric motor (outer diameter of 120 mm), radial stiffness of the hinge should average about 620 kN/mm, which is almost three times higher than radial stiffness of the hinge 13–4007 of Trelleborg Antivibration Solutions (Sweden), having an outer diameter of 127 mm, and seven times higher than radial stiffness of the hinges 13–1180 and 13–2624 of the same dimensions of the same manufacturer. Thus, double-layer hinges make it possible to significantly surpass the load capacity of single-layer hinges of some foreign manufacturers, using domestic materials and components and domestic technological base.

However, it should be taken into account that if for a single-layer spherical RMH, as a criterion for assessing the allowable radial loads, it is possible to use the value of the relative compression strain of the rubber layer, determined by the radial deformation of the entire RMH, as suggested in [2], then in a two-layer one, it is also necessary to consider that relative deformation of the RMH outer and inner rubber bushings can be different. Let's explain this with the following example.

The works [3; 4], based on the equations of the theory of elasticity in displacements in a spherical coordinate system, propose the following formula for determining radial stiffness of single-layer spherical RMH, kN/mm:

$$s_r = 6\pi G R_1 \frac{\left(\frac{R_2}{R_1}\right)^{-1} - \left(\frac{R_2}{R_1}\right)^{4}}{5 - 4{,}5\left[\left(\frac{R_2}{R_1}\right)^{2} + \left(\frac{R_2}{R_1}\right)^{-2}\right] + 2\left[\left(\frac{R_2}{R_1}\right)^{3} + \left(\frac{R_2}{R_1}\right)^{-3}\right]}$$

where G – shear modulus, mPa,

R_1 – outer cage sphere radius, mm;

R_2 – inner cage sphere radius, mm.

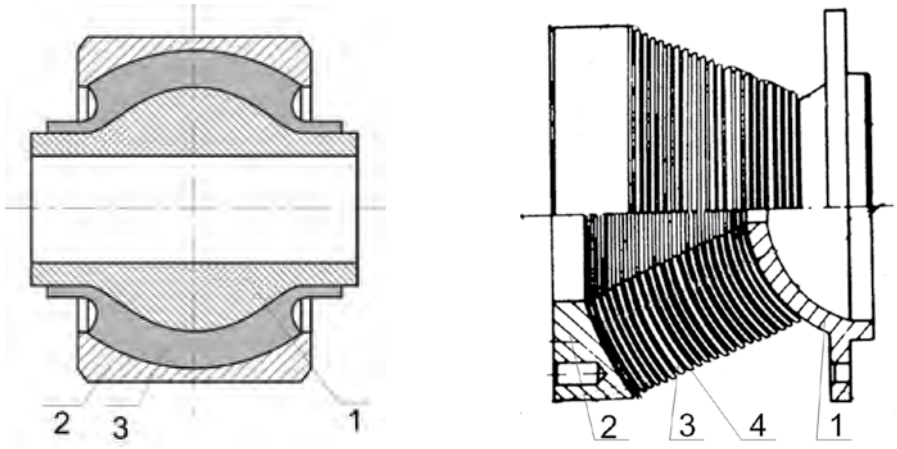
According to studies [5], calculations of stiffness of rubber-metal elements based on dependences given analytically are usually approximate, so we will evaluate the ratio of stiffness using formula (1).

Let us imagine a two-layer spherical hinge in the form of two single-layer ones with the same opening angle, for which the intermediate sleeve is internal and external, respectively. As an example, let's take the hinge used in the suspension according to the drawing of 2TE25A diesel locomotive, provided that the rubber layer is separated in the middle by an intermediate sleeve 2 mm thick. Then for the outer layer R_1 = 55 mm, R_2 = 51 mm; for the inner layer, respectively, R_1 = 49 mm, R_2 = 45 mm. We conditionally accept the value G = 1,4 MPa, since in the example under consideration the absolute value of stiffness is not of interest. Substituting these data into formula (1), we obtain the ratio of radial stiffness of outer and inner bushings s_{ro}/s_{rin} = 1,5. This example shows that with a simple separation of the rubber layer by an intermediate sleeve and the application of a radial load, the radial deformation of the rubber in the outer and inner layers will differ significantly, as a result of which the strength and durability of a spherical two-layer RMH under the influence of a radial load cannot be estimated based on the total radial deformation of RMH, correlated with total thickness of rubber layers.

