

DYNAMICS OF VIBROSHOCK DAMPER WITH INERTIAL HYDRAULIC CONVERTER**

Grigory Ya. Panovko¹, Alexander E. Shokhin*¹

¹Institute of Mechanical Engineering Research, Moscow, Russia
gpanovko@yandex.ru
shohinsn@mail.ru

Keywords: Damper, Inertial Hydraulic Converter, Hydraulic Mounts, Vibroisolation, Shock and Vibration.

Abstract. *The problem of protection of technical systems and technological machines from shock and vibration by using antivibration mounts with inertial hydraulic converter is discussed. A rheological model of inertial hydraulic converter, which is a system of two hydraulic cylinders of unilateral operating principle, connected by a rigid hydraulic tube, is described. Dynamics of fluid in hydraulic converter is described by the Navier-Stokes equations for a compressible fluid and the equation of state of the fluid in assumption of its isentropic motion. It allows to take into account the ripple effects occurred in the hydraulic converter at impact and vibro-impact loading. The effect of the diameter and length of the hydraulic channel on the dynamic properties of the antivibration damper at shock loading is studied by numerical simulation in finite element package LS-DYNA. It is found that increasing the length of the channel and reducing the channel diameter lead to an increase in the transmitted dynamic force. It is also shown that significant local dynamic pressure of the fluid in hydraulic converter occur at impact loading, that must be considered to ensure the required reliability of antivibration mounts.*

1 INTRODUCTION

Hydraulic vibration dampers (hydraulic mounts) are used for vibration isolation of various technical objects in the automobile and aircraft industry, as well as other areas of technology [1]. A structure of hydraulic mounts (Fig. 1) generally includes a hollow metal body *1*, a thick-walled rubber shell *2* vulcanized to the upper end of the housing *1*, and an elastomeric shaped membrane *3*, attached to the lower end of the housing *1*. A volume bounded by the housing the shell and the membrane filled with fluid and is divided into two chambers by an internal plate *4* - working chamber (upper) and compensation chamber (lower). The upper and lower chambers are connected by channels *5* formed in the metal portion of the plate *4* in form of tubes.

Unlike other types of passive vibration isolation systems, hydraulic mounts having main advantages of both rubber vibration dampers and hydraulic dissipative elements, provide the ability to obtain a frequency range with low dynamic stiffness through internal dynamic reactive forces due to the inertia of the fluid in the channels [1]. To achieve the effect, so-called inertial channels are used in hydraulic mounts, i.e. channels which length is substantially greater than their diameter. As shown in [1-3], the main effect of reducing vibration by hydraulic mount tuned to a given frequency of external excitation is due to the inertial motion of working fluid contained in the channels connecting the chambers. In this case the reduced mass in the equivalent mechanical system is greater than the actual weight of the fluid in the channels. Such a hydro-mechanical system in vibration dampers are usually called inertial hydraulic converter (IHC).

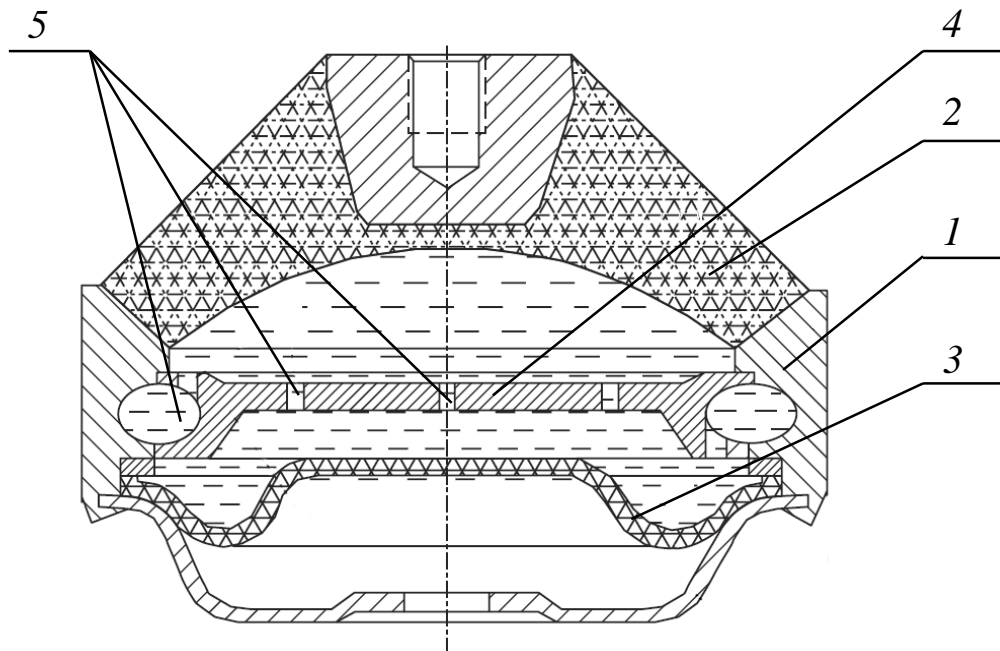


Figure 1: hydraulic mount.

