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Generalized dynamic model of hydrodynamic vibration dampener subject to viscous damping

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Abstract. Use as an elastic body of the working fluid improves the efficiency of the hydraulic damper, reliability, and durability, in this case, the issues of improving the reliability of existing locomotives hydraulic dampers by upgrading separate the components for major repairs with the extension of the useful life are relevant. To increase the efficiency of damping dynamic vibrations and reduce the load on the support units and running gear elements, a modernized design of a hydro friction damper with improved damping properties was developed, for which an application was filed for the Patent of the Republic of Uzbekistan for Invention No. IAP 2021 0002.

1 Introduction

In terms of self-financing and connection with the transition to a market economy, one of the most important tasks for our Republic's economic and political independence is to increase freight and passenger traffic, which entails an increase in the fleet of vehicles, including locomotives and high-speed trains. Their use is associated with an application in vehicle suspension systems is complex curved profile and different types of hydraulic dampers and vibration absorbers, which leads to an increase in smoothness of rolling stock and to improve the strength and elastic-dissipative properties of the suspension to reduce the dynamic effects on the path and the transported goods.

Analysis of research on the calculation of hydraulic vibration dampers of vehicles has shown that almost all spring suspension designs of modern electric locomotives, electric trains, metro cars, and trams use hydraulic dampers various types. Hydraulic vibration dampers served as the basis for the creation of new types of adjustable suspensions with automatically controlled parameters speed and high-speed electric rolling stock (for example, in France, Germany, Switzerland, Japan, Spain, China, Russia, and Uzbekistan) [1-3]. To increase the speed of movement and the capacity of railways, it is of great importance to improve the dynamic qualities of electric rolling stock, which depend on the correct choice and stability of the hydraulic vibration dampers and the parameters of the spring suspension. These indicators depend on: the interaction forces between wheel and rail, the stability of the wheels on the rail, and the smooth running locomotive.

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2 Materials and Methods

To substantiate the created perspective design of the vibration dampener, a generalized dynamic model of the hydro friction vibration dampener was developed, taking into account the viscous damping [4].

The generalized model under study consists of three coaxially arranged elastic cylindrical shells, one of which is filled with an ideal compressible fluid moving at a constant velocity *V* and having a pulsating internal pressure (a variable pressure field in time and along the length of the $P_l\,(x, t)$) [4, 5].

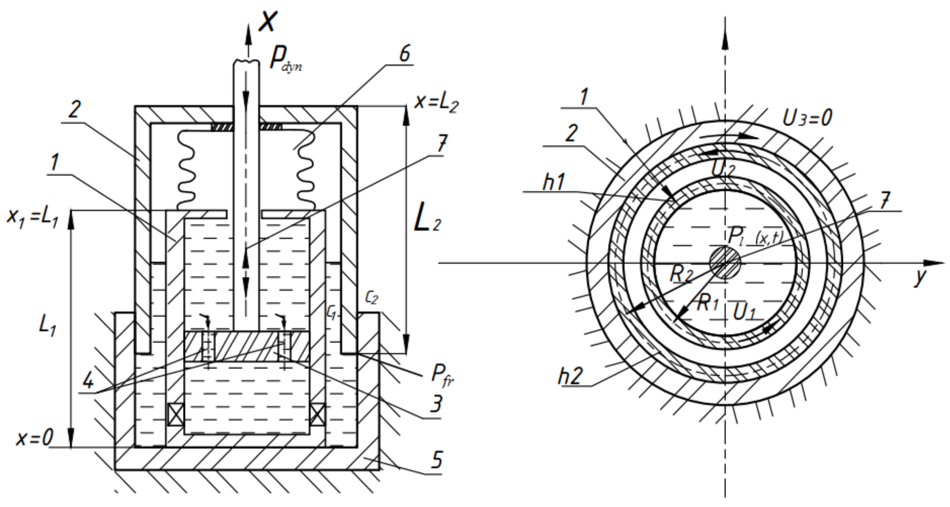

Fig.1. Calculation scheme for modeling the process of dynamic loading of the hydro friction damper of the proposed design [4] and the acting forces, (where it is indicated: 1 is working cylinder, 2 is outer casing (movable), 3 is piston, 4 are throttle channels, 5 is fixed casing part, 6 are bellows, 7 is rod)

Let us turn to the equations of the shell, taking into account the propagation of waves in its material. We take here a variant of the linearized theory of Kirchhoff - shells, considering the deflections of the shells as small U_1, U_2, W_1, W_2 - in contrast with the thickness of the shell [6, 7].

Circular cylindrical shells have external radii R_1, R_2, wall thickness h_1, h_2, length – L_1, L_2, elastic pinching act in the inner shell 1 at the ends, pulsating pressure waves in the liquid $R_j\,(x, t)$.

Between the inner shell 2 and the outer shell 3, which is rigidly fixed, there is a friction force expressed by the formula:

$$P_{fr}(x,t) = f_{fr} \cdot P_{dyn} \tag{1}$$

There is an external pulsed dynamic load on the outside [8]:

$$P_{dyn}(x,t) = P_a \cdot \cos\omega t \tag{2}$$

The *OX* axis is compatible with the longitudinal axis of the shell L_1. Displacement of the middle surface of the shell 1 in directions forming denote by U_1, and the displacement of the middle surface of the shell 2 is rigidly fixed relative to the outer shell U_2. Finally, the radial displacement, respectively – W_1, W_2 [8].

