

Modernization of spring suspension of trailed vehicles

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Abstract. This article presents materials related to the development of a promising design of adaptive torsion bar suspension for trailed special and cargo vehicles, both trailers and semi-trailers, designed for the transportation of bulk and general cargo. A number of technical solutions, created at the level of inventions, are proposed, which make it possible to simplify the existing types of spring suspensions, reduce their metal consumption, improve the smoothness of trailed vehicles (TV) and increase their operational reliability due to the operation of suspensions in automatic mode. when they overcome micro and macro unevenness of the roadway. Analytical studies were carried out with the development of a design scheme and a methodology for calculating the design of an adaptive torsion spring for static and dynamic strength.

1 Introduction

The development of road transport in Russia is inextricably linked with the creation of the domestic auto industry, which to date has achieved certain success, both qualitatively and quantitatively [1]. Automotive rolling stock is divided into freight, passenger and special, and the first is used to transport a wide range of goods at various distances, reaching a thousand kilometers or more. A special place in this type of rolling stock is occupied by heavy road trains, consisting of tractor vehicles with a capacity of up to 300 kW and more and semi-trailers with a carrying capacity of 32,000 kg and more [2]. At the same time, one of the most important technological processes in the agro-industrial complex of the country is occupied by transport operations for the transportation of various agricultural goods, which are also carried out using both automobile and tractor trains [3].

It should be noted that an important structural element of trailers and semi-trailers is their running gear, which consists of a front and rear suspension, including leaf springs with axles and wheels equipped with shoe brakes driven by the pneumatic brake system of the tractor [4].

Despite their efficiency of use, both automobile and tractor trailers and semi-trailers have a number of significant disadvantages, such as their rather high metal consumption, insufficient reliability, significant labor intensity in maintenance and repair [5].

Comparing the design of spring suspension not only for tractor, but also for automobile trailers, both domestic and foreign, it can be seen that the overwhelming majority of them

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consist of leaf springs [6]. Despite their simplicity of design and efficiency of use, they have significant disadvantages such as - high metal consumption and rigidity, high labor intensity during repair work, relatively high cost [7]. At the same time, in the practice of this direction in mechanical engineering, not only leaf springs are used as spring suspension, but also helical compression springs, pneumatic and torsion springs.

The purpose of the study is the development of a promising design of an adaptive spring suspension for an autotractor-trailed vehicle, created at the level of the invention and equipped with an adaptive hydromechanical vibration damper.

2 Research methods

It is known [8-10] that when designing spring suspension of vehicles, both leaf springs and coil springs (springs) are calculated for strength and stiffness. The calculation method in this case is as follows. In the first case, the material for spring sheet steel is 55C2 and 60C2 steel in accordance with GOST 7419-55 and GOST 14959-69, followed by heat treatment to HB 363-432. The bending stress in the spring is determined by the dependence [11];

$$\sigma_H = \frac{3pl}{bh^2(i_k + i_c)} \leq [\sigma_H], \quad (1)$$

where: p – highest workload;

l – half the length of the spring in a straightened state;

b – leaf width of the spring;

h – leaf thickness of the spring;

i_k and i_c – number of root and stepped sheets;

Then calculate the calculated value of the deflection by the formula:

$$f = \frac{6p(l - \frac{a}{b})}{Ebh^3(3i_k + 2i_c)}, \quad (2)$$

and finally bending stresses of the main sheets are determined according to the dependence:

$$\sigma_{HC} = \frac{3pl\mu^3}{bh^2(i_k\mu^3 + i_{pr})}, \quad (3)$$

where: μ - the ratio of the thickness of the main sheets to the thickness of the stepped spring sheets;

i_{pr} – reduced number of leaves of the stepped part of the spring $i_{pr} = i_c + 1$.

After that, the normal bending stresses in the stepped part of the spring are calculated according to the dependence:

$$\sigma_{HC} = \frac{3pl}{bh_c^2(i_k\mu^2 + i_{pr})}, \quad (4)$$

In such calculations, as permissible stresses, taking into account dynamic loads, take $[\sigma]_{HC} = 800 - 900 \text{ MPa}$. Then set the spring stiffness:

$$F = \frac{P_{cm}}{f_{cm}} \text{ N/mm.} \quad (5)$$

Calculation of helical springs also begins with the choice of material, which is used as steel 60C2, 55C2, 65C2BA, with hardness HB 370-440 and determine the amount of deflection according to the dependence:

$$f = \frac{8D_{av}^2 n \cdot p}{d^4 G}, \quad (6)$$

where: p – work load;
 n – number of working turns;
 G – shear modulus equal to $8 \cdot 10^4$ MPa;
 D_{av} – average spring diameter;
 d – spring coil diameter.