The problem of effective use of vibration dampers with IHC is the choice of the design parameters required to achieve the desired effect of inertia and to ensure the specified dynamic performance under vibration and shock loading conditions. This task is usually carried out on the basis of mathematical modeling of hydraulic mount dynamics. The main difficulty is that the need to consider a coupled problem of fluid dynamics and dynamics of mechanical system. It is certain known mathematical difficulties, so in practice models in which real hydro-mechanical system appears in the form of mechanical or electro-mechanical

analogues are usually used [1,2,4,5]. However, these models do not describe the dynamics of fluid in hydraulic mounts, and are not applicable to study rapidly changing processes, which is particularly important for the effective application of hydraulic mounts at vibro-impact loading modes.

The main purpose of this paper is to study the influence of inertial channel parameters on the dynamic properties of hydraulic mount equipped with IHC subjected to impact loading. Numerical simulation of dynamics of hydraulic mount is carried out by finite element method in LS-DYNA software. The results of study the influence of the inertial channel parameters on the dynamic properties of hydraulic mount are presented.

2 HYDRAULIC MOUNT MODEL

Consider a model of hydraulic mount with single inertial channel. Design scheme of hydraulic mount (Fig. 2) can be represented as two hydrocylinders of unilateral operation principle located opposite to each other and interconnected by cylindrical channel, all formed in a single body 2 and is rigidly fixed to the stationary base 1. Upper and lower pistons of hydraulic cylinders 3 and 4, having mass m_1 and m_2 , respectively, are connected to the housing 2 by means of springs 5 and 6, having stiffness c_1 and c_2 , respectively. Each piston has one degree of freedom, corresponding to translational movement along the longitudinal axis y of the hydrocylinder. It is assumed that there is no friction between the pistons and cylinder walls. This is due to the fact that a piston in real hydraulic mount structure is normally a deformable shell, which is rigidly fixed to the housing and therefore it does not form any friction pair. The region bounded by the pistons and the housing filled with a fluid. Impact force is applied to the upper piston 3.

To study the dynamic characteristics of vibration dampers with IHC under impact loading it is necessary to consider a coupled problem of dynamics of compressible fluid and mechanical system. The solution to this problem is most conveniently done using numerical methods, namely the finite element method. For this purpose a finite element model of hydraulic mount shown in Fig. 3 has been developed. Analysis of the model is done in the LS-DYNA software.

A rheological fluid (area 2 in Fig. 3) and an empty space (area 2*) are simulated by solid 8-node hexahedron finite element type SOLID [6], having six degrees of freedom in each node: displacements along the axes x , y and z , and rotations about these axes. Areas 2 and 2* form a single space with different initial density (the density of area 2* initially is assumed to be zero) and other similar physical properties. Fluid motion within that space is considered on a single Eulerian mesh and is described by the Navier-Stokes equations for compressible barotropic fluid:

- equations of motion of a compressible medium:

$$\rho \frac{d\bar{\mathbf{v}}}{dt} = \rho \bar{\mathbf{F}} - \text{grad}(p) + (\mu_0 + \frac{\mu}{3}) \text{grad}(\text{div}(\bar{\mathbf{v}})) + \mu \nabla^2 \bar{\mathbf{v}};$$

- equation of continuity:

$$\frac{\partial \rho}{\partial t} + \text{div}(\rho \bar{\mathbf{v}}) = 0;$$

- equation of state:

$$f(p, \rho) = 0;$$

where: ρ is the fluid density, $\bar{\mathbf{v}}$ is the fluid velocity vector, $\bar{\mathbf{F}}$ is vector of body forces, acting on the fluid, p is pressure, $\mu_0 = 2\mu(1 + \nu)/3(1 - 2\nu)$ is bulk viscosity, μ is viscosity, ν is the Poisson's ratio.

