ICCPT 2019: Current Problems of Transport: Proceedings of the 1st International Scientific Conference, May 28-29, 2019, Ternopil, Ukraine

Suspension of a car with nonlinear elastic characteristics based on a four-link lever mechanism

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Abstract: The possibility of creating a suspension of a car with a nonlinear elastic characteristic based on a four-link lever mechanism is shown. The kinematics and dynamics of the mechanical suspension system are analyzed. The kinematic laws and distribution of forces in the nodes of the suspension are found. The dependence of general rigidity on the stiffness of the main and additional elastic elements is obtained for obtaining the required nonlinear elastic characteristic. An example of a given nonlinear elastic characteristic is presented.

Keywords: suspension of a car, nonlinear elastic characteristic, four-link lever mechanism.

1. Introduction

During operation of cars periodically its sub sorted mass as a result of change of payload. Most significantly this is noticeable for the rear axle. If for cars the change of load on the rear axle is 45-60%, for buses – 200-250%, then for trucks the change of load reaches 250-400%.

Suspension of a car with a linear elastic characteristic is not able to provide the proper smoothness of the course, as when changing the load, the static arrows of the deflection and, as a result, the frequency of oscillations change.

From the theory of the automobile it is known that in order to maintain the required level of internal frequency of oscillations with variable load, it is necessary to ensure the stability of the static deflection z_0 change in the stiffness of the suspension.

Such a requirement is satisfied by the nonlinear elastic characteristic of the suspension, which corresponds to the condition [1,2,6]:

$$\frac{R_z}{c_p} = z = z_0 = const \tag{1}$$

,

where R_z – dynamic load of the suspension; c_p – the rigidity of the suspension is given; z– vertical displacement of the submerged mass.

In turn, this condition will be fulfilled, as is known [1], if the characteristic of the suspension changes according to the law of the indicator function:

Peer-review under responsibility of the Scientific Committee of the 1st International Scientific Conference ICCPT 2019: Current Problems of Transport

$$R_z = R_{z0} \cdot e^{(\frac{z}{z_0}-1)}, \tag{2}$$

where R_{z0} – static load of the suspension N.

It should be noted that ensuring these conditions will lead to constant frequency fluctuations, regardless of the load change, with the following assumptions:

- all the payload falls on the rear axle, which is almost performed for trucks and passenger cars;
- the oscillations of the submerged mass above the front and rear axles do not depend on each other. This is possible with the mass distribution factor $\varepsilon = 0,85 \div 1,2$ [1,2,4-6].

Known ways to achieve non-linearity by using in the spring suspensions or additional elastic elements with a progressive characteristic [1,3,4]. But such systems do not provide a constant level of smoothness at various values of the submerged mass. Systems with additional elastic elements connected at the values of the submersible mass close to the nominal load-carrying capacity have poor smoothness at the unloaded condition of the car.

To the fullest extent, condition (2) can provide an adaptive suspension in which the damping degree of the shock absorber is given in accordance with the road surface condition, the parameters of motion, and may also change to the current value of the submersible mass [7-12]. In adaptive suspensions used in modern cars, the degree of damping of the shock absorber varies in two ways: by means of electromagnetic valves or by means of magnetorheological liquids.

By the first way work the well-known suspensions Adaptive Variable Suspension, AVS from Toyota; Continuous Damping Control, Opel CDS; Adaptive Chassis Control, DCC from Volkswagen; Electronic Damper Control, EDC from BMW (active suspension Adaptive Drive); Adaptive Damping System, ADS from Mersedes-Benz (Airmatic Dual Control Pneumatic Suspension) [13-17].

Shock absorbers based on magnetorheological fluids are used in MagneRide suspensions from General Motors (cars Cadillac, Chevrolet); Magnetic Ride by Audi [18].

With significant advantages, the adaptive suspension due to the complexity of the design of the shock absorber, the presence of trace elements (sensors), computer control has a high cost, both its own and maintenance, of course, restricts its use.

That is why occurs the task of designing simple, relatively cheap suspensions with a nonlinear characteristic, close to the condition (2) for cargo and passenger cars of class N1, B1.