There are two ways to approach this problem when designing a hinge.

Firstly, we can consider the relative deformation of the rubber layer only for the inner, most loaded bushing, assuming that the load on the outer one will be less. This approach is the simplest but leads to additional material consumption in manufacture of the outer sleeve, which has a deliberately overvalued load capacity.

Secondly, one can try to create an equal-strength structure with the same relative deformation of the outer and inner bushings. However, implementation of this approach is hampered by the fact that today not only is there



a) b)

Pic. 1. Schemes of spherical RMH: a – single-layer; b – thin-layer rubber-metal elements (TLRME); 1 – inner cage; 2 – outer cage; 3 – rubber layer; 4 – intermediate cage (reinforcement).
(E.g., Sokhan, O. N. Designing bushings of rotors for helicopters: Study guide. Moscow, MAI publ., 1981, 56 p. [Electronic resource]: https://perviydoc.ru/v5091/сохань_о.н._конструирование_втулок_несущих_винтов_вертолётов. Last accessed 24.10.2022).

no generally accepted method for optimising the parameters of two-layer hinges, but the very possibility of creating equal-strength hinges without significantly complicating their manufacturing technology has not been proven.

The *objective* of this article is an attempt to solve this problem.

Analysis of the Problem and Search for Methods to Solve It

First, it should be noted that at the moment two-layer spherical RMH have no structural analogues produced by foreign companies, and the authors have not found any scientific publications on creation of equally strong structures of two-layer spherical RMH at the time of writing. Moreover, practically no patent solutions have been found for equal-strength designs of such hinges. So, in the patent [6], the rubber of the bushings does not close after assembling the hinge, which increases the free surface of rubber bushings and reduces radial stiffness of the hinge, and, consequently, its load capacity, while, as far back as in [7], the authors proposed hinge, after assembly of which there is no internal cavity. Studies [3] found that for a single-layer hinge, when the gap between the outer bushings is closed, radial stiffness, and, accordingly, the load capacity, increases by 1,7…2 times.

Theoretical studies of multilayer rubber-metal elements are currently being carried out in the field of flat supports (for example, in [8]), as well as of spherical rubber-metal thrust bearings with a side surface close to conical [9], the nature of deformation of which under radial loading differs from the nature of deformation of a rubber layer of spherical RMH used in the undercarriage of rail vehicles.

The *analysis* of designs of spherical RMH used in various fields of technology, carried out by the authors, showed that today two approaches to the design of these products have spontaneously developed (Pic. 1a, b).

Single-layer spherical RMH (Pic. 1a) are designed and manufactured by the main foreign manufacturers (Trelleborg, GMT Rubber-Metal-Technik LTD, Vulcanite, Vibrachok-Paulstra, etc.) as general machine engineering products. According to the catalogues of companies, the products are RMHs with a detachable three-segment outer cage with radial connectors, with a rubber bushing opening angle close to the maximum possible for structural and technological reasons (the strength of the axis on which the hinge is mounted, or wedges for fastening and the ability to manufacture RMH by the method pressing or casting) and close to 90º. The thickness of the rubber bushing is taken within approximately 15…25 % of the radius of the outer spherical surface of the rubber bushing. The choice of such parameters is mainly dictated by the desire to obtain the highest load capacity of RMH with a single-layer structure and the requirements for maximum angles of rotation and skew within 7…10º.

