The prospects of creation of the draft gear with the polyurethane resin elastic element for the rolling stock

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Abstract: In article the main requirements of normative documents to characteristics of the draft gears are analysed. The principles of operation of the existing draft gears are considered, the main directions of perfecting of the existing designs are defined and in essence new solutions are proposed. Test data of a test piece which allow to draw the unique conclusion on prospects of the developed design are provided.

1. Introduction

Beside workloads, that are necessary for transportation of different types of cargo, railway wagons have a lot of parasite loads, that appear because of ups and downs of the way, acceleration, braking actions and collisions of wagons. These loads dominate in life limitation of railway wagons. To increase the life of wagons it is necessary to decrease level of parasite loads and as a consequence all loads, that appear. Such method is called amortization. One of the most powerful ways of amortization is changing of power characteristic of the whole machine or its certain line of action. For transport, civil and metallurgic machines main ways of influence are energy and deformation influence. For horizontal forces that appear in coupler, the main influence is made by energy, hit influence. In this case power characteristic of between wagons connections should ensure absence of above the level axial loads.

2. Problem

Nowadays railway wagons supplied with the draft gears, that terminate energy in case of axial coupling of wagons. Such set is called buffer and has two elements: first one is elastic element or energy accumulator and the second one is friction element or vibration damper. First one accumulates energy and second one dissipates it and converts it in heat. The draft gears are used by humanity for many years and they become better with wagon characteristics. At the same time the size of them is rather conservative and doesn’t change for many years. Because of growth of weight and speed of wagons claims are grow too. Nowadays these claims are described In GOST 32913-2014 “Draft gears of the rolling stocks”. Also there is no solution for the problem of developing of draft gear with strict measures (230x320x570 mm) and at the same time simple, reliable and maintainable device that will conform to energy intensity regulations from GOST 32913-2014.
3. Article’s goals
In this article the vertical power line of the wagon, damping in wagons, axle buffer rod and other devices that decrease dynamic loadings on wheels of wagon are set aside. It is wanted to analyze horizontal loadings applied to the wagon and describe ways of horizontal parasite loads dropping. There is a task to develop draft gear for the rolling stock, where the energy intensity and damping facility will be increased and construction of the device will be simplified and the price will be reduced.

4. Analyses of last research and publications
Common questions about loadings in railway wagons and it’s protection with the help of draft gears are described in [1-7]. Damping of horizontal buffing loading are described in articles [8-11]. Variation of power characteristics of draft gears is shown in [12]. Design features of elastomer-friational draft gears are analyzed in [13-16]. Calculation methods of parts of buffer rods are shown in articles [17-20]. New brands of construction polyurethanes for elastic elements of buffer rods and their mechanical characteristics are described in [21, 22].

5. Materials and research results
In GOST 321913-2014- all draft gears are divided in three groups T1, T2, T3 by main technical characteristics. These properties are shown in Table 1.

Table 1. Performance standards of energy intensity of draft gears of freight rolling stock (GOST 3219-2014)

| Property                              | Draft gear’s group |
|---------------------------------------|--------------------|
|                                      | T1   | T2   | T3   |
| Static energy intensity, not more than, kJ (Est) | 30   | 40   | 60   |
| Nominal energy intensity, not more than, kJ (Es)  | 70   | 100  | 140  |
| Maximal energy intensity, not more than, kJ (Em)  | 90   | 130  | 190  |
| Nominal energy intensity as delivered, kJ         | For draft gears with friction point not more than 50 |
| Construction motion, mm                     | Not more than 120 |
| Impact speed of wagons with mass 100±5 each, during which the nominal power not exceeded, km/h | 7.5  | 9.0  | 11.0 |

On the horizontal powers, appearing impact of wagons, two regulations are introduced: nominal power \( F_n = 2000 \) kN, with energy intensity of T1 draft gear, which is \( U_n = 70 \) kJ and maximal power \( F_m = 300 \) kN, with energy intensity \( U_m = 90 \) kJ. Such two regulations have no reasons and sometimes misinform.

Analyses of draft gears that are used nowadays based on table 2 lets us assess their advantages and disadvantages. Despite of various constructions of draft gears, there are only two that can be claimed on acceptability to T1,T2,T3 classes, they are elastomer and frictioanl draft gears. Stem from main requirment (table.1) to the draft gears (stress intensity), it is possible look at elastomer draft gears, that have guidravlic system and are demphers. For closing static force they have extra elastic element with small stress intensity. Such devices can be made for all classes of draft gears because of the characteristics, but economic efficiency is unobvious. Details of this devices are made from high-strength materials, accuracy requirments for them are also very high. It leads to the complex technology and high price of draft gear, that is two times higher than draft gears with other constructions. Moreover, lifetime of such devices is much lower than was expected and serviceability is very high priced and there is no ability to repair them at the engine house. Frictional draft gears, introduced by spring-frictional and frictional with polymer support block, much cheaper anr not so complex, moreover they can be repaired in engine house. New designs of these devices have improved support and frictional blocks. In support blocks instead of steel cylindrical spiral
springs, polymer elements blocks are used, they are good for compression and work with loading near 300…400 kN instead 180 kN for steel springs. For frictional blocks there was testing with ceramic layers that have higher constant of friction.