Meshing the space is carried out on a uniform grid of cubic elements with decreasing size of the elements in the area of channel-to-cylinder chambers connection, as well as in contact of fluid with pistons 3 and 4. For the nodes located on the external surface of areas 2 and 2*, except for the outer edge of area 2* boundary conditions are given in form of zero velocity in all directions. Properties of the fluid and the empty space are defined by the following parameters: density ρ , viscosity μ , and equation of state in the form of Gruneisen [6]:

$$p = \frac{\rho_0 C^2 \lambda [1 + (1 - \frac{\gamma_0}{2}) \lambda - a \lambda^2]}{1 - (S_1 - 1) \lambda - S_2 \frac{\lambda^2}{\lambda + 1} - S_3 \frac{\lambda^3}{(\lambda + 1)^2}} + (\gamma_0 + a \lambda) E,$$

where E is the internal energy per initial volume, C, S_1, S_2, S_3 are coefficients of the curve describing the dependency between impact rate u_s and particle speed u_p in the form of $u_s = C + S_1 u_p + S_2 u_p^2 + S_3 u_p^3$, γ_0 is the Gruneisen gamma, a is the first order correction to γ_0 ; $\lambda = \frac{\rho}{\rho_0} - 1$; ρ, ρ_0 are current and initial fluid densities, respectively.

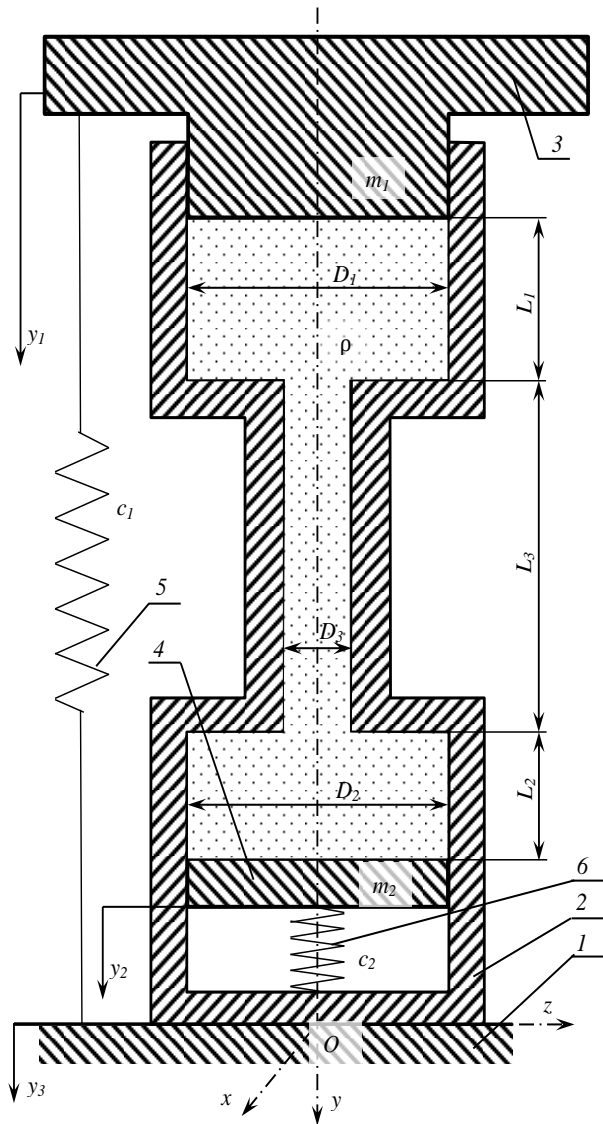


Figure 2: design scheme of hydraulic mount.