Spherical thin-layer rubber-metal elements (TLRME), also referred to in the domestic

technical literature as spherical elastomeric bearings (Pic. 1b), are used in the attachment points of main rotors of domestic and foreign helicopters [10]. As a rule, they are thrust bearings made of many alternating layers of rubber 1–1,2 mm thick and of metal 0,8–1 mm thick, glued together [11] in the form of a truncated spherical cone with a hole. The shape of RMH is determined not by the requirements of uniform strength, but by the stability conditions of a rubber-metal package having a base width less than its height during movement of nodes and the application of loads. According to [11], the service life of a spherical TLRME is limited to four years due to the condition of rubber aging. As established in [12–14], the mathematical modelling of operation of TLRME is preliminary, since «taking into account the complexity of the physical and mechanical transformations… in the process of cyclic deformation of rubber, the specific composition of the material, the influence of the dynamic component of the load, the scale factor, frequency and speed of loading, functional purpose and reliability requirements, the cyclic strength of TLRME requires experimental confirmation on full-size models» ([12]).

Thus, the analysis of the existing experience in creation of RMH did not allow us to identify the empirical ratios of parameters that ensure the equal strength of the structure, therefore, the authors decided to carry out further search by the *method* of mathematical modelling.

The analysis of design of a spherical two-layer RMH showed that achievement of the same relative radial deformation of the outer and inner rubber bushings can be achieved in the following ways:

a) Use of rubber of different hardness for the outer and inner bushings with the same thickness of the bushings and equal opening angles.

b) Changing the shape factor for the outer sleeve by reducing its opening angle.

c) Different thicknesses of outer and inner sleeves.

d) An increase in the free surface of the outer sleeve, for example, due to incomplete closure of the parts of the sleeve during the installation of RMH.

It is difficult to implement the method «a» in practice, and not so much because of the technological difficulties of pressing or casting two different rubber mixtures into a mould, but because of a significant spread in the hardness values of rubber of the same brand, as a result of which the ratio of the shear modulus values for different batches of products will vary significantly. It is also technologically difficult to provide the required shape and dimensions of the voids at the junction of the hinge parts during installation in the case of using the «d» method. In this regard, the authors decided to conduct research regarding options «b» and «c». The task of modelling was to check the possibility of achieving an equal value of the relative radial deformation of the outer and inner bushings while maintaining the geometric parameters of RMH, which make it possible to manufacture it by pressing or casting.

Creation of a Parameterised Mathematical Model

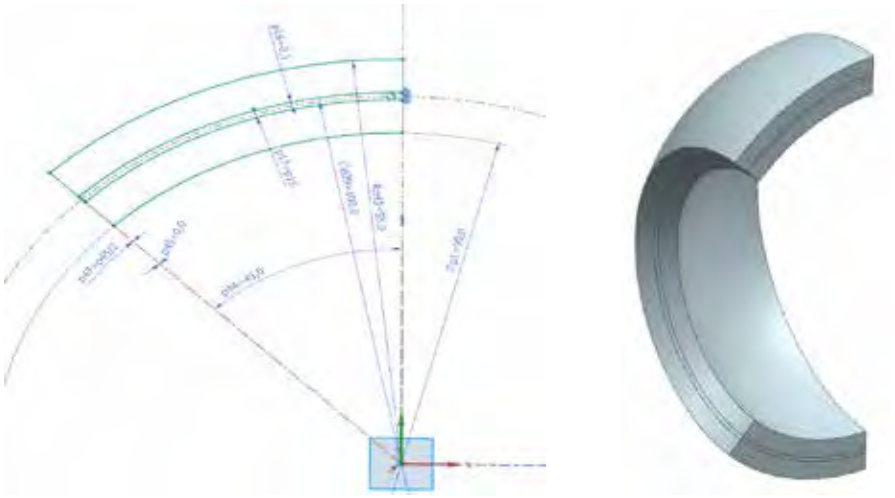
The calculation of spherical RMH by the finite element method (FEM) is associated with certain difficulties caused by variability of rubber properties [15] and other features previously noted by the authors in [1]. To solve the problem, a calculation scheme was formed, which is a three-dimensional finite element model of a quarter of RMH elastic bushing. The allocation of a quarter is due to the symmetry of RMH in two mutually perpendicular planes. This will lead to a reduction in the dimension of the problem, as well as will increase the convenience of processing the results.

The calculation scheme was formed on the basis of a solid-state three-dimensional model (Pics. 2, 3), during creation of which the values of the opening angles of the outer, inner, and intermediate sleeves were established by parametric dependencies, which allows automatically rebuilding the calculation scheme in multivariate calculations.

