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IMPROVED METHOD OF CALCULATING THE RATIONAL DIMENSIONS OF PARTS OF HYDRAULIC SHOCK ABSORBERS OF ELECTRIC TRAINS AT HIGH SPEEDS

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ABSTRACT

The article presents an improved method for calculating the rational dimensions of parts of hydraulic shock absorbers of electric trains at high speeds.

The increase in passenger and freight traffic is one of the most important tasks for the economic and political independence of the Republic of Uzbekistan, which, in turn, entails an increase in the fleet of vehicles, including locomotives and high-speed trains. Their use is associated with the use of curvilinear systems of complex profiles in the suspension of vehicles and various types of absorbers, air springs, and hydraulic shock absorbers; these elements increase the smoothness of movement of the rolling stock and improve the strength and elastic-dissipative properties of the suspension in order to reduce dynamic effects on the track and transported cargo [1].

Modern programs for modeling and calculating the rolling stock of electric vehicles subjected to operational loads contribute to the creation of a more accurate and extensive specification of requirements for spring suspension and vibration damping systems, as well as the development of new methods for calculating hydraulic shock absorbers of electric rolling stock [2, 3, 4].

The purpose of this article is to develop an improved method for calculating the rational dimensions of parts of hydraulic shock absorbers of high-speed electric trains; the method consists of 6 stages, the numerical solution for which is conducted in the *MathCad 15 programming environment*. The method the authors propose is based on publications [5, 6].

Stage 1. OPTIMIZATION OF THE PROCESS OF FILLING THE HYDRAULIC SHOCK ABSORBER (HSA).

1.1. The fluid flow in the bypass part from the inlet section to the piston according to the Bernoulli equation [3, 4] includes hydrodynamic and velocity pressure, dissipative losses due to friction and local resistance, estimated by the corresponding generalized coefficients $\boldsymbol{\xi}_{\text{Т}}$ and $\boldsymbol{\xi}_{\text{М}}$, where $\boldsymbol{di}$, $\boldsymbol{\ell i}$ are the diameters and lengths of the channel; $\boldsymbol{F}_{\text{П}}$, $\boldsymbol{F}_{\text{К}}$ are the sections of the piston and inlet channel

$$Z_{\text{П}} - Z_{\text{В}} + \frac{\text{Р}_{\text{П}} - \text{Р}_{\text{В}}}{\gamma} + \frac{\upsilon_{\text{П}}^2 - \upsilon_{\text{В}}^2}{2g} = (\boldsymbol{\xi}_{\text{Т}} + \boldsymbol{\xi}_{\text{М}}) \frac{\upsilon_{\text{П}}^2}{2g} \quad ; \quad (1)$$

$$\boldsymbol{\xi}_{\text{Т}} = \sum_{i=1}^{i} \lambda_i \frac{\boldsymbol{\ell i}}{\boldsymbol{di}} \left(\frac{\boldsymbol{F}_{\text{П}}}{\boldsymbol{F}_{\text{К}}}\right)^2 \quad ; \quad \boldsymbol{\xi}_{\text{М}} = \sum_{k=1}^{i} \boldsymbol{\xi}_{\boldsymbol{k}} \frac{\boldsymbol{\ell i}}{\boldsymbol{di}} \left(\frac{\boldsymbol{F}_{\text{П}}}{\boldsymbol{F}_{\text{К}}}\right)^2 . \quad (2)$$

1.2. The pressure drop $\Delta\text{Р} = \text{Р}_{\text{П}} - \text{Р}_{\text{В}}$ for the harmonic law of piston motion with the selection of static components ΔP^0, Δz^0 is derived from the Bernoulli equation [3, 4] and is determined by the following formula

$$z_{\text{П}} = a \sin \omega t \quad ; \quad (3)$$

$$\dot{z_B} = \delta_B \dot{z_{\text{П}}} \quad ; \quad \delta_B = \frac{\boldsymbol{F}_{\text{П}}}{\boldsymbol{F}_{\text{В}}} \quad ; \quad (4)$$

$$\Delta\text{Р} = \Delta P^0 - \gamma \Delta z^0 + \gamma_{\text{В}}(\delta_B - 1)\, a \sin \omega t + \frac{a^2\omega^2}{2g}(\delta_B^2 - 1 + \boldsymbol{\xi}_{\text{Т}} + \boldsymbol{\xi}_{\text{М}})(\cos \omega t)^2 \quad , \quad (5)$$

for $\delta_B \approx 1$ and $z_{\text{П}} = a$;