2. Materials and Methods

To obtain a nonlinear elastic characteristic we use the kinematic scheme of the suspension on the basis of a four-link lever mechanism (Figure 1) [19, 20].

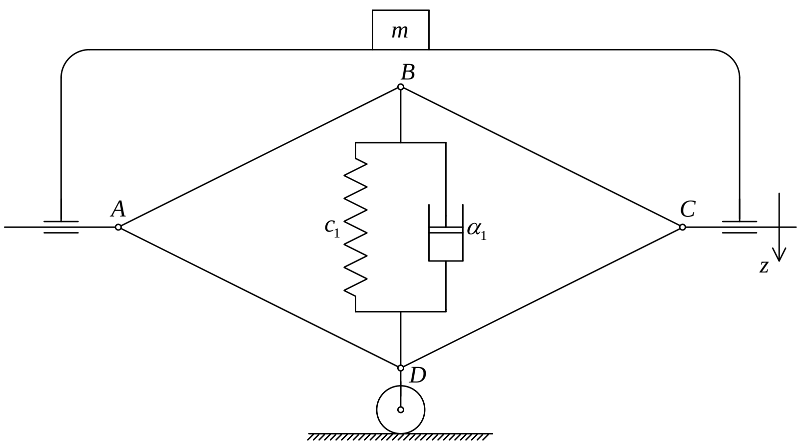

Figure 1. Kinematic scheme of the suspension on the basis of a four-link lever mechanism.

Under this scheme the sprung mass rests through longitudinal guides on the side hinges A and C of the four-link lever mechanism ABCD. The elastic and damping elements are fixed between the upper B and the lower D joints of this mechanism. In this case, the lower hinge is based on unscattered mass.

Let's consider the kinematics of such a scheme.

If you immediately fix the lower hinge D of the four-link lever mechanism and, for the submersible mass, set the vertical displacement *z* (Figure 2), then the upper hinge B moves to *2z*, since each of the vertical half diagonals of the diamond ABCD will deform to *z*.

This leads to the fact that the magnitude of the oscillations of the submerged mass will be doubled between the upper and lower hinges of the quadrilateral lever mechanism, and, therefore, the elastic element will be perceived as a double deformation *ΔBD = 2z*. This allows you to use an elastic element with less rigidity in the suspension while maintaining the same level of internal frequency of oscillation of the submerged mass.

Let's find the horizontal displacement *y* of the side hinges in relation to the vertical displacement of the sub-mass *z*. Consider the scheme in Figure 2.

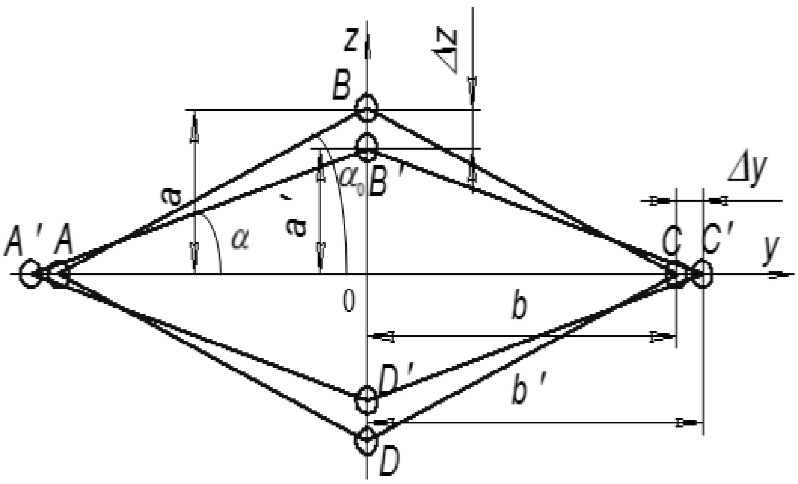

Figure 2. The scheme of finding the horizontal displacement y of the side hinges relative to the vertical displacement of the sprung mass *z*.