The finite element mesh is made with hexahedral volumetric elements with a breakdown dimension of 1,5 mm. As the material of outer and inner rubber layers, an orthotropic material is used, which has a linear force characteristic, which is true for small displacements of the inner ring of the hinge, according to experimental data. Standard steel was chosen as the material of the inner spacer sleeve, $E = 210000$ MPa/mm^2, $\mu = 0{,}3$.

At the central point of RMH there is a connecting element that connects it to all nodes of the inner surface of the elastic sleeve by absolutely rigid links.

In space, the calculation scheme is attached to the surface of the elastic bushing, which is in



a) b)

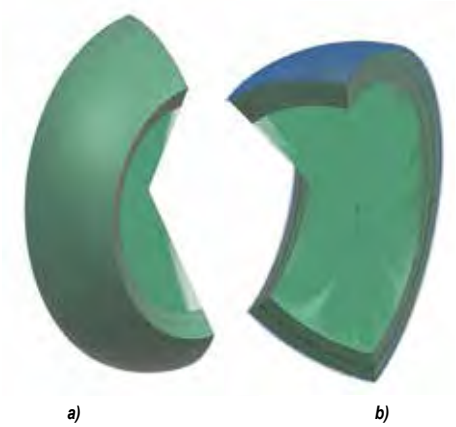
Pic. 2. Parameterised geometric model of a spherical two-layer RMH, adopted by the authors for modelling: a – main parameters of the model, b – geometric model of the elastic bushing.

contact with the outer ring of the hinge, by a rigid seal. Corresponding symmetrical bonds are imposed on the surfaces located on cutting planes. The middle node is given a forced movement of 1 mm in the radial direction.

The analysis of the stress-strain state of the elastic bushing showed that at equal opening angles of the upper, middle, and inner bushings of RMH equal to 90º, the displacements of the nodes of the middle bushing in the zone of maximum deformation are almost the same, which indicates that the deformations of the intermediate bushing (steel reinforcement) are negligible in comparison with the deformations of the rubber layers (Pic. 4).

With total movement of nodes in the radial direction by 1 mm, the deformation of the outer rubber layer was 0,397 mm. The displacement of the lower node of the inner rubber sleeve is 1 mm, and of the upper node is 0,398, then the total deformation of the inner layer along the median axis of RMH will be 1–0,398 = 0,602 mm. In other words, the absolute deformation of the inner layer is 1,5 times greater than that of the outer one (and, consequently, the relative deformation of the rubber of the layers, since their thickness is the same in this case).

Thus, the simulation results confirm the conclusion made based on the data obtained using formula (1): the load capacity of two-layer spherical RMH must be evaluated for the layer with the highest relative deformation. Let us turn to the analysis of the possibilities of creating RMH with the same relative deformation of the layers.



a) b)

Pic. 3. Finite element model of RMH elastic bushing adopted by the authors for modelling:
a – finite element mesh, b – constraints of the calculation scheme.

Modelling RMH with Different Values of Bushing Opening Angles

To equalise the values of the relative deformation of rubber layers, it is proposed to change the ratio of these parameters of the rubber layers by changing the values of the opening angles of the outer and middle bushings.

The opening angle of the middle bushing is assumed to be half the sum of the corresponding angles of the outer and inner bushings (parameter p47, Pic. 2).

The criterion of a uniformly deformed state will be approximation of the value of displacements of the nodes of the middle sleeve along the vertical axis of the hinge to the value of 0,5 mm. With equal thicknesses of rubber

Table 1

Movements of the middle bushing at different opening angles of outer bushing based on simulation results [obtained by the authors]

Opening angle of outer bushing	86º	82º	78º	74º	70º	66º	62º	58º	56º
Displacement of middle bushing	0,403	0,411	0,419	0,43	0,443	0,456	0,473	0,49	0,502

Table 2

The values of parameters of hinges with equal elasticity of rubber layers, according to the simulation results [obtained by the authors]

Radial stiffness, kN/mm	232,6	217	188,2	160,5	136,1	118,6	97,6
Opening angle of inner bushing, degrees	90	85	80	75	70	65	60
Opening angle of outer bushing, degrees	56	55	52	49	46	44	41

layers at such a displacement value, their equal relative deformations will be observed.

