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MATHEMATICAL MODELING OF WORKING PROCESSES IN A LIQUID-MAGNETIC DAMPER

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ABSTRACT

A mathematical model of the workflow has been developed, which made it possible to study the influence of design and geometric parameters, and also the influence of the working fluid temperature on the performance of a hybrid liquid-magnetic damper. The combination of an innovative hybrid structure, magnetic field and hydraulic throttling in the damper gives it new properties, however, during development, arises the problem of choosing rational parameters of its elements to ensure a given operating characteristic. To solve the problem posed, we analyzed the physical processes occurring during the operation of the damper and built a mathematical model using a cyclic-modular approach. Based on the results of mathematical modeling, a prototype of a liquid-magnetic damper was made.

INTRODUCTION

The development of technologies in the field of vibration damping puts more and more demands on modern dampers. To date, an urgent issue in the design of hydraulic dampers is the study of working processes in its working chambers. The characteristic of the hydraulic damper is mainly determined by the viscosity of the working fluid and the elements of the valve-throttle group, as well as hydrodynamic processes, which are accompanied by a local increase in temperature and disruption of the integrity of the working fluid flow under certain operating modes and the occurrence of a two-phase flow [1-5]. The use of damper devices in prostheses (Fig. 1) also requires a fairly accurate adjustment of the characteristics depending on the input law or its stabilization under changing operating conditions [4-7].

Even though in the field of prosthetics in recent years, new prostheses with improved operational properties have been developed, the issue of making a simple, thermostable, easy-to-use, with a stable characteristic, an adaptive but at the same time cost-effective prosthesis remains relevant. Also, prosthetics outside our country leads to problems with the subsequent maintenance of the prosthesis, which incurs additional material costs for a person who needs prosthetics.

To improve the characteristics of the new high-performance damper, a new hybrid design of the liquid-magnetic structure (Fig. 2) has been used, it has several advantages over other dampers, in particular, durability and simplicity of design.

EXPOSITION

However, this symbiosis of two hybrid structures requires in-depth verification at the design stage and is insufficiently studied, which is associated with the complexity of hydrodynamic processes, magnetic interaction, theoretical description and formulation experimental research [7-12].

The proposed schematic diagram is taken as the basis for the mathematical model of the liquid-magnetic damper (Fig. 1c) [7], which has a sealed hydraulic chamber and excludes the possibility of leaks of the working fluid.

Also, the damper (Fig. 1c) includes: a rod A, which is connected to a hollow sealed piston, in the middle of which there is a liquid, and a movable piston B, which is formed by ring magnets.

The piston contains a turbulent calibrated throttle hole that allows fluid to flow from one part of the piston chamber to another when the rod moves. Ring magnets located at the edges of the sleeve interact with the magnetic piston, forcing it to move under the influence of magnetic repulsion forces and hydrostatic force arising from changes in pressure in parts of the piston chamber when the rod moves. This creates resistance forces on the road.

Building the mathematical model

To study working processes and select their rational parameters, numerical modeling is used. The complexity of the damper and the presence of physically dissimilar components of the damper elements determined the choice of a special approach to constructing its mathematical model. A cyclical-modular approach was applied, which

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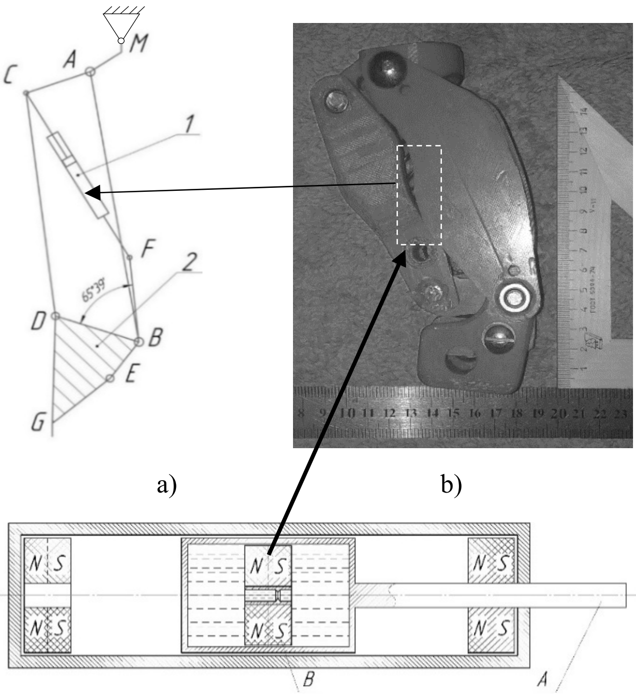
made it possible to reduce the time for building the model and the time for setting it up [4].