Typical design example of spring-frictional and polymer-frictional devices might be APM-110-K-01 and APM-120-T1, PMKP-110, RT-120. On these devices wanted properties for T1 class were reached, however, such devices have disadvantages [15].
1. Wanted properties can be reached only after prework, at the beginning the main characteristic (stress intensity) two times smaller.
2. Such devices have bend in (stiff) characteristic with force bounces.
3. Energy intensity \( U_n = 70 \text{ kJ} \) and \( U_{\text{max}} = 90 \text{ kJ} \) can be reached on ultimate loads \( F_n = 2.0 \text{ MN} \) and \( F_m = 3.0 \text{ MN} \).

Despite the fact that all requirements can be satisfied, these devices cannot be perspective. For efficiency assessment of draft gears priority properties should be introduced. The main task of improvement of draft gear is to increase durability of wagon. That’s why the main property that should be controlled is the force that impact the wagon.

Nominal and maximal forces and stress intensity that are introduced in GOST 32913-2014 are conflicting, that makes it difficult to develop device that will meet requirements. When stiffness of draft gear will not be enough, required characteristics of stress intensity will not be reached even in with full device stroke, at the same time in condition of high stiffness compressing forces will reach requirements with slow device stroke and required characteristics of stress intensity will not be reached again.

Type of draft gear, that should be installed on the wagon is chosen depending on the cargo. Higher class can take higher impact energy and as a consequence impact on the wagon. During tests, when moving wagon with mass 100 tons kick standing one with the mass 94 tons normative axial loading 3.5 MN was got at next striking velocities
- 10-12 km/h class T1
- 13-15 km/h class T2
- 16-18 km/h class T3

Another important question that should be solved is quality assessment of the draft gear. Let’s look at it on the example of the class 1 draft gear.

On the figure 1 there are two curves of T1 class draft gears. Both of these characteristics correspond to GOST 32913-2014. However, loading \( F_m \), when needed stress intensity reached, differs a lot from each other; for the first curve it is \( F_m = 3.0 \text{ MN} \), for the second \( F_m = 1.5 \text{ MN} \).

Such a big difference leads to different strength of devices and different lifetime of wagons with such devices. At the same time, there is no information, which draft gear is more perspective. More perspective is such device, which better decrease parasite loadings.

From the figure 1 we can see that better characteristic is curve number 2, soft characteristic. Its area (characteristic of stress intensity) might be calculated with:

\[
U_{\text{max}} = \alpha F_m \lambda_m .
\]

\( F_m \) and \( \lambda_m \) – maximal forces and compression of device (on the power characteristic they are overall dimensions):
\( \alpha \) – characteristic charge ratio

This coefficient may take the value (from theory)
\( 0 \leq \alpha \leq 1.0 \).

From reality this range is smaller. Analysis of modern draft gears gives us next range:
\( 0.2 \leq \alpha \leq 0.8 \).
This coefficient is quality ratio of power characteristic. Smallest coefficients have springer-frictional draft gears. So draft gears AMP-120-T1 have quality ratio $\alpha = 0.25$, at the $F_M = 3.0$ MN and compression $\lambda = 120$ mm. That is not enough for keeping good lifetime of the wagon. For hydraulic draft gears, for example Z73, this coefficient is three times higher, that allows to get higher stress intensity or decrease $F_M$, that allows to increase lifetime of the wagon.

Another big problem with draft gears is the price. Because elastomer (hydraulic) devices are very expensive and nonrepairable in engine houses. That's why the main task is to develop effective draft gear with quality ratio $\alpha \geq 0.5$ and price not higher than $500$, that is two times cheaper than the hydraulic devices. It is possible if elastic element of such draft gear will not be liquid but solid, for example constructional polyurethan.

Solid elastomers are weakly compressible materials that is why roughness of detail, made from elastomer will increase, if loading will be like all-around compression [21,22]. Such stress state can be seen in places of elastic element where lateral strain embarrassed, for example in the zone of elastomer contact with bearing area, where friction forces block free expansion of compressed elastomer or in the places of elastomer connection with more rough material.