$$\Delta\text{Р}_{\text{Т}} = \Delta P^0 - \gamma \Delta z^0 + (\boldsymbol{\xi}_{\text{Т}} + \boldsymbol{\xi}_{\text{М}}) \frac{a^2\omega^2}{2g} \quad , \quad (6)$$

where $\upsilon_{\text{П}} = \dot{z_{\text{П}}} = a\omega \cos \omega t$ is the HSA piston speed; $\delta_B = \frac{\boldsymbol{F}_{\text{П}}}{\boldsymbol{F}_{\text{В}}}$ is the section ratio; $\boldsymbol{\xi}_{\text{Т}}$ and $\boldsymbol{\xi}_{\text{М}}$ are the coefficients of friction resistance and local resistance.

1.3. To improve the filling process, it is necessary to reduce the dynamic drop $\Delta\text{Р}_{\text{Д}}$, i.e., to reduce frictional resistance $\boldsymbol{\xi}_{\text{Т}}$ and local resistance $\boldsymbol{\xi}_{\text{М}}$.

1.4. An important condition for filling is to ensure that the fluid level is no lower than the bypass device. The working fluid must fill the high-pressure cavity and most of the recuperative cavity, providing the fulfillment of the following condition

$$V = (F_{\text{Ц}} - F_{\text{Ш}})\, \boldsymbol{\ell}_{\text{П}} + F_{\text{Р}}\, \boldsymbol{\ell}_{\text{Р}} \quad , \quad (7)$$

where $F_{\text{Ц}}$, $F_{\text{Ш}}$, $F_{\text{Р}}$ are the cross-sectional areas of the cylinder, rod, and recuperative cavity, respectively; $\boldsymbol{\ell}_{\text{П}}$, is the largest stroke of the piston and $\boldsymbol{\ell}_{\text{Р}}$ is the maximum length of the tank.

Fluid leaks during the period between overhauls are allowed no lower than the lowest level V_{min} determined by testing the absorber on the bench under conditions close to operational ones

$$\Delta V = V - V_{min} \quad . \quad (8)$$

1.5. The filling process is improved by removing air from high-pressure cavities by placing orifices at the highest points of the cavities and preventing fluid from churning and foaming.

1.6. An essential factor for effective filling is to ensure the maximum bypass section and minimum inertia (mass) of the bypass disc (valve):

$$\boldsymbol{F}_{\text{П}} \to \boldsymbol{max}\, , \ \boldsymbol{m}_{\text{Д}} \to \boldsymbol{min}\, . \quad (9)$$

1.7. The filling process deteriorates with an increase in the oscillation frequency due to a sharp increase in the inertial and high-speed components of the resistance to the fluid flow and a decrease in the filling duration:

$$z_{\text{И}} = \frac{\upsilon_{\text{П}}^2}{2g} \ ; \ z_{\upsilon} = (\boldsymbol{\xi}_{\text{Т}} + \boldsymbol{\xi}_{\text{М}}) \frac{\upsilon_{\text{П}}^2}{2g} \ ; \quad (10)$$

$\upsilon_{\Pi} = a\omega$, $t_H = \frac{\tau}{2} = \frac{\pi}{\omega}$. (11)

Stage 2. OPTIMIZATION OF THE PROCESS OF HYDRAULIC SHOCK ABSORBER COMPRESSION.

2.1. The unsteady motion of fluid in the hydraulic damper according to the Bernoulli equation [3, 4] during the throttle process depends on the following parameters: fluid density ρ and the flow rate of the fluid velocity averaged over the flow section, the coefficient of irregularities in distribution of velocities in the throttle channel and the clearances, the coefficient of friction resistance on a straight line section of the *i*-th channel of length $\boldsymbol{\ell_i}$ and diameter $\boldsymbol{d_i}$ **,** the coefficient of the *i*-th local resistance, at which the speed changes in magnitude and direction:

$\mathrm{P} = \rho\upsilon_{Ж}^2(2k^2)^{-1}$, (12)

where $k^{-2} = \alpha + \sum_{i=1}^{i}\boldsymbol{\xi}_{\mathrm{T}\boldsymbol{i}}\frac{\ell_i}{d_i} + \sum_{k=1}^{k}\boldsymbol{\xi}_{\boldsymbol{MkL}}$, (13)

$\rho = \frac{\gamma}{g}$, k , $\upsilon_{Ж}$, $\alpha \geq 1$,

$\boldsymbol{\xi}_{\mathrm{T}\boldsymbol{i}}$ and $\boldsymbol{\xi}_{\mathrm{M}\boldsymbol{i}}$ are the friction resistance coefficients on the straight section of the *i*-th channel and *i*-th local resistance.