The stiffness of the coil spring is determined by the dependence:

$$F = \frac{Gd^4}{8d^3 n}, \quad (7)$$

The coil springs are then calculated for strength according to the dependence:

$$\tau = \frac{k_1 8pD}{\pi d^3} \leq [\tau], \quad (8)$$

where: k_1 – coefficient set depending on the spring index $C = D/d$ by dependence: $k_1 = 1,25/C + 0,85/C^2 + 1/C^3$,

$[\tau]$ – permissible shear stresses taking into account the dynamic load are equal 750 – 850 MPa.

Analysis of a large number of bibliographic sources, as well as foreign and domestic patents, made it possible to develop a technical solution recognized by the invention (patent RU2475390 by the authors Slivinsky E.V., Savin L.A., Klimov D.N., Radin S.Yu., Gridchin D.V.), which makes it possible to simplify the design of the spring suspension of automobile trailers and semitrailers and, in particular, to use torsion bars instead of leaf springs according to the layout scheme similar to the independent suspension of the front and rear axles [12].

Fig. 1 shows a general view of the adaptive torsion spring in its longitudinal plane with a partial section and two views of it in sections BB and in AA.

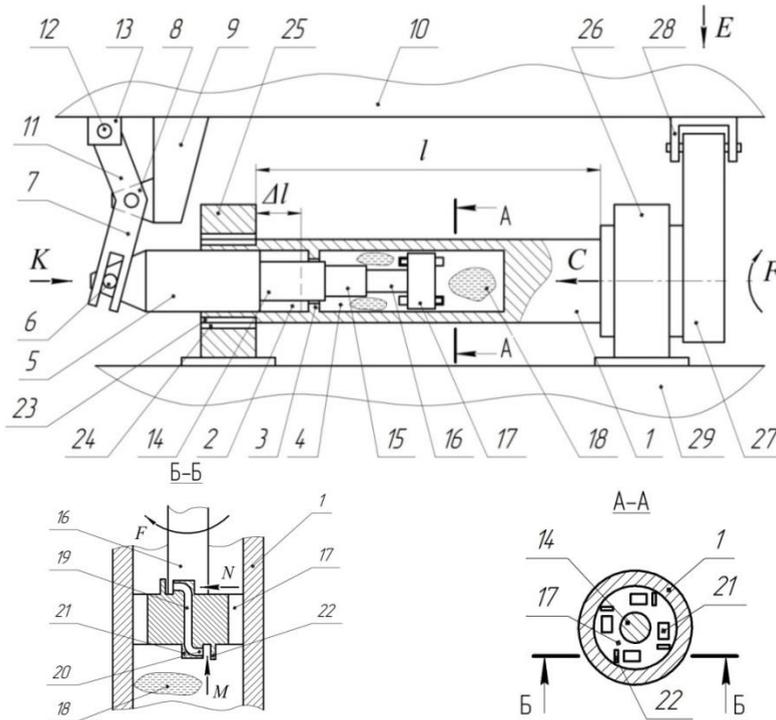


Fig. 1. Adaptive torsion spring by patent RU 2475390.

The adaptive torsion spring consists of a rod 1, in which a rectangular channel 2 is made, which passes after the partition 3 into a circular channel 4. In a rectangular channel 2 movably, in its longitudinal plane, a rectangular bar 5 is placed, which is hinged, with the help of pin 6, is connected to one of the arms 7 of the two-arm lever 8, pivotally mounted on the bracket 9, rigidly fixed to the body 10 of the semitrailer. The other arm 11 of the two-armed lever 8, also pivotally, with the help of the pin 12, is interconnected with the bracket 13 mounted on the body 10. The rectangular bar 5 is equipped with a rod 14 having different diameters 15 and 16, made along its length, which is rigidly attached to the piston 17, located movably in a circular channel 4 and filled with a working fluid 18. Piston 17 is equipped with horizontal throttling channels 19, passing into vertical throttling channels 20, made in tides 21, adjacent with a gap to the ribs 22. Rod 1 with one the side is equipped with splines 23, which are interconnected with the response splines 24 made in the support 25, and on the other - with the possibility of angular turns - is located in the bearing support 26 and is equipped with a lever 27 pivotally connected to the bracket 28 of the vehicle body 10. Supports 25 and 26 are rigidly fixed to the vehicle frame 29. The peculiarity of such a spring suspension arrangement is that the latter is placed between the body and the frame, for example, of a trailer in cases where the transported cargo or passengers require a sufficiently high comfort in terms of damping the vibrations of the bodies when the dynamic components of the loads are transferred to them. from the irregularities of the path. This is the so-called special rolling stock.