Let's analyse loading scheme, during which elastomer loses shape in closed volume. As an example, we'll look at the device, where strike energy is taken up during all-around compression and that is put in robust body. Using Hook's law it is easy to show that body strength depends on filling. Even in condition of static loading Poisson's ratio's value of filling determines value of the pressure that impact on body's walls. When $\mu = 0$, there is no pressure.

In condition of striking loading, value of the pressure depends on input energy and system's (that take strike) roughness values. Stiffness of such device defined by volume modulus of elasticity $K$ of filling-elastomer, that connected with other constants by

$$K = \frac{E}{3(1-2\mu)}$$

From formula, it can be seen that Poisson's ratio of filling material influences on device's stiffness and, as a consequence, on values of dynamic loadings. If the filling is elastomer such influence becomes the most important. Test, that were made in «Strength of materials» laboratory of Peter the Great Polytecnic University showed:

**Figure 1.** Power curves (loading) of T1 class draft gears: 1- stiff characteristic; 2-soft characteristic
1. All elastomers, despite different values of Young's modulus during compression $E_c$, showed similar $K$ modulus, that was nearly 3000 MPa.

2. For mostly used home-produced elastomers next values of Poisson's ratio were got a) Polyurethane SKU-PFL-70- $\mu = 0.4984$; b) Polyurethane SKU-PFL-100- $\mu = 0.4970$ c) Rubber V-14- $\mu = 0.4993$.

In case of ignoring deformation of steel body, that will go to resource of stress intensity, in first approximation then deformation's energy intensity of elastomer in closed volume will be:

$$\frac{\sigma_{\text{med}}^2}{2K} = \frac{F^2}{2K}$$

Then adjust energy intensity to the maximal in condition of one axial compression:

$$\frac{p^2}{2K} \approx 0.09E_c$$

From where

$$p_* = \sqrt{0.18K \cdot \frac{E_c}{6}} = 23.2\sqrt{E_c}.$$  

For polyurethane SKU-PFL-100 with $E_c = 60$ MPa we have

$$p_* = 23.2\sqrt{60} = 180$$MPa

For Rubber V-14 with $E_c = 15$ MPa

$$p_* = 23.2\sqrt{15} = 90$$MPa

With such pressure values, energy intensity is the same as the energy intensity of the element under one axial compression. With the bigger pressure the main scheme is all around compression.

Let’s try to approximately take into account body’s deformation. In case of using for body development high strengthened spring steel ($\sigma_v = 2 \cdot 10^5$ MPa, $\sigma_v = 1600$ MPa), then element’s relative strain will be $\varepsilon = 0.008$. Adjustment of element’s cross-section area will be $\alpha = 0.016$. So the deformation $\varepsilon$ will rise on the same value. Before deformation was $\varepsilon = 180/3000 = 0.06$, so $\varepsilon = 6\%$. Extra deformation will be

$$\frac{1.6}{6} \times 100\% = 26.6\%$$

Energy intensity will rise on the same value. Extreme pressure will be:

$$p_* = 180\sqrt{1.266} = 160$$MPa

For the such scheme the most important is possibility of further increase of pressure (for example from 160MPa to 300MPa), that ideally increases energy intensity in 3.5 times. Solution of such problem allowed to develop our draft gear for the rolling stock.

Draft gear for the rolling stock consists of cylindrical body 1, where step-shaped plunger 2 is located (figure 2). Plunger is centered in plugs 3 and 4, press-fitted front 5 and back 6 covers. Elastic element is installed in body 1. Covers tightened with bolt pins 8 and screws 9. Plunger 2 goes inside elastic element 7 with bigger diameter and goes outside with smaller one.

Assembly of draft gear. Plugs 3 and 4 press-fitted in covers 5 and 6. Then back cover should be installed horizontally, then bolt pins should be screwed in it. Then body 1, where elastic element 7 and step-shaped plunger 2 are located, should be installed on the cover 6. After this cover 5 should be installed on plunger 2, on the bolt pins screws should be engaged. In such condition draft gear should be installed in press and be tightened, loading should be applied to the covers 5 and 6, then screws 9 should be tightened.

Work of draft gear. During compression and stretching of coupler, draft gear is compressed, forces are applied on he one side to the plunger 2 and from the other side to the back cover 6. Plunger 2 goes into elastic element 7 and compresses it. Elastic element at this moment is under all-around compression. Force of elastic stiffness, that impact plunger in axial direction, proportional to relative volume change, volume modulus of elasticity and difference between squares of plunger's 2 cross-sections in two sections- higher and lower its cone area. Aside from that, there are friction forces of elastomer that
influence plunger in axial direction. They are proportional to relative elastomer volume change, volume modulus of elasticity, friction ratio between plunger and elastomer and plunger's square, that is in contact with elastomer. By variation of device's parameters, it is possible to change relation between elastic force $F_y$ and friction force $F_f$.