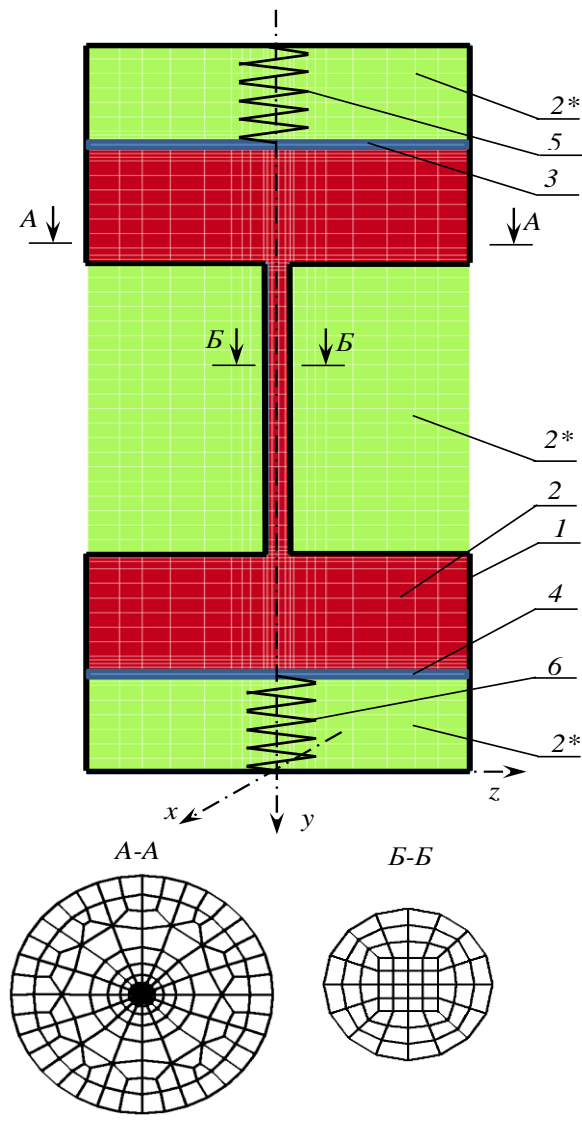


Figure 3: finite element model of hydraulic mount.

Housing of hydraulic mount (polyline l in Fig. 3) is modeled as massless rigid shell, which bounds the space of possible movement of fluid. It is considered that the housing is rigidly connected with the base (fixed or moveable, depending on the conditions of the problem), therefore a separate modeling of base is not performed. To simulate the housing 4-node shell elements type SHELL [6] are applied. Meshing is carried out so that the nodes of the housing finite elements coincide with the nodes of mesh of regions 2 and 2*.

Pistons 3 and 4 are modeled by solid 8-node hexahedron finite elements type SOLID, having one degree of freedom at each node: displacement in direction of axis y . Meshing is carried out on a uniform cubic grid with the elements size equal to the size of elements in the surrounding areas 2 and 2*. Each piston is considered as rigid body, which motion is described by following equations:

- for upper piston 3:

$$m_1 \frac{d^2 y_1}{dt^2} = F_o(t) + F_1(y_1) - \int_{A_1} p_3(x, z, t) dA_1;$$

- for lower piston 4:

$$m_2 \frac{d^2 y_2}{dt^2} = \int_{A_2} p_4(x, z, t) dA_2 - F_2(y_2);$$

where y_1, y_2 are coordinates of pistons 3 and 4, F_1, F_2 are forces in springs 5 and 6, F_o is external force, acting on piston 3, A_1, A_2 are cross-sectional areas of pistons 3 and 4, p_3, p_4 – are pressures in cross-sections of contact of fluid to pistons 3 and 4. The material properties are specified by density of pistons, which is chosen so that mass of the upper piston 3 must be equal to m_1 (see Fig. 2) and mass of the lower piston 4 must be equal m_2 .

Coil-springs 5 and 6 (Fig. 3) are modeled by 2-node massless finite elements type DESCRETE [6], having one degree of freedom at each node - displacement in direction of axis y . Each spring is rigidly connected to the housing l and to the corresponding piston. Linear force-strain characteristics of springs are accepted, so their properties are specified by stiffness constants, and the force produced is determined by the difference in displacements of their ends.

Simulation of dynamics of hydraulic mount at impact loading is carried out as follows. At the initial time all the elements of the hydraulic mount are at rest. Each of the pistons is subjected to constant in time, and uniformly distributed over the area of the pistons "atmospheric" pressure $p = 1 \cdot 10^5$ Pa, directed toward the fluid, which is balanced by the internal pressure of the fluid. At time t_1 the upper piston 3 (Fig. 2) is given a velocity v_0 , i.e. completely inelastic impact on the upper piston with instantaneous change in its speed is modeled. Pressure and velocity of the fluid in the chambers and the channel, displacements of pistons, reaction forces of the base are determined in the simulation. Efficiency of hydraulic mount at impact is estimated by peak values of reaction force of the base.

The simulation was performed for the following values of the system parameters:

- geometric and mass parameters: diameters of hydrocylinders chambers $D_1 = D_2 = 30$ mm (Fig. 2), initial height of the fluid in the chambers $L_1 = L_2 = 30$ mm, mass of upper piston $m_1 = 1$ kg, mass of lower piston $m_2 = 0,02$ kg, to study the influence of the inertial channel parameters on the hydraulic mount characteristics simulations were carried out at the channel diameters $D_3 = 2, 4, 5, 6, 7, 8$ mm, and channel lengths $L_3 = 5, 20, 40, 60, 80, 100, 120$ mm;

- physical properties of coil-springs and fluid: stiffness constants of upper and lower springs $c_1 = 5 \cdot 10^4$ N/m, $c_2 = 5 \cdot 10^3$ N/m, fluid type – water, density $\rho = 1000$ kg/m³,

viscosities: $\mu = 1.052 \cdot 10^{-3} \text{ Pa} \cdot \text{s}$, $\mu_0 = 2.81$; equation of state parameters: $C = 1480$, $S_1 = 1.75$, $S_2 = S_3 = 0$, $\gamma_0 = 0.28$, $a = 0$, $V_0 = 0.999954$, where V_0 – relative volume of fluid in the initial moment of time determined by the condition of static equilibrium with the external constant pressure (atmospheric) $p = 1 \cdot 10^5 \text{ Pa}$ acting on the pistons.