c)

Fig. 1. Kinematic diagram of a knee prosthesis (a), a prototype of a prosthesis (b) and a liquid-magnetic damper (c) [7]

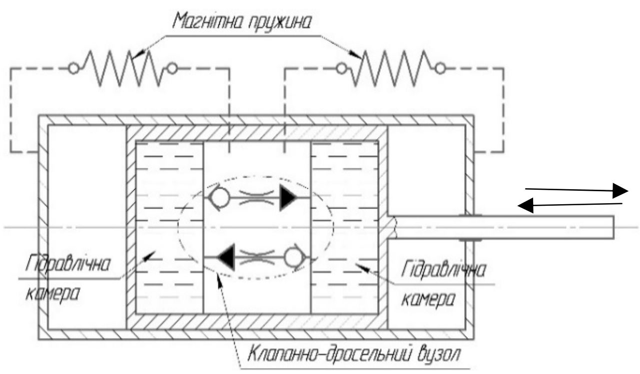


Fig. 2. Schematic diagram of the liquid-magnetic damper

Liquid dampers have a certain characteristic of the dependence of the damping force on the velocity of the piston. Dampers with linear characteristics are especially sensitive to the effect of temperature, in which the overflow of the working fluid in the throttle element is carried out in a laminar flow. The resistance force of the damper for a laminar throttle is determined by the relationship [6]:

$$F = k \cdot v_p$$

where v_p - piston velocity, k - damping coefficient.

The relationship between the modules of the damper and the processes occurring in it is represented by a diagram of the structure of the model (Fig. 3).

The functioning of the damper, according to the diagram, is as follows. The input action in the form of velocity affects the rod A, depicted by the element "mass" 2, under the action of which it moves, which is further, through the flow branching element, is transmitted to the element 3 "conditional connection", which, in combination with the support 4, simulates the limitation of the stroke.

At the same time, the movement is transmitted through the flow splitter 5 to the element 13, which simulates a friction pair. The movement of the rod A is transmitted through the flow splitter 7 to the element 8 - "hydromechanical converter", at the outlet of which a flow is formed, which leads to an increase in pressure in the element 20 - "chamber of variable volume". Under the influence of the increasing pressure in the chamber 20, a flow occurs through the throttle 9, the value of which is determined by the pressure difference in the chambers 20 and 22. The pressure in the chamber 22 leads to the emergence of a force on the hydromechanical transducers 10, under the action of which the piston B is driven, depicted by element 12 "mass" and through the flow splitters is transmitted through the element "conditional connection" 14 to the input of the hydromechanical converter 8. Also, when the mass 12 moves, the distance between the magnets and the corresponding repulsive force change, which is shown in the diagram by the "springs" 17 and 19, and also occurs the frictional force is depicted by the interaction of the element 18 with the support.