2.2. The main losses of the throttled fluid occur at a sudden flow expansion from the cross section of the throttle channel and leakage $\boldsymbol{f_i}$ to the effective area of the piston $\boldsymbol{F_a}$, proportional to the square of the velocity and outflow. If the cross-section of the throttle channel $f_1 = f_Д$ is substantially higher than the cross-section of leakage $\boldsymbol{f_i}$ then formula (12), given in subparagraph *2.1*, can be considered as an outflow through the throttle channel (*i = 1*); and an outflow through the leakage is taken into account by corresponding coefficient *ξ*.

$f = \sum f_i$ for $F \gg f$; $k^{-2} \approx 1{,}5$. (14)

The resistance in a long throttle channel or in the clearance between the rod and the guide is proportional to the first degree of the fluid outflow rate and is determined by:

- dynamic coefficient of viscosity *μ;*
- the length of the throttle channel $\boldsymbol{l}_Д$ or the clearance $\boldsymbol{l}_3$;
- throttle radius $\boldsymbol{r}$ or annular clearance *Δ* .

$P = 8\mu\, lg\upsilon_{Ж}r^{-2}$; $P = 12\mu l_3\upsilon_{Ж}\Delta^{-2}$. (15)

2.3. Formula (13) can also be represented as a flow rate in the cylinder, depending on the effective area and speed of the piston. In the throttle mode of operation for $\boldsymbol{f}_Д \gg \boldsymbol{f_i}$ the fluid flow rate is determined by the cross-section of the throttle channel. Therefore, the force on the piston rod depends on the fluid volume, its density, sections $\boldsymbol{f_i}$, throttle channel, clearances and safety valve.

$Q = V = \mathrm{F}_Э\upsilon = \mathrm{f}\upsilon_{СЖ}$; (16)

$Q = kf\sqrt{\frac{2P}{\rho}}$; (17)

$f = \boldsymbol{f}_Д\boldsymbol{\xi}^{-1}$; (18)

$\boldsymbol{P} = \boldsymbol{p}\,\mathrm{F}_Э$; $\mathrm{V} = \mathrm{F_a}(\mathrm{x} - \mathrm{x_0})$. *(19)*

2.4. The quadratic viscous resistance of the absorber, taking into account the continuity of the hydraulic flow, is determined by the sections and velocities of the working fluid.

The calculated cross-section of the throttle channel and clearances is found for known parameter *λ* , given area $\boldsymbol{F_a}$ and fluid density ρ

$\acute{\mathrm{Fa}}\ \upsilon = \mathrm{f}\upsilon_{СЖ}$*;* *(20)*

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$P_\gamma = \gamma\, \upsilon^2 = \rho\, Fa$; *(21)*

$P_\gamma = \gamma\, f^2 \upsilon_{Ж}^2 F_a^2$; *(22)*

$f^2 = \rho\, F_a^3 / 2\gamma k^2$; *(23)*

$f = \sum f_i$; $f_1 = f_Д$. (24)

2.5. A change in parameter γ due to a change in sections $\boldsymbol{f_i}$ can be estimated using the Lagrange theorem. From here, for the permissible change in $\Delta\boldsymbol{\gamma}$, the tolerances for clearances in the joints (compactions) of the hydraulic shock absorbers are found.

2.6. The influence of the cross sections of various leakages on the resistance parameter $\boldsymbol{\beta}$ is different and linear for small ranges. Clearances during operation must be limited. Due to the leakage of the safety valve, the force on the hydraulic damper rod is approximated by a linear function of the amplitude of the piston speed

$P_T = \beta\, a\, \omega$. (25)

2.7. At an increase in the oil viscosity, the resistance of the absorber increases, but less than the calculated values due to the deterioration of the filling process. The degree of resistance nonlinearity increases.

2.8. At an increase in the mass of the safety valve, the restriction forces exceed the calculated ones. Impact loads are taken rigid. A slow and loose fit of the bypass valve disc leads to a loss of the working stroke and a decrease in the damper resistance. Therefore, light discs are tightened by a weak spring to the seat.

2.9. Incomplete filling and air ingress into the high-pressure cavity lead to the loss of the working stroke, a decrease in the resistance force, and an increase in the degree of its non-linearity.