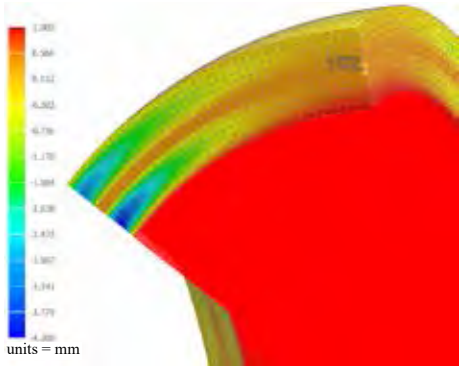
The results of variable calculations for various opening angles of the outer bushing are presented in Table 1.

The analysis allowed to obtain the shape of the elastic bushing, which ensures a uniformly deformed state (Pics. 5, 6).

Using the parametrised model described above, we obtained the values of the opening angles for RMH having different widths (Table 2), providing the condition of uniform elasticity.

As can be seen from Table 2, to achieve uniform elasticity, the opening angle of the outer reinforcement should be approximately 1,5 times less than the opening angle of the inner one. This allows to reduce the width of the outer reinforcement of RMH by more than 25 %, reduce the weight of the outer reinforcement, and also makes it easier to fill the space between bushings with rubber mass during the manufacture of the hinge.

The disadvantages of this RMH design include the fact that during preliminary deformation of rubber bushings during installation, as well as under the influence of radial loads, the pressure from the inside on the edges of the intermediate cage, created by the inner rubber bushing, will not be compensated by the pressure from the outside, because the edge of the intermediate cage is under the free surface of the outer rubber bushing. This means that it is necessary to study the stress state of the intermediate cage to determine its strength and rigidity, and, possibly, to increase the thickness of the cage edges. Another problem that requires further research is caused by the fact that in an equal-strength hinge, the rubber bushing has an acute angle at the point of junction of the inner surface with metal parts.

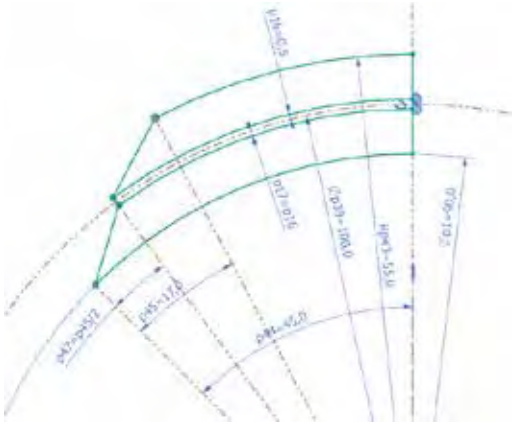


Pic. 4. Displacement of nodes of the middle bushing in the radial direction according to the results of simulation [obtained by the authors].

According to [2], if a part has a form factor of more than 1,5, then its strength needs to be determined by the strength of the rubber-metal connection. As indicated in [16], the strength of the bonding of rubber made of synthetic rubbers with metal using the Leikonat adhesive for rubber compounds can vary from 3 to 10 MPa. This means that with a given shape of the free surface of the rubber bushing, it is necessary to apply the most durable connection of rubber with metal parts. As already noted by the authors in [1], the characteristic defects of spherical RMH operating in the suspension units of traction electric motors of locomotives is delamination of rubber elements from metal parts [17–19]. The solution of the noted problems is the task of further research.

Modelling RMH with Different Thicknesses of Rubber Bushings

Using the same model, the authors searched for a uniformly deformed state of rubber with different thicknesses of rubber bushings for a spherical two-layer RMH with an opening angle of elastic bushings of 90º.



Pic. 5. Parameters of the shape of a hinge having uniformly elastic rubber layers, adopted by the authors for modelling.