- moment of impact $t_I = 10^{-9} \text{ s}$; upper piston velocity after impact $v_0 = 0.5 \text{ m/s}$.

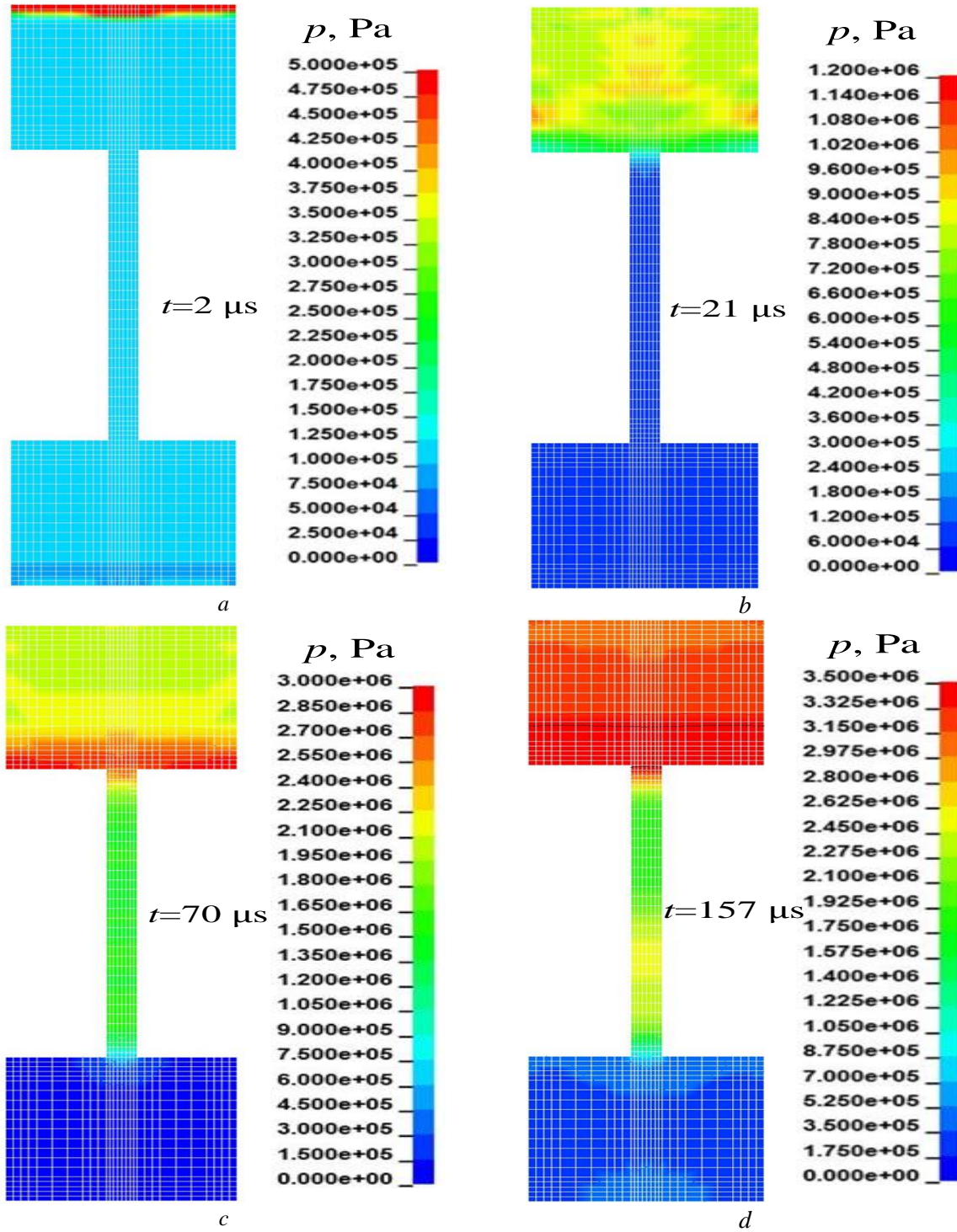


Figure 4: pressure distribution in longitudinal section of the hydraulic mount: *a* – at $t=2 \mu\text{s}$, *b* – at $t=21 \mu\text{s}$, *c* – at $t=70 \mu\text{s}$, *d* – at $t=157 \mu\text{s}$