3 Results and Discussion

Based on the data of [4, 9], it is possible to write down the equations of vibrations of two circular cylindrical shells, one of which is filled with a moving compressible liquid with pulsating pressure along the length and time under dynamic impact P_{dyn}, in movements:

$$\frac{E_1 h_1}{1-\mu^2}\left(\frac{\partial^2 U_1}{\partial x^2}+\frac{\mu \partial W_1}{R_1 \partial x}\right)+\beta\frac{\partial U_1}{\partial t}=\rho_1 h_1\frac{\partial^2 U_1}{\partial t^2} \tag{3}$$

$$\frac{E_2 h_2}{1-\mu^2}\left(\frac{\partial^2 U_2}{\partial x^2}+\frac{\mu \partial W_2}{R_2 \partial x}\right)+K_2 U_2=\rho_2 h_2\frac{\partial^2 U_2}{\partial t^2}+P_{fr}+P_{dyn} \tag{4}$$

In equations (3), (4), the notation E_i, μ_i, ρ_i are the modulus of elasticity, the Poisson's coefficient, and the density of the shell material, where $i = 1$.

Consider the solution for the system of equations (3) ÷ (4), taking into account the sequential complication.

The oscillations of the two coaxial shells consider the assumption of small transverse expansion (to be exact $\frac{\partial \boldsymbol{W}_1}{\partial \boldsymbol{x}}$ and $\frac{\partial \boldsymbol{W}_2}{\partial \boldsymbol{x}}$ tend to 0). Taking this into account, the assumed system (3) ÷ (4) will take the form:

$$\frac{E_1 h_1}{1-\mu^2}\cdot\frac{\partial^2 U_1}{\partial x^2}-\beta\frac{\partial U_1}{\partial t}-c_1 U_1=\rho_1 h_1\frac{\partial^2 U_1}{\partial t^2}+\bar{f}_1\cdot P_{dyn}(x,t) \tag{5}$$

$$\frac{E_2 h_2}{1-\mu^2}\cdot\frac{\partial^2 U_2}{\partial x^2}-c_2 U_2=\rho_2 h_2\frac{\partial^2 U_2}{\partial t^2}+\bar{f}_2\cdot P_{dyn}(x,t) \tag{6}$$

Divide equations (5) and (6) by $\rho_1 h_1$ and $\rho_2 h_2$ respectively, and transform them by entering the notation:

$$\frac{\partial^2 U_1}{\partial t^2}-a_1^2\frac{\partial^2 U_1}{\partial x^2}+\overline{\beta}_2\frac{\partial U_1}{\partial t}+b_1^2 U_1=d_1 P_{dyn}(t) \tag{7}$$

$$\frac{\partial^2 U_2}{\partial t^2}-a_2^2\frac{\partial^2 U_2}{\partial x^2}+b_2^2 U_2=d_2 P_{dyn}(t) \tag{8}$$

where

$$a_1^2=\frac{E_1 h_1}{(1-\mu^2)\rho_1 h_1}=\frac{E_1}{(1-\mu^2)\rho_1};\quad \overline{\beta}=\frac{\beta}{\rho_1 h_1};\quad b_1^2=\frac{c_1}{\rho_1 h_1};\quad d_1=\frac{\bar{f}_1}{\rho_1 h_1};$$
$$a_2^2=\frac{E_2}{(1-\mu^2)\rho_2};\quad b_2^2=\frac{c_2}{\rho_2 h_2};\quad d_2=\frac{\bar{f}_2}{\rho_2 h_2} \tag{9}$$

As a result of solving this equation, using the Fourier method and the Laplace transform, we get [10, 11, 12]:
for equation (5)

$$U_{k1}(t)=\overline{P}_{a1}\cdot\left[\frac{\cos\omega_a t\cdot(p_3p_4-\omega_a^2)-\omega_a\cdot\sin\omega_a t\cdot(p_3+p_4)}{(p_4^2+\omega_a^2)(p_3^2+\omega_a^2)}+\right.$$
$$\left.+\frac{p_3e^{p_3t}\cdot(p_4^2+\omega_a^2)-p_4e^{p_4t}\cdot(p_3^2+\omega_a^2)}{(p_4^2+\omega_a^2)(p_3^2+\omega_a^2)(p_3-p_4)}\right] \tag{10}$$

similarly for equation (8)

$$U_{k2}(t)=\overline{P}_{a2}\frac{\cos\omega_a t-\cos\lambda_{k2}t}{\lambda_{k2}^2-\omega_a^2} \tag{11}$$

The general form of the solution of equations (5) and (6) will be according to the Fourier method [11]:

$$U_{k1}(x,t)=\sum_{k=1}^{N}\left\{U_{k1}(x)\cdot U_{k1}(t)\right\} \tag{12}$$

$$U_{k2}(x,t)=\sum_{k=1}^{N}\left\{U_{k2}(x)\cdot U_{k2}(t)\right\} \tag{13}$$

Where k = 1, 2, ...N = 5 is the number of harmonics under vibrations of 2 shells (respectively) under dynamic external loading [13].