3 Results of the study

Let us consider an example of a preliminary calculation of the main parameters of the proposed adaptive torsion bar suspension, for example, for a MAZ-8926 car trailer. To do this, we will use the design scheme (see Fig. 2).

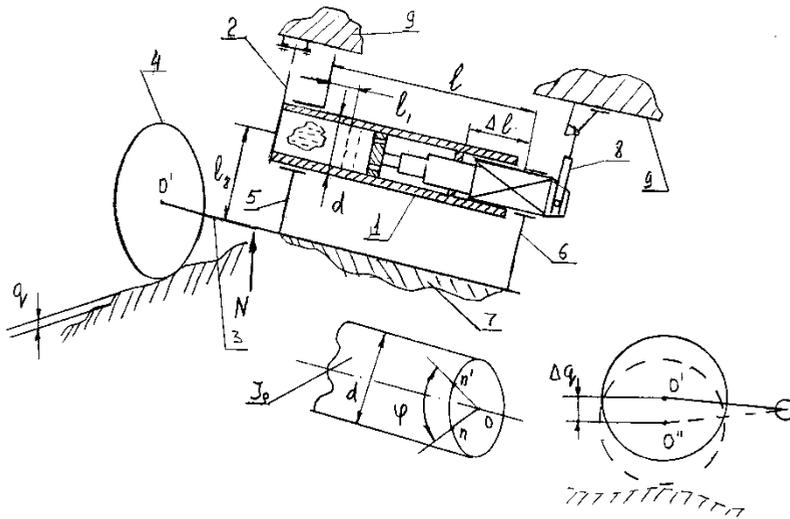


Fig. 2. Design scheme.

The design diagram of the adaptive torsion spring shows the torsion bar 1, which is pivotally connected with its lever 2 to the body 9. The axle 3 of the trailer wheel 4 is rigidly mounted on the trailer frame 7. The torsion bar 1 is also mounted on the trailer frame 7 with the help of supports 5 and 6. Inside the torsion bar 1, an adaptive hydromechanical damper is movably installed, the rod of which has the ability to move progressively and is controlled by a lever 8 pivotally attached to the body of the trailer 9 (the design of the adaptive spring is described in more detail above).

It is known [13] that such a trailer has the following characteristics: load-carrying capacity 8.0 t, unladen weight of the trailer 3.81 t, track 1970 mm, base 3700 mm, number of front wheels - 2 and rear wheels - 2.

Assuming that the four wheels of the trailer are subject to a static load $8,0+3,81 = 11,81t$, then a force will be applied to the balance bar of one torsion spring $N_{cm} = 11,81/4 = 2,95t$, while the static torque $M_t^C = N_{cm} l_1 = 2,95 \cdot 0,4 = 1,18t \cdot m$. According to the known data obtained during testing of car trailers, the coefficient of dynamics at a speed of 60 km / h on a dirt road is 1.2 [14]. Then we can assume that the working load (dynamic) for one spring set for our example will be $2,95 \cdot 1,2 = 3,54t$. Finally, we accept $N_\delta = 3,54 t$. Then, the moment applied to the torsion will be determined $M_t = N_\delta l_1 = 3,54 \cdot 0,4 = 1,42 t \cdot m$. Let us calculate the diameter of the torsion bar according to the dependence [15], assuming that the material for it is steel grade 65S2BA according to GOST 14959 - 79 with $\sigma_B = 1862MPa$:

$$d_T = \sqrt[3]{\frac{16M_t}{\pi[\tau]}} = \sqrt[3]{\frac{16 \cdot 1,42 \cdot 2 \cdot 10^6}{3,14 \cdot 931}} = \sqrt[3]{77719} = 43mm, \quad (9)$$

where $[\tau] = 0,5 \sigma_B = 0,5 \cdot 1862 = 931MPa$;

According to GOST 2590-88 for rolled products with a circular cross-section and, taking into account that the torsion bar is made hollow, we take its outer diameter equal to 80 mm, while the inner diameter, from the point of view of the torsion bar strength, is equivalent to the calculated 43 mm, we take it equal to 54 mm.