Further, the diagram of the structure of the model made it possible to draw up a system of mathematical equations (1) and (2) that describe the processes during the operation of the damper:

$$(1)\begin{cases} \dfrac{dp_{223}}{dt} = \dfrac{E}{V_0 \mp \Delta V} . q_{222} \\ F_{101} = F_{112} + F_{113} \rightarrow F_{112} = F_{101} - F_{113} \\ \dfrac{dv_B}{dt} = \dfrac{F_{112}}{m_B} \\ F_{113} = F_{133} + F_{132} \\ F_{132} = 0,\ при\ h_{132} \neq h_{73} \\ F_{132} = F_{73},\ при\ h_{132} = h_{73} \\ F_{133} = F_{152} + F_{153} \\ F_{152} = k_{17} . (\pm h_{13}) \\ F_{153} = F_{162} + F_{163} \\ F_{162} = k_{19} . (\mp h_B) \\ F_{163} = b_B . v_B \end{cases}$$

$$(2)\begin{cases} v(t) = v_{11} \\ v_{11} = v_{12} = v_{13} \\ \dfrac{dv_2}{dt} = \dfrac{F_2}{m_A} \rightarrow F_2 = F_{12} = \dfrac{dv_{12}}{dt} . m_A \\ F_{11} = F_{52} + F_{53} \\ F_{52} = b_A . v_A \\ F_{73} = p_{82} . S \\ q_{82} = q_{202} + q_{203} \rightarrow q_{202} = q_{82} - q_{203} \\ \dfrac{dp_{82}}{dt} = \dfrac{E}{V_0 \pm \Delta V} . q_{202} \\ q_{82} = v_A - S \\ q_{223} = v_B . S \end{cases}$$

where $v(t)$ – rod velocity, $q_{3.2}$ - flow through the throttle in subcritical mode; $q_{4.2}$ - air flow through the throttle in supercritical mode; μ - flow coefficient in throttles; f -

throttle cross-sectional area; V_0 - initial chamber volume; ΔV - interchangeable chamber volume; v - piston velocity; S - effective piston area; b - viscous friction coefficient; $F_{8.2}$ - force on the piston; h - moving.

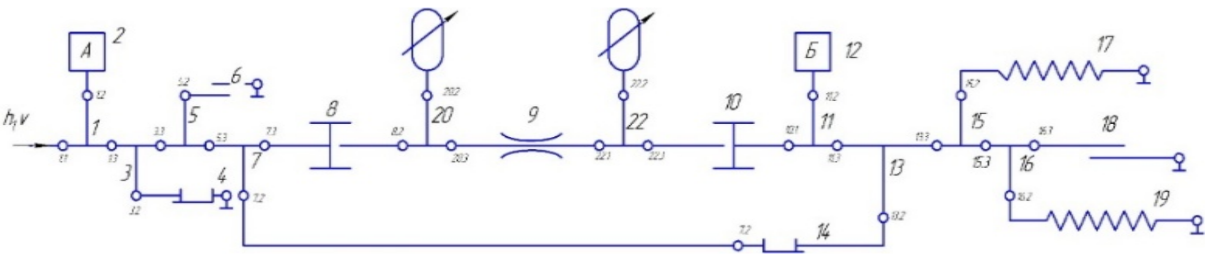


Fig. 3. Diagram of the structure of the model of the liquid-magnetic damper

Checking the adequacy and correctness of the model

The working processes in the damper, which are described by the systems of equations (1) and (2), which include: inertia of the moving masses of the rod and piston, the forces of viscous friction arising at the contact points of the elements, the change in pressure in the chamber of variable volume, taking into account the compressibility of the fluid at volume change, fluid flow through a turbulent throttle in the piston, taking into account the interaction of repulsion of magnets, presenting them in the form of magnetic springs.

Checking the adequacy of the model (Fig. 4) showed the correctness of its work.

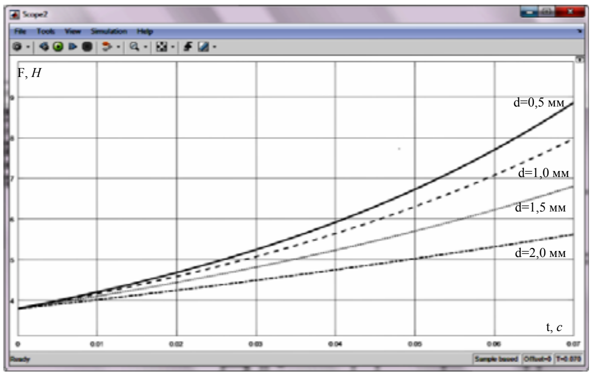


Fig. 4. Dependence of the resistance of the damper on the diameter of the throttle