Let's introduce the notation:

h_o – initial thickness of the outer rubber layer.

h_{in} – initial thickness of the inner rubber layer.

δ_l – movement of the lower node of the middle bushing with a single displacement of the inner ring.

δ_{up} – movement of the upper node of the middle bushing with a single displacement of the inner ring.

δh_o – absolute deformation of the outer layer.

δh_{in} – absolute deformation of the inner layer.

ε_o – absolute deformation of the outer rubber layer.

ε_{in} – absolute deformation of the inner rubber layer.

Since the deformation of the inner layer is greater for equal thicknesses, it is proposed to find an equilibrium state increasing stepwise the thickness of the outer layer and decreasing the thickness of the lower layer, respectively. The criterion for such a state will be equality of relative deformations of elastic layers.

Pic. 6. The shape of the hinge with equal elasticity of rubber layers according to the simulation results [obtained by the authors].

For an elastic bushing with an opening angle of 90º and a layer thickness $h_o = h_{in} = 4{,}5$ mm, we have $\delta_l = 0{,}397$ mm, $\delta_{up} = 0{,}398$ mm (Pic. 4). Since the nodes of the outer layer are rigidly fixed, $\delta h_o = \delta_l = 0{,}397$ mm. The lower ring has a displacement of 1 mm, so $\delta h_{in} = 1 - \delta_{up} = 1–0{,}398 = 0{,}602$ mm.

$\varepsilon_o = 100 \cdot \delta h_o / h_o = 100 \cdot 0{,}397/4{,}5 = 8{,}8$ %, $\varepsilon_{in} = 100 \cdot \delta h_{in} / h_{in} = 100 \cdot 0{,}602/4{,}5 = 13{,}4$ %.

By changing the values of the diameter of the axis of the middle bushing (parameter øp39, Pic. 5), we rebuild the model for further analysis.

The obtained values are given in Table 3 [performed by the authors]

As follows from Table 3, for the considered hinge with the parameters of the elastic bushings shown in Pic. 5, equal relative deformation of the bushings is ensured when the diameter of the centre line of the middle bushing is 99,1 mm, which is 0,9 mm less than the original size, while the permissible deviation in rigidity of the bushings ± 5 % corresponds to the deviation of the average diameter of the intermediate cage ± 0,1 mm, which is less than ± 0,1 % of the diameter. These accuracy requirements must be taken into account when designing the intermediate cage and moulds.

On the other hand, in the variant with different thicknesses of the rubber bushings, the angles between free surfaces of the rubber bushings and the metal parts to which they are vulcanised in the assembled hinge will be close to 90º, and thus the edges of the rubber bushings will be pressed against the metal parts due to the force created as a result of preliminary compression deformation during the assembly of RMH. This facilitates the technological task of ensuring reliability of the connection between rubber bushings and metal during vulcanisation. As a result, from the point of view of the theory of rational choice of decisions in design [20], the choice of the option to achieve equal strength of the hinge must be made based on the specifics of the specific technological base for production of tooling and rubber products.

Thus, from the simulation results obtained, it follows that it is practically possible to obtain a two-layer spherical RMH with equal relative deformations of the rubber of the outer and inner bushings both by reducing the opening angle of the outer bushing of the hinge, and by choosing the ratio of the thickness of the rubber layers of the outer and inner bushings. At the same time, in the first case, the problem arises of strength

Table 3

Parameter value	Middle bushing axle diameter, mm						
	100	99,8	99,6	99,4	99,2	99,1	99
h_o, mm	4,5	4,6	4,7	4,8	4,9	4,95	5
h_{in}, mm	4,5	4,4	4,3	4,2	4,1	4,05	4
δl, mm	0,397	0,432	0,467	0,5	0,533	0,55	0,566
δ_{up}, mm	0,398	0,433	0,468	0,501	0,534	0,551	0,567
δh_o, mm	0,397	0,432	0,467	0,5	0,533	0,55	0,566
δh_{in}, mm	0,602	0,567	0,532	0,499	0,466	0,449	0,433
ε_o, %	8,8	9,4	9,9	10,4	10,9	11,1	11,3
ε_{in}, %	13,4	12,9	12,4	11,9	11,4	11,1	10,8

and rigidity of the intermediate cage and the strength of the rubber-metal bond; in the second case, the problem arises of the accuracy of tooling manufacturing, in particular, of the choice of acceptable gaps for fixing metal bushings in the mould.