3 ANALISYS OF RESULTS

Fig. 4 shows graphs of pressure distribution in the fluid in longitudinal section of hydraulic mount with channel diameter of 5 mm and length of 60 mm for different instants of time. Upon impact a shock wave in form of pressure jump in the fluid in the section of contact with upper piston occurs (Fig. 4.a). At the same moment part of the impact energy transmits directly to the lower piston by the fluid column bounded by the diameter of the channel, leading to appearance of the shock wave in the fluid in the section of contact with lower piston. Further, both wave fronts propagate toward each other. At time $t=20 \mu\text{s}$ shock wave front reaches the section of the entrance to the channel (Fig. 4.b). Part of the front interacts with end walls of hydrocylinders, then it reflects forming a wave going in the opposite direction. Another part of the front continues to move in original direction through inertial channel. At $t=70 \mu\text{s}$ (Fig. 4.c) wave front going from upper piston reaches chamber of the lower hydrocylinder. At $t=157 \mu\text{s}$ (Fig. 4.d) maximum levels of pressure in the upper hydrocylinder and inertial channel are observed, and the pressure distribution along the channel is periodic. Further interaction of primary and reflected waves leads to significant levels of pressure non-linearly distributed in the hydrocylinders and in the channel.

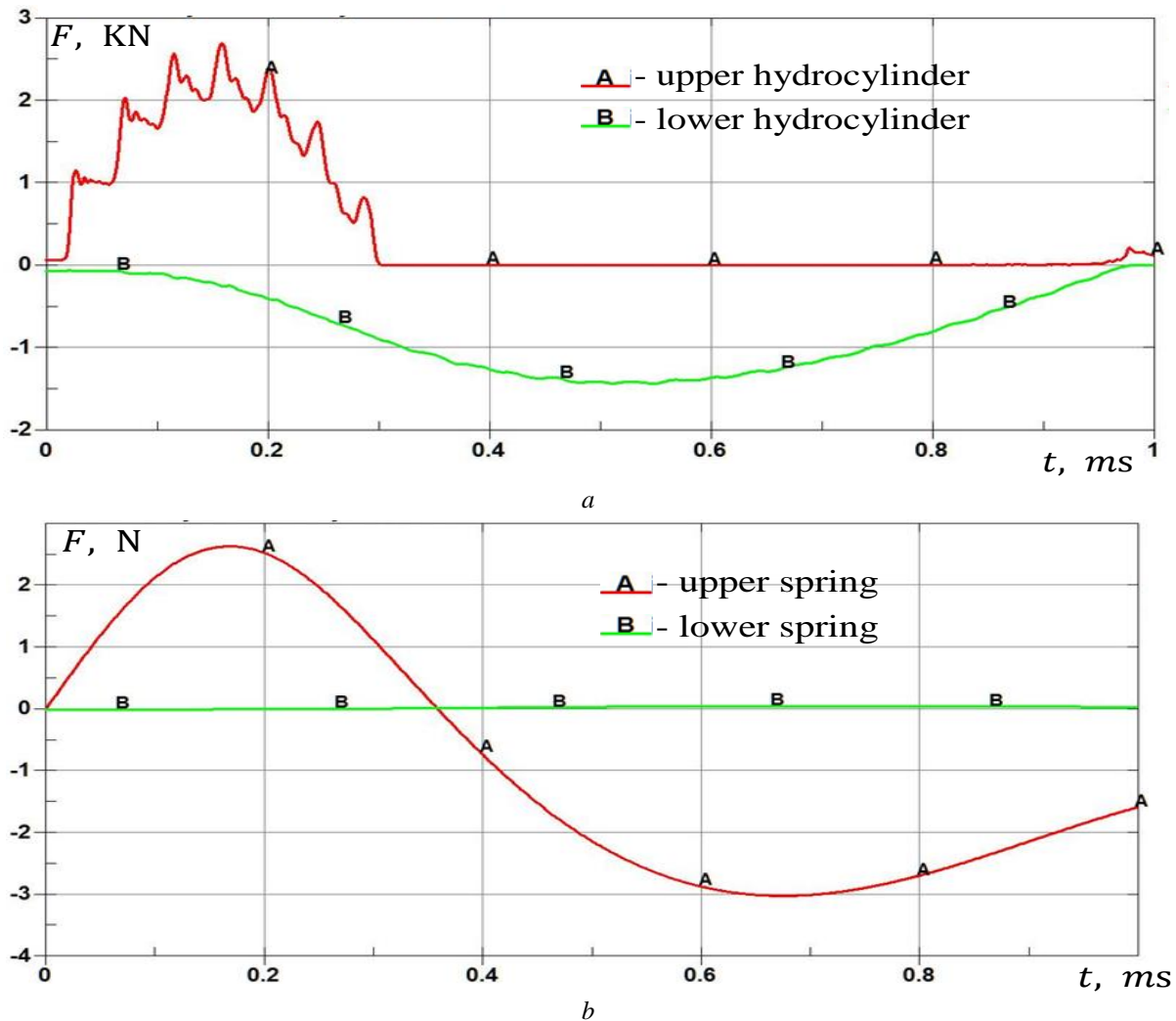


Figure 5: forces acting on the base

Fig. 5.a shows the graphs of forces, which the fluid acts on the end walls of the upper and lower cylinders in the longitudinal direction of the hydraulic mount. It is seen that the exposure on the base is transmitted with a delay relative to the moment of impact. The delay of $20 \mu\text{s}$ practically corresponds to the period of time during which the sound travels a distance from the piston to the entrance to the channel through the fluid. Periodic jumps of the force on the graphs correspond to the moments of arrival of the shock front of the primary and secondary waves to the corresponding end walls of the cylinders.

Fig. 5.b shows reaction forces in springs of upper and lower cylinders. It is seen that reaction forces values in springs are significantly lower, about 10^3 times lower than the value of forces the fluid acts on the housing of hydraulic mount.

Fig. 6 shows the graph of the resultant force acting on the hydraulic mount housing towards its longitudinal axis, which is the sum of the above forces. It can be seen that elastic waves in the fluid attenuate rapidly, as evidenced by reduction of amplitude jumps and quite smooth and uniform form of the force curve starting from $t=2\text{ms}$.

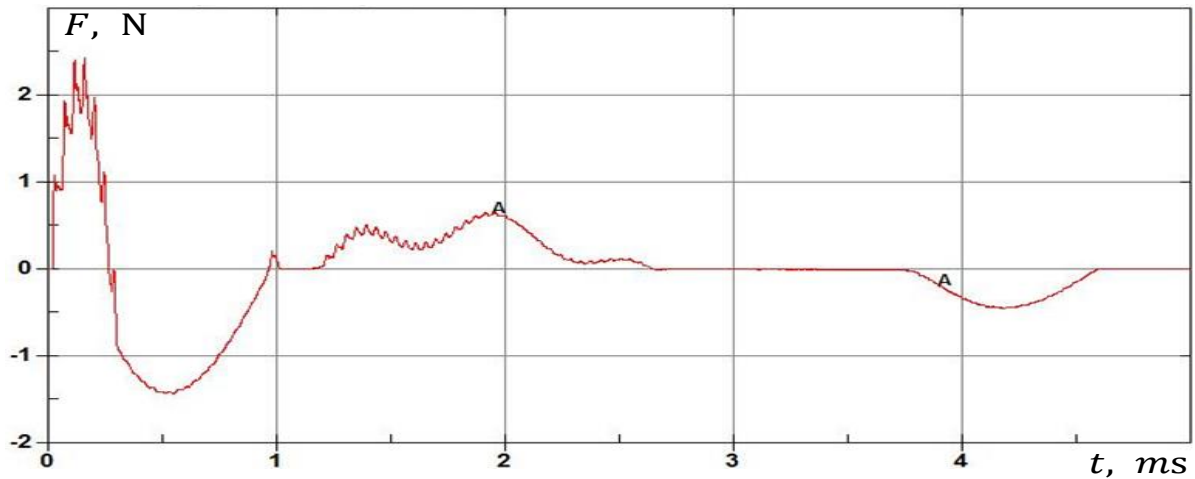


Figure 6: resultant force acting on the base.