The ABO triangle (Figure 2) formed by the joints A and B of the lever quadrilateral ABCD and the point O of the intersection of the diagonals of the diamond corresponds to the state of the static deflection of the suspension. The A1B1O triangle corresponds to the current suspension condition with oscillations. The value BB1 corresponds to the deformation of the vertical semia dionon of the diamond and is equal to *z*. The value A1A is the horizontal displacement of the side hinge Δy. The angle α_0 is the angle between the horizontal diagonal of the diamond AC and the side AB and corresponds to the state of the static deflection z0, $<\alpha=<OA_1B_1$ which is the angle corresponding to the state of the deformed suspension by the value of z. Indicate the semi-diagonal $OB=a_0$, $OB_1=a$, $OA=OC=b_0$, $OA_1=OC_1=b$, side of the diamond $AB=l$.

From the triangle ABO: $\alpha_0 = l\cdot\sin\alpha_0$, $в_0 = l\cdot\cos\alpha_0$.

From the triangle A1B1O: $\alpha = l\cdot\sin\alpha$, $в = l\cdot\cos\alpha$.

Then $$z=\alpha_0-\alpha=l(\sin\alpha_0-\sin\alpha), \tag{3}$$

from which occurs $$\sin\alpha=\sin\alpha_0-\frac{z}{l} \tag{4}$$

$$y=в-в_0=l(\cos\alpha-\cos\alpha_0). \tag{5}$$

Using equation (4), we find $cos\alpha$ and substitute in equation (5):

$$y = l(\sqrt{1-(\sin\alpha_0 - \frac{z}{l})^2} - \cos\alpha_0) \tag{6}$$

Expression (6) is a nonlinear functional dependence of the horizontal displacement in the side hinges A and C of the ABCD diamond from the vertical deformation of the suspension *z*.

The nonlinearity of the function $y = f(z)$ allows the use in the suspension of additional elastic elements with rigidity c_2, placed horizontally along the axis of the displacement of the side hinges of the four-link lever mechanism (Figure 3) to obtain a nonlinear elastic characteristic.

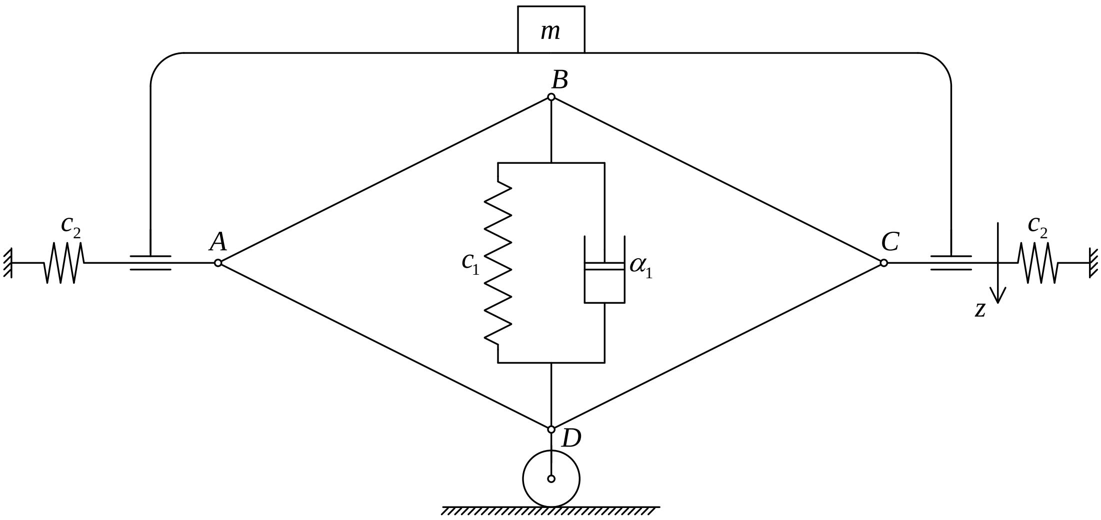

Figure 3. Kinematic scheme of the suspension on the basis of a four-link lever mechanism with additional elastic elements.