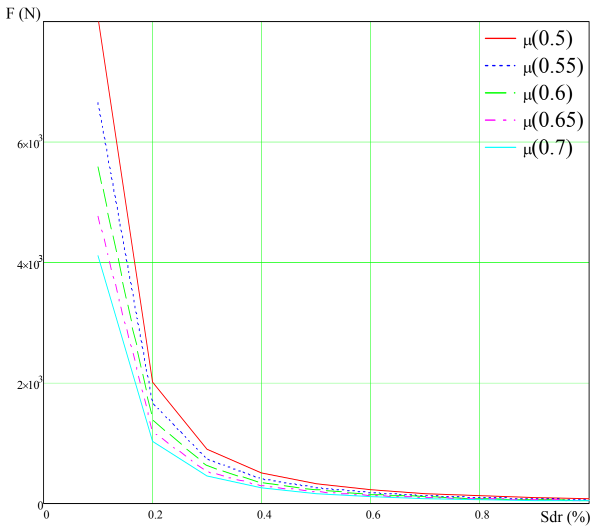


Fig.5. Dependence of the resistance force of the damper F (N) on the diameter of the choke Sdr (%) at different discharge coefficients (μ) (h) = 0.011m

The developed mathematical model of the liquid-magnetic damper allows to study the influence of its parameters and the mechanical part of the spatial mechanism and the temperature of the working fluid (Fig. 5) [5-6] on the resistance forces and the process of vibration damping.

The use of the model will make it possible to confirm the effectiveness of compensation for changes in temperature characteristics by correcting the areas of the throttle holes of the damper valve-throttle assembly and to clarify the required amount of change in the areas of the throttle holes.

For additional verification of the adequacy of the developed mathematical model, simulation was also performed in the Simulink environment using the Simscape application package.

The mathematical model for the Simscape application package is described by the system of equations (3):

$$
(3)\begin{cases}
F = m\dfrac{dv}{dt} \\
F_{tr} = \sqrt{2e}\left(F_{brk} - F_c\right).exp\left(-\left(\dfrac{v}{v_{st}}\right)^2\right).\dfrac{v}{v_{st}} + \\
+ F_c.tanh\left(\dfrac{v}{v_{coui}}\right) + fv \\
v_{st} = v_{brk}\sqrt{2} \\
v_{coui} = v_{brk}/10 \\
v = v_R - v_c \\
F_l = \begin{cases} K_p.\left(x - g_p\right) + D_p.v \;\; for \;\; x \ge g_p \\ 0 \;\; for \;\; g_n < x < g_p \\ K_n.\left(x - g_n\right) + D_n.v \;\; for \;\; x \le g_n \end{cases} \\
v = \dfrac{dx}{dt} \\
E = E_1 \dfrac{1 + \alpha\left(\dfrac{p_a}{p_a + p}\right)^{1/n}}{1 + \alpha\dfrac{p_a^{1/n}}{n.\left(p_a + p\right)^{\frac{n+1}{n}}}E_1} \\
q = \dfrac{V_0 + A\left(x_0 + x.or\right)}{E}.\dfrac{dp}{dt} \\
F = \varepsilon.p.A \\
q = C_D.A\sqrt{\dfrac{2}{\rho}}.\dfrac{p_r}{\left(p_r^2 + p_{cr}^2\right)^{1/4}}
\end{cases}
$$