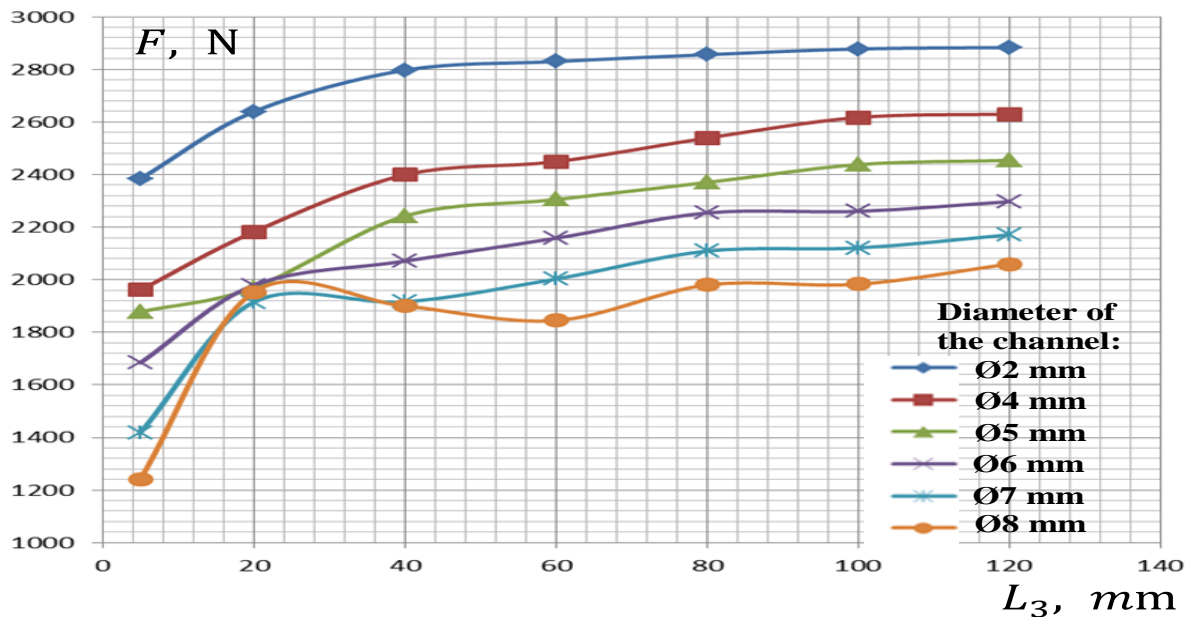


Figure 7: peak values of dynamic force F acting on the base depending on the channel length L_3 and diameter.

Fig. 7 shows graphs of peak values of force acting on the housing at impact depending on diameter and length of the channel. It is seen that the increase in diameter of the inertial channel reduces the force transmitted to the housing. If the channel diameter of from 2 to 6 mm, the force transmitted increases monotonically with the length of the channel, asymptotically approaches a certain value. If the channel diameter of 7 and 8 mm, a decrease of the transmitted force occur when the length of channel changes from 20 to 60 mm. With further increase in length of the channel there is a monotonic increase in force value.

4 CONCLUSION

The model of hydraulic mount with IHC proposed in this paper allows to simulate it's behavior at vibro-impact loading modes taking into account features of dynamics of viscous compressible fluid and the fluid-structure interaction.

Characteristics of the pressure distribution in the fluid (Fig. 4), obtained in simulation, show that impact loads leads to appearance of local areas with high pressure in hydraulic mounts. The magnitudes of pressure in these areas and their location are variable in time. This factor must be taken into account when designing hydraulic mounts since it may be one of the reasons for rapid failure of their elements.

Analysis of influence of the inertial channel parameters (Fig. 5-7) showed that the channel length should be reduced and its diameter should be increased to reduce dynamic forces transmitted to the base at impact. However, as shown in [7], an increase in length and reduction in diameter of the channel leads to increased vibration isolation effect. Therefore, in order to select the optimal parameters of the channel a thorough investigation of probable operating conditions should be carried out.

It should also be noted that the application of this finite element model to study vibrational loading modes is often difficult because it takes considerably more time-consuming for the calculation than in the study of impact loading modes.

REFERENCES

- [1] B.A. Gordeev, V.I. Erofeev, A.V. Sinev, O.O. Mugin, *Vibration isolation system using inertia and damping of rheological fluids*. Fizmatlit, Moscow, 2004.
- [2] V.S. Baklanov, A.S. Gorobtsov, S.K. Kartsov, A.V. Sinev et. al., Analysis of the reactive properties of dynamic stiffness and transfer functions of hydraulic mounts. *Problems of mechanical engineering and machine reliability*, **3**, 33-37, 1999.
- [3] A.N. Brisin, A.E. Shokhin, A.V. Sinev, Features of vibration isolation systems based on inertial transducer with an additional slot damping . *Bulletin of the Nizhny Novgorod University N.I. Lobachevsky*, **4/1**, 72-74, 2011.
- [4] Matthew Brach R., Hadow G. Alan. On the Dynamic Response of Hydraulic Engine Mounts. *SAE paper № 931321*.
- [5] Kyu. H. Lee, et. al., Performance Design of Hydraulic Mount for Low Frequency Engine Vibration and Noise Control. *SAE Paper 941777*, 1994
- [6] *LS-DYNA theory manual*. Livermore Software Technology Corporation, 2006.
- [7] B.A. Gordeev, A.V. Sinev, The effectiveness of hydraulic mount for power unit vibration damping, depending on the size of the connecting tube and the properties of the fluid. *Problems of mechanical engineering and reliability of the machines*, **1**, 2001.