If the relation is right
$$\frac{F_y}{F_y + F_f} = 0.4$$

Then return of the plunger to the basic place is provided. If next relation is right
$$\frac{F_y}{F_y + F_f} = 0.6$$

Then necessary energy dissipation (nearly 60%) is provided (due to inner and outer friction forces). So elastic force, that impact the plunger, should be 40-60% from gross active strength for such construction. Examples of real draft gear of the rolling stock are shown on figure 3 and figure 4. Elastic element was made from polyurethane SKU-PFL-100.

Body and bolts were made from steel 40 H, plunger and screws- from steel 45. Plunger was made in three models with different cone ratio of sections. After first tests, one plunger was chosen with section angle $\alpha = 30^\circ$, that provided the best return to the basic position. From the results the efficiency of construction was showed. Working curve of the model is shown on figure 5.
During development of real draft gear next requirements were accepted (class T2 –GOST 32913-2014):

- Nominal energy intensity 100 kJ
- Maximal energy intensity 140 kJ
- Maximal compression force of the device 1600…2200 kN
- Maximal compression 120 mm

Device was developed with the result that elastic compression force of polyurethane was 40…50% from gross active strength. Other is friction force between polyurethane and plunger, walls of the body. In this case return of the plunger and body’s strength are provided. Gross force of static closure of draft gear should be 1600kN. If half of this force is elastic force compression of elastomer then elastic force should be 800kN.

For details of the test device was taken steel 40H. Device's body is cylinder with inner diameter D=180mm and outer diameter $D_0 = 240$mm. Cylinder doesn't have grooves and other changes of the form. This cylinder is loaded with tensile stresses in circumferential direction.

Outer diameter of plunger $d_o = 100$mm  
Basic diameter of plunger $d = 70$mm  
Extra plunger's square $A_{pl} = 44$ sm$^2$  
Diameter of retraining rod $d_r = 42$ mm
Computation pressure inside the body \( p^* = 180 \) MPa
Volume of elastic element 7300 sm³
Working curve of tested device is shown on figure 6.

![Graph: Working curve of tested draft gear](image)

**Figure 6.** Working curve of tested draft gear

Cover's tension bolts were made with cone head (cone ratio 1:20). Screws also were made with cone head (1:10). Cone ration was chosen with the result that during tightening screws, bolt was self stopped. Cone bolheads are in the buck cover, in such way that way don't make the device bigger and doesn't engage its mounting on the stop shoulder. Cone ratio was chosen bigger to decrease frictional moment between screw and cone basement. Screws also are inside the front cover not to increase the measurements of the device. Cone ares of the screws lets better distribute loading in threaded assemblage.

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Covers of the device are complicated, they stop the plugs. It provides plunger's plug's rail motion. Existence of different plugs provides an opportunity to change the diameter and plunger's configuration without changing the forms of basic details (covers, body, bolts). Cover's construction, that has cylindrical offsets, provides an opprtunity of preloading of the device, compressing it with two covers (without applying force to the plunger). Such way doesn't change the measurements of the device.

6. Conclusions
1. The absorbing devices of cars and locomotives by the principle of action represent the buffer devices perceiving external influence by energy and generating horizontal force.
2. Proceeding from need of increase in a resource of the car, the power characteristic of draft gear has to have bigger power consumption in the parameters «the maximum force-the maximum draught». Only devices of hydraulic type conform to this requirement.
3. The absorbing devices have to be economic in production and operation - devices of frictional type conform to this requirement.
4. The main objective of improvement of draft gear is development of devices with high quality of the power characteristic \( \alpha \geq 0,5 \) and low price (up to $500). To such requirements satisfy only the draft gear with an elastic element from solid constructional elastomer (polyurethane) can.
5. Tests of model and a prototype of draft gear of the autocoupling device of the rail vehicle have confirmed high loading of his details. Necessary power characteristics of the device are reached in that case when his steel details (in addition to elastomeric UE) carry out a role of springs, reserving elastic energy when loading.
6. Development of details of such form that distribution of tension in them was, whenever possible, more uniform was one of the main objectives. Then it is possible to count on the maximum power
consumption of these details. The specified principle managed to be realized for two main details (the cylindrical case and coupling bolts). As a result of tests all calculated parameters have been reached.

7. Dynamic tests of a prototype have to become the following stage that will allow to draw more thorough conclusions on applicability of this design for the rolling stock demanding the draft gear's installation of the class T2.

Acknowledgments. The research was partially supported by FASIE. The authors declare that there is no conflict of interest regarding the publication of this paper.

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