where F_{tr} – friction force; F_c – sliding friction; F_{brk} – shear friction force; v_{brk} – shear rate; v_{St} – limiting velocity according to Shtribeck; v_{Coul} – limit velocity for sliding friction; v_R, v_C – surface movement velocity (signals on element channels); v – relative velocity; f – viscous friction coefficient; F_I – force of interaction between with movable and fixed link; g_p – the initial value of the gap between the coordinate of the movable link and the upper limit of movement; g_n - the initial value of the gap between the coordinate of the movable link and the lower limit of movement; x - the current coordinate of the moving link; K_p – contact stiffness upper bound; K_n- contact stiffness lower bound; D_p– damping coefficient upper bound; D_n – damping coefficient lower bound; v – moving link velocity; E_l – elastic modulus of pure liquid; p_α- atmosphere pressure; α – the relative proportion of dissolved gas (air) in a liquid at atmospheric pressure; n – polytropic coefficient for gas; q – volumetric flowrate in the throttle; p – differential pressure across the throttle; p_r – throttle pressure loss; C_D – discharge coefficient in throttles; A – throttle cross-sectional area; ρ – fluid density; p_{cr} – turbulent critical pressure.

The testing of the mathematical model showed the correctness of the model and the interaction of components. So the results of modeling the "rebound" mode (Fig. 6) show that the volume of the liquid chamber decreases - this is explained by the action of inertial and magnetic forces, as well as forces of viscous friction on the internal piston.

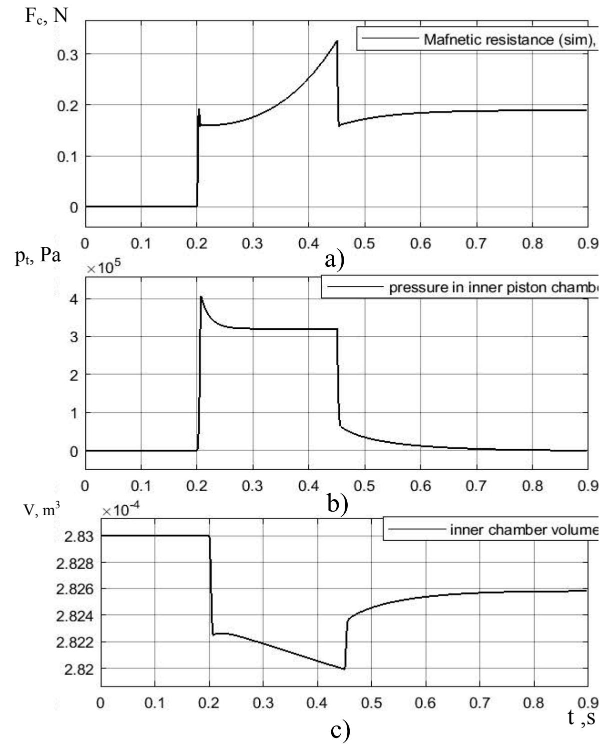


Fig. 6. Dependence of working processes in the chambers of the liquid damper in the "rebound" mode (a - characteristic of the change in the "stiffness" of the magnetic spring; b - characteristic of the change in the pressure difference across the throttle of the internal piston; c - change in the volume of the hydraulic chamber when the internal piston moves)

The total force vector is directed against the movement of the main piston. When simulating the "compression" mode, the presence of oscillatory processes of a shorter duration is observed than in the "rebound" mode. After the force acting on the main piston is removed, the accumulated reactive force returns the main piston to its original position. In this case, a transient process is observed with a peak value that does not exceed 10% of the total amplitude.

As can be seen from the simulation results of the "rebound" mode, the volume of the liquid chamber decreases, which is explained by the action of inertial and magnetic forces and forces of viscous friction on the internal piston. The total force vector is directed against the movement of the main piston.

CONCLUSION

The developed mathematical models made it possible to additionally take into account the operating conditions of the liquid-magnetic damper and the influence of its parameters on the damping process.

The functional diagram of the damper is determined and the means of its implementation is selected. Carried out model studies for a given range of temperature and load changes.

Based on the simulation results, the degree of temperature influence from 20°C to 50°C was established. In the "compression" mode, the maximum change in the resistance force was 20%, and in the "rebound" mode, 40%.

The conducted model studies have shown that the proposed damper design allows satisfactory planned performance.

The developed mathematical models made it possible to confirm that, taking into account certain design modifications (introduction of a "compensation" module into the damper design) and the use of a working fluid with a lower sensitivity to viscosity changes in a wide temperature range, it will provide the planned characteristics of the damper under changing operating conditions.

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