Stage 3. CALCULATION OF THE WEAR INTENSITY OF THE LIMITING FRICTION PAIR AND THE OVERHAUL LIFE OF HSA.

The scheme to determine the overhaul life of the hydraulic damper KVZ-LIIZhT by the wear of conjugated pairs is shown in Figure 1, namely, the dependence of the wear of rod $\Delta_{Ш}$ and guide $\Delta_{Н}$ on the service life of the hydraulic shock absorber; here

$\Delta = \Delta_{Ш} + \Delta_{Н}$. (26)

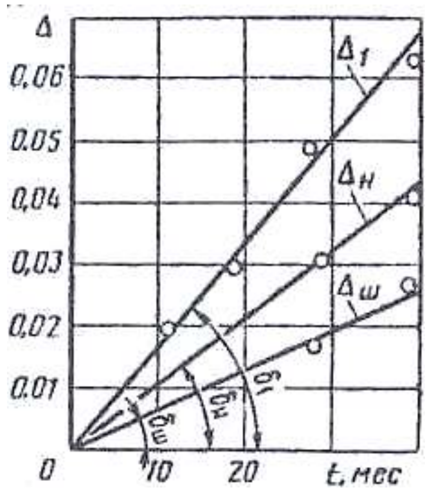

Figure 1. Dependence of wear of rod $\Delta_{Ш}$ and guide $\Delta_{Н}$ on the service life of the hydraulic shock absorber, here $\Delta = \Delta_{Ш} + \Delta_{Н}$.

The calculation of the wear intensity of the limiting friction pair and the overhaul life of the hydraulic shock absorber is conducted according to the following formulas

$\delta_1 = \frac{\Delta_1}{t} = 0{,}0017$ mm/month; (27)

$T = \Delta_{1a} - \Delta_{13}\, \delta_1^{-1} = (0{,}06 - 0{,}02)\, 0{,}0017^{-1} \approx 24$ month. (28)

Stage 4. CALCULATION OF THE LINEARIZED PARAMETER OF HSA RESISTANCE.

The linearized parameter of HSA resistance should be within $\beta = 50 \pm 130\ \ kN\ f/m$, depending on the sections of leakage $\beta = 50\ \pm 130\ \ kN\, f/m$

$\Delta\beta = 50 \pm 0{,}25\ \beta_0$, (29)

where $\beta_0 = 105\ \ kN\, f/m$.

Depending on the type of repair $\Delta\ \beta_З = 20$, and $\Delta\ \beta_Д = 30\ \ kN\, f/m$.

The change in $\Delta\beta$ from the cross section of the throttle channel and leakages is determined by the following formula

$\Delta\beta = 2Bf^{-3} \sum_{i=1}^{N} \Delta f_i$, (30)

where $B = 8\rho\ F_Э^3\ \frac{a\omega}{6\pi k^2}$, and f_i is the section of the throttle and leakages.

Factory tolerances for a limiting pair give a difference in the resistance parameter $\Delta\beta\ \le 50$, rational - $\Delta\beta\ \le 20$ and for depot repairs $\Delta\ \beta_Д \le 35$. Permissible derating in operation

$\Delta\beta = 50\ \pm 25\beta_0$, (31)

from the maximum value is timed for the next repair.

Stage 5. DETERMINATION OF THE REPAIR PERIOD OF HSA BY THE WEAR OF CONJUGATED PAIRS.

The overhaul period $\boldsymbol{L} = \Delta\boldsymbol{\beta}\ \boldsymbol{\delta}^{-1}$ is determined by the average intensity of the decrease in parameter $\boldsymbol{\beta}$ per unit run. To increase the run between overhauls, it is appropriate to install sealing rings in the guides or adjust the cross-section of the throttle channel while monitoring the hydraulic shock absorber in operation.

Stage 6. DETERMINATION OF THE OVERHAUL LIFE OF THE HYDRAULIC DAMPER.

The resource of the hydraulic vibration damper (Figure 1) is determined by the wear of the limiting rubbing pair. In the absence of a sealing ring in the guide of such a pair, a rod-guide is used. The overhaul life is equal to the ratio of the total increase in the clearance permissible in operation $\Delta_{1m} = \Delta_{1a} - \Delta_{13}$ to the wear rate per unit time

$\delta_1 = \frac{\Delta}{t}$, i.e. $T = \frac{\Delta_{1m}}{\delta_1}$.

Wear and damage to the parts of the hydraulic vibration damper and their reliability are determined by the dynamic loading and operating conditions of the limiting parts.

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