The obtained system of equations (7) and (8) takes into account the forces of viscous friction in the details of the proposed damper with the coefficient of energy absorption dynamic loads arising from irregularities in the track [14]. It allows to assess the influence of the geometrical dimensions and parameters of the structure on the parameters of the bodywork of the car electric rolling stock and the details of the damping [15, 16, 17].

Numerical studies were performed in the Mathcad 15 programming environment and are presented in Figures 2, 3, and 4.

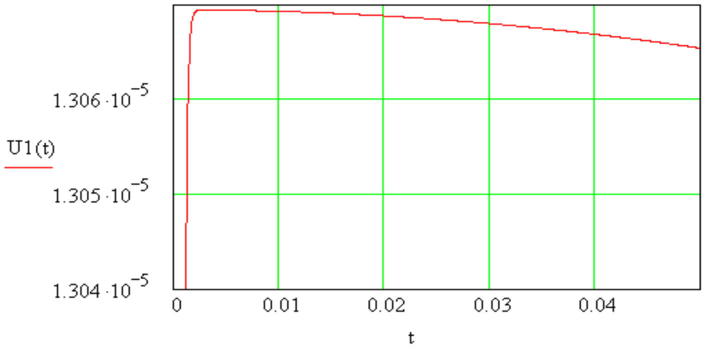

Fig.2. Dynamic displacements of the shell 1, in time $U_1(t)$.

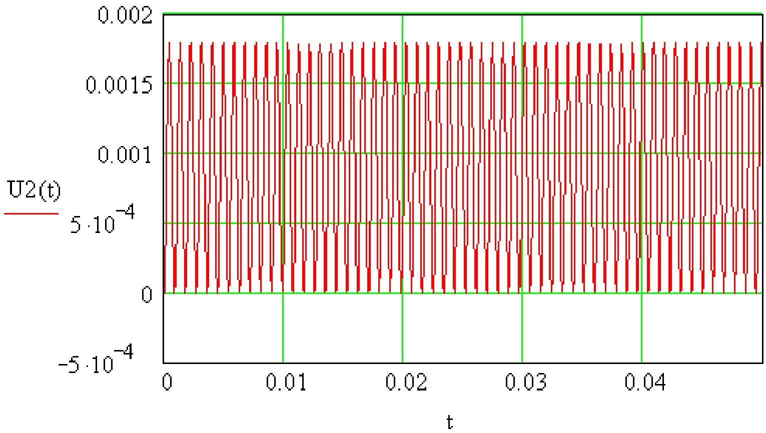

Fig.3. Dynamic displacements of 2 shells in time $U_2(t)$.

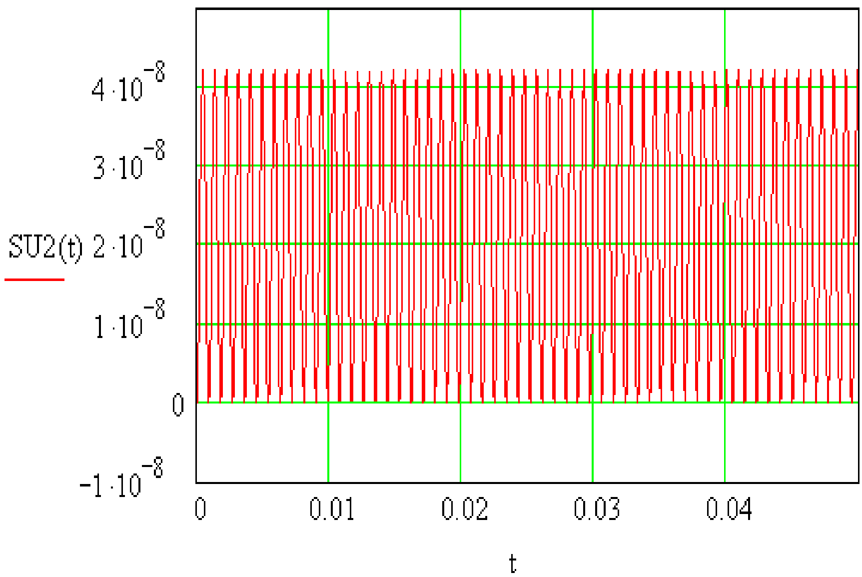

Fig. 4. Total displacements of the second shell of the hydro friction vibration damper in time.

4 Conclusions

This mechanical model allows us to describe the dynamic processes occurring in the system and calculate the dampener parameters taking into account the pre-set dynamic characteristics, which are important when calculating and designing new vibration dampeners for rolling stock, as well as when upgrading existing ones [4, 18]. The resulting mathematical model also allows us to evaluate the influence of structural, power, mass parameters and track irregularities on the process of vibrations of the car body in the vertical and longitudinal horizontal planes, as well as the value of the dynamic load of the vibration dampener parts [19, 20].

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