Since the fundamental possibility of creating a two-layer spherical RMH with equally loaded bushings might be considered proven, as one of the further areas of work on the study of such RMH, the authors propose to search for a RMH design that would provide a rational compromise between the requirements for the strength of RMH elements and the accuracy of manufacturing tooling.

CONCLUSIONS

1. It has been established that for spherical two-layer RMH with equal thickness of layers of rubber bushings and equal angle of their opening, there is a significant difference in their radial stiffness and relative deformation of rubber. With hinge dimensions corresponding to those of the hinge used in the undercarriage of locomotives, there is a difference in the relative deformation of the inner and outer bushings by about 1,5 times. As a result, the load capacity of spherical two-layer RMH must be determined by the value of the relative deformation of the rubber of the most loaded bushing. It is necessary to search for a uniformly deformable design of a spherical two-layer RMH.

2. In the process of analysing the existing structures of RMH, two possible ways to create a uniformly deformable spherical RMH were identified, implemented within the framework of the existing technology for manufacturing RMH by pressing and casting: by changing the opening angle of the outer and intermediate reinforcement and by changing the ratio of the thicknesses of the outer and inner rubber bushings.

3. In the process of research, a parametrised geometric model of a spherical two-layer RMH and a finite element model of an elastic bushing have been proposed. As a result of modelling the original version of the hinge with the same thicknesses and opening angles of elastic bushings, the ratio of radial stiffnesses of the outer and inner bushings was obtained; it is close to the previously determined one based on the equations of the theory of elasticity in displacements in a spherical coordinate system.

4. It has been established that to achieve uniform elasticity by changing the opening angle, the opening angle of the outer reinforcement of RMH should be approximately 1,5 times less than the opening angle of the inner one. This makes it possible to reduce the width of the outer reinforcement of RMH by 25 %, however, this raises the problem of strength and rigidity of the outer edges of the intermediate reinforcement.

5. It has been established that equal elasticity of the hinge bushings can be achieved due to their different thickness, while to achieve the non-uniform rigidity of the bushings within ± 5 %, it is required to ensure the deviation of the intermediate cage diameter by less than ± 0,1 % during manufacturing of the hinge.

6. The obtained research results prove the practical possibility of creating an equally strong (with equal bushing rigidity) spherical two-layer RMH. The issue of searching for a compromise design of RMH, acceptable from the point of view of the loading of the intermediate holder and the requirements for manufacturing accuracy, requires further study.

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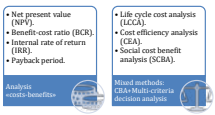
III

MANAGEMENT, CONTROL AND ECONOMICS



TRANSPORT POLICY 172

Decision-making based on multicriteria approach: concept, search for optimal criteria and application for existing projects.

INTERNATIONAL COOPERATION 180

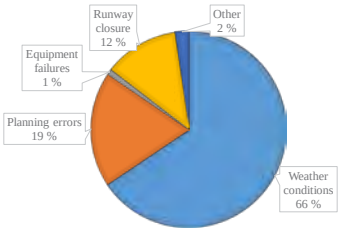
Russian-Chinese interaction in the sphere of rail transport.

SUSTAINABLE DEVELOPMENT 188

International standards, state regulation, suggestions to optimise assessment.

ORGANISATION MANAGEMENT 194

Structure of methods within the value chain. Universality and uniqueness of methodology referring transport companies.



AIRLINE COMPANIES AND AIRPORTS 205

Simulation model. How to counter disruptions in operations?

RAIL PASSENGER TRANSPORTATION 213, 224

Theory and practices of application of gravity model. Forecast model and its consistency.

Life cycle contract: legal and technological implementation aspects.

OVERSIZED CARGO 228

Algorithm of preparing the transportation as a warranty of safe and timely delivery.