Let us determine the angle of twisting of the torsion bar under its static loading by the formula:

$$\phi_C = \frac{32M_l l}{G\pi \cdot d^4} = 10 \frac{M_l l}{G \cdot d^4} = 10 \frac{11,8 \cdot 10^6 \cdot 900}{8 \cdot 10^4 \cdot 43^4} = 0,38rad = 22^\circ, \quad (10)$$

where l – working length of the torsion bar equal to 900 mm.

Now let us determine the twist angle of the torsion bar under its dynamic loading, assuming that it is not equipped with a device for adjusting its working length l , described in patent RU 2278039:

$$\phi_D = \frac{32M_l l}{G\pi \cdot d^4} = 10 \frac{M_l l}{G \cdot d^4} = 10 \frac{14,2 \cdot 10^6 \cdot 900}{8 \cdot 10^4 \cdot 43^4} = 0,46rad = 27^\circ. \quad (11)$$

The movement of the body in these cases will accordingly be:

$$\Delta_C = 2 \cdot l_1 \sin \frac{\phi_C}{2} = 2 \cdot 400 \cdot \sin \frac{22^\circ}{2} = 800 \cdot 0,1908 = 152,64mm, \quad (12)$$

$$\Delta_D = 2 \cdot l_1 \sin \frac{\phi_D}{2} = 2 \cdot 400 \cdot \sin \frac{27^\circ}{2} = 800 \cdot 0,2334 = 186,72mm. \quad (13)$$

However, taking into account the dynamic coefficient equal to 1.2, the movement of the trailer in the vertical plane Δ_D will be less, since the working length of the torsion bar will decrease from the length l to length l_1 , for example 80 mm. In this case, the angle of twisting of the torsion in the dynamics will be equal to:

$$\phi_D = \frac{32M_l l^1}{G\pi \cdot d^4} = 10 \frac{M_l l^1}{G \cdot d^4} = 10 \frac{14,2 \cdot 10^6 \cdot 820}{8 \cdot 10^4 \cdot 43^4} = 0,42rad = 24^\circ. \quad (14)$$

and the movement of the body in this case will be:

$$\Delta_D = 2 \cdot l_1 \sin \frac{\phi_D}{2} = 2 \cdot 400 \cdot \sin \frac{24^\circ}{2} = 800 \cdot 0,2079 = 166,32mm, \quad (15)$$

$$\tau = \frac{16M_l}{\pi d^3} = \frac{16 \cdot 14,2 \cdot 10^6}{3,14 \cdot 43^3} = 56,9MPa \leq 931MPa. \quad (16)$$

therefore, the strength of the torsion bar is ensured.

Let us determine the torsion bar stiffness under the action of both a static load and a dynamic one according to dependencies:

$$F_C = \frac{N_C}{\Delta_C} = \frac{29500}{152,64} = 193,26N / mm, \quad (17)$$

$$F_D = \frac{N_C}{\Delta_D} = \frac{35400}{186,72} = 189,58N / mm. \quad (18)$$

It can be seen that an increase in the torsion bar stiffness will allow creating a resistance force of the above N_b and, thereby, damping body vibrations in this range of its loading.

Considering the variety of road train designs used in the Russian Federation and abroad, in order to automate the calculation in each specific case of the parameters characterizing the design of the torsion, a program was developed in the Delphi language, in which, by varying the initial data, it is possible to calculate the necessary parameters by engineering calculation of the proposed technical solution.

4 Conclusion

Analysis of both domestic and foreign designs of autotractor trains shows that the latter have a significant drawback, which consists in the fact that spring suspensions are mainly used in their designs, including sets of leaf springs. Such spring suspension is complex in design, rather metal-consuming and has a generally constant bending stiffness, which significantly affects the smooth running of trailers and semi-trailers that make up the trailer links of autotractor trains. Taking into account such disadvantages, at the level of inventions (RU2475390, RU2499686, RU2499684, RU2499687, RU2499685, etc.), technical solutions of torsion resistors have been developed, allowing in automatic mode during the movement of an autotractor train with different speeds on roads with different micro- and macro-profile, to change its torsional stiffness and, thereby, to reduce the dynamic loads arising in the bearing systems of their trailed links, as well as to increase the smoothness of the movement of the latter.

To establish rational geometrical and kinematic parameters of one of these devices, made, for example, according to patent RU2475390, a design scheme and a physical model have been developed, and a method has been proposed for their calculation, both for conducting preliminary engineering calculations and for carrying out research and development and R&D, which ultimately makes it possible to give not only an appropriate assessment of the power loading and vibrations of the trailer link of an autotractor train, but also to establish the criteria for the operability of the proposed technical solution with the aim of its possible further implementation into practice.

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