3. Discussion

Consider the dynamics of the car suspension on the basis of a four-link lever mechanism with the additional elastic elements c_2 shown in Figure 3.

On Figure 4 are shown the forces acting in the suspension arrangement, assuming that there is no friction in the kinematic pairs.

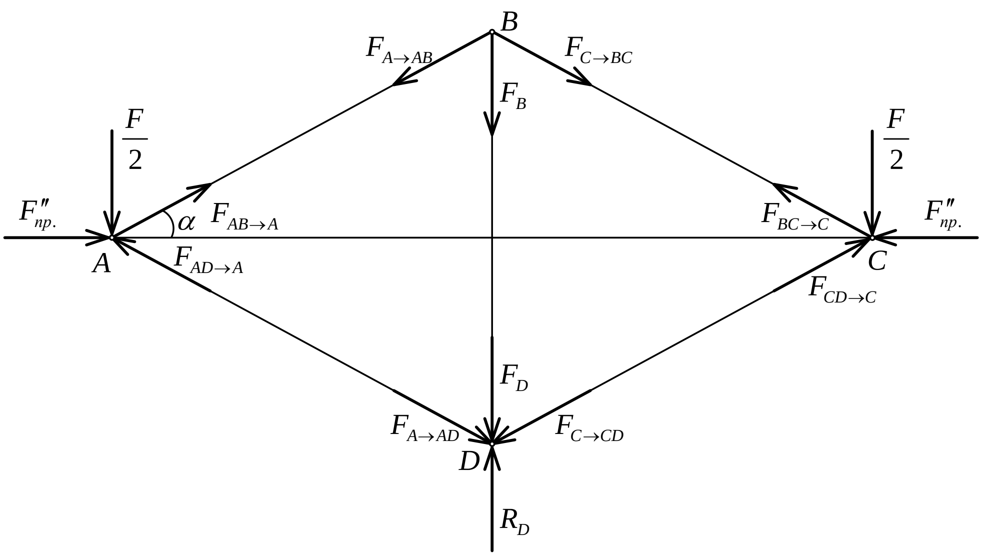

Figure 4. Forces acting in the device suspension.

External force $F = c \cdot z$, where c – overall rigidity of the suspension. Force F, given the symmetry of the circuit, it is transmitted to the side hinges A and C, as a force $\frac{F}{2}$. At the same time, as a result of the action of the nodes A and C, respectively, on the rods AB and BC there are reactions $F_{AB\to A}$ and $F_{BC\to C}$, but in the rods AD and CD – the reaction $F_{AD\to A}$ and $F_{CD\to C}$. F_c' and F_c'' – forces of the strength of the elasticity of the main vertical, located between the joints B and D, and additional horizontal elastic elements with stiffenings c_1 and c_2 accordingly.

Having considered the equilibrium of the hinge A (Figure 5), we find that

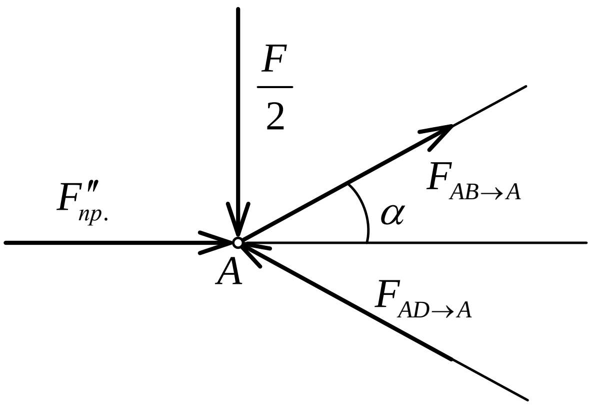

Figure 5. The equilibrium of the hinge A.

$$F_{AB\to A} = \frac{\frac{F}{2} - F_c'' \cdot tg\alpha}{2 \cdot \sin\alpha} \tag{7}$$

Taking into account that $\bar{F}_{AB\to B} = \bar{F}_{A\to AB} = -\bar{F}_{AB\to A}$ from the equilibrium of the hinge B (Figure 6) we have:

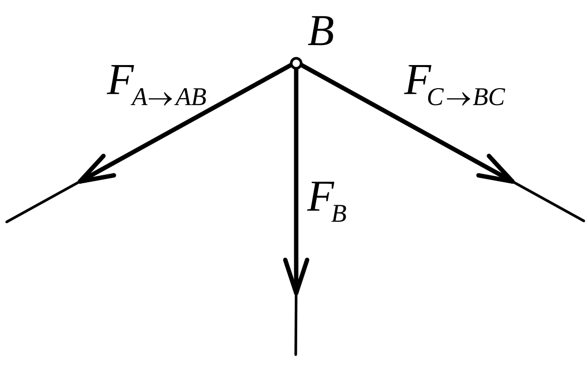

Figure 6. The equilibrium of the hinge B.

$$F_c' = 2 \cdot F_{AB\to A} \sin\alpha \,. \tag{8}$$

By substituting (7) into (8), we obtain:

$$F_c' = \frac{F}{2} - F_c'' \cdot tg\alpha \,, \tag{9}$$

where

$$\frac{F}{2} = F_c' + F_c'' \cdot tg\alpha .$$

Taking into account that at a static deflection of suspension brackets $F = cz_0$, $F_c' = c_1 \cdot 2z_0$, $F_c'' = c_2 y$, we have:

$$\frac{cz_0}{2} = c_1 2z_0 + c_2 y \cdot tg\alpha$$

$$c = 4c_1 + \frac{2c_2}{z_0} y \cdot tg\alpha , \qquad (10)$$

where y is based on formula (6).

Dependence (10) is a nonlinear dependence of the total stiffness of the suspension on its vertical deformation z.

On Figure 7 is shown an example of an elastic characterization of a suspension on the basis of a four-link lever mechanism with additional elastic elements, obtained for the parameters: sprung mass m = 100 kg, c1 = 3600 N/m, c2 = 8000 N/m.

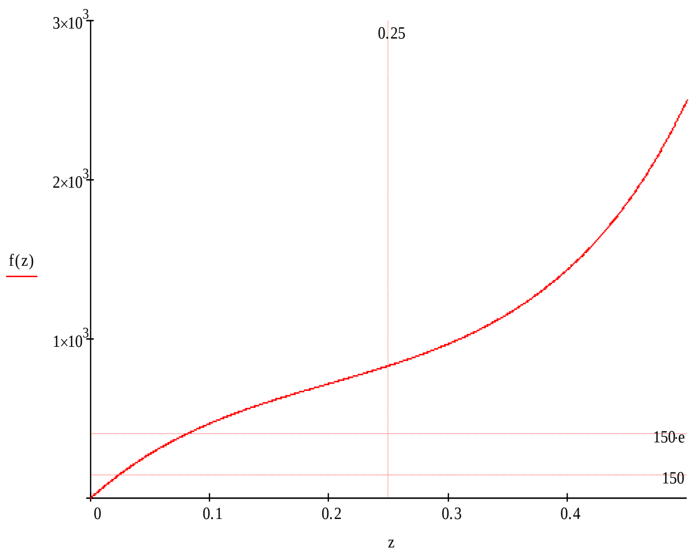

Figure 7. Elastic characterization of a suspension on the basis of a four-link lever mechanism with additional elastic elements.

4. Conclusions

The scheme of a suspension of a vehicle on the basis of a four-link lever mechanism is investigated, which involves the use of a main elastic element that perceives the double vertical movement of the sprung mass and additional elastic elements that perceive the horizontal deformation of the lever mechanism. On the basis of the kinematic analysis of the scheme, a nonlinear relationship between vertical suspension deformation and horizontal deformation of the four-link lever mechanism is established.

On the basis of dynamic analysis, the distribution of forces in the scheme is determined and a nonlinear dependence of the general stiffness of the suspension and its vertical deformation is established. This allows to obtain a nonlinear elastic characteristic of the suspension and, in turn, can provide a constant smoothness of motion in the variable sprung mass.

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