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Vibration insulation of hydraulic system control components

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This paper deals with the effects of external mechanical vibrations on hydraulic valves. A theoretical analysis of the contribution of selected vibration insulators to a reduction in hydraulic valve housing vibrations was carried out. The results of preliminary experimental tests of simple vibration insulators are reported.

Keywords: *mechanical vibrations, pressure fluctuations, hydraulic valve*

1. Introduction

Major features of hydraulic systems are periodic changes of pressure around an average value, commonly referred to as pressure fluctuations. Their consequences are definitely negative. The cyclic operation of the pump's displacement components [1] or the self-excitation of the control components in hydraulic valves [2] due to the action of the flowing liquid [4] or to external mechanical vibrations [3, 5, 6] are among the causes of pressure fluctuations. Pressure fluctuations cause the individual system components to vibrate. This has an adverse effect, particularly on the precision of positioning of, for example, the cutting tool in a machine tool. This also applies (although to a smaller degree) to mobile machines which are the source of vibrations affecting the rigidly fixed hydraulic valves. Generally, the complex problem of the transmission of vibrations by a machine or a piece of equipment can be divided into three interconnected categories:

- vibration sources,
- vibration transmission paths,
- effects.

The most frequent cause of vibrations are disturbances connected with the motion or operation of the machine, for example when a mobile machine moves on an uneven surface or when the rotating parts are unbalanced during material machining. Another major vibration source are drive units, for example a combustion engine performing a periodic variable-characteristic work cycle [7, 8]. An operating hydraulic system is also a source of mechanical vibrations caused mainly by pressure surges and the periodic operation of the displacement pump. Since the generated vibrations have different frequencies the paths of their transmission are also different. The irregularities of the surface on which a mobile machine moves cause excitations in a frequency range of 0.5–250 Hz [9–11]. The latter includes excitations generated by the driving (combustion) engine and the displacement pump kinematics, manifesting themselves in pressure fluctuations in the machine's hydraulic system. The vibrations due to the resistance of flowing air are in a frequency range of 250–16 000 Hz and they are caused by airflow separation from the machine's components. Also the flow of the working medium in the hydraulic system causes vibration and noise. Sometimes cavitation occurs, generating high-frequency noise. The vibrations generated and transmitted by a machine produce various effects. Mechanical vibrations affect the machine operator. The components of the systems with which the machine is equipped, particularly hydraulic components and systems are also subject to mechanical vibrations. Such components are required to have good dynamic properties and to be characterized by stability, positioning precision, operating reliability and certainty and little noisiness. Modern proportional hydraulic valves or hydraulic microvalves are particularly exposed to external mechanical vibrations since the disturbing forces in them can amount to the controlling forces, which may lead to many adverse effects, such as stability loss, positioning inaccuracy, damage to seals and increased noisiness [12].

2. Flexible fixing of hydraulic valve

As mentioned above, in order to minimize the vibration of the hydraulic valve's control element it seems sensible to isolate the valve housing from the external mechanical vibrations of the base (for example the vibrating frame of a mobile machine or a machine tool). For the analysis of the effect of the flexible fixing of a hydraulic valve on the vibration of its housing a special clamping holder for the hydraulic distributor was designed. The latter is on its two sides supported by a system of springs with a known linear characteristic and a known pre-deflection (Figure 1).

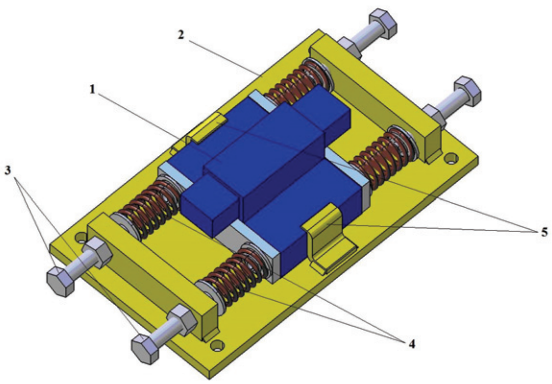

Fig. 1. Valve holder: 1 – hydraulic valve (distributor), 2 – holder base,
3 – spring pre-deflection bolts, 4 – springs, 5 – securing catches

The design of the holder is such that the valve mounted in it is constrained by springs (with an equivalent stiffness) and it moves on the holder base (2 in Figure 1) rubbing against it in accordance with the dry friction model. On its two sides the valve is supported by springs. A scheme of the hydraulic system in which the proportional distributor type 4WRE 6 E08-12/24Z4/M operates is shown in Figure 2.

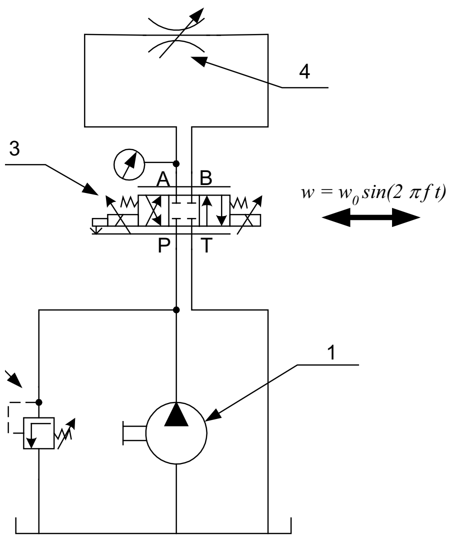

Fig. 2. Scheme of hydraulic system incorporating investigated component: 1 – feed pump, 2 – relief valve, 3 – investigated component, 4 – adjustable throttle valve

For a two-mass system the model of the proportional distributor operating in the hydraulic system shown in Figure 2 can be represented by the following system of four equations:

$$\begin{cases} m_1 \cdot \ddot{X}_1 + \pi \cdot d_t \cdot \dfrac{l}{h} \cdot \mu \cdot \left(\dot{X}_1 - \dot{X}_2\right) + 0.72 \cdot \dfrac{1}{\sqrt{\xi}} \cdot 2 \cdot s_s \cdot \dfrac{\left(X_1 - x_p\right)^2}{x_m} \cdot \left(p_1 - p_2\right) + \\ \qquad + c_1 \cdot \left(X_1 - X_2\right) = F_M, \\ Q_p - 1.5 \cdot s_s \cdot \dfrac{\left(X_1 - x_p\right)^2}{x_m} \cdot \sqrt{\dfrac{2}{\rho} \cdot \left(p_1 - p_2\right)} - a_{p1} \cdot p_1 - c_{k1} \cdot \dot{p}_1 = 0, \\ Q_p - a_{p1} \cdot p_1 - c_{k1} \cdot \dot{p}_1 - c_{k2} \cdot \dot{p}_2 - C_{q1} \cdot A_a \cdot \sqrt{\dfrac{2 \cdot p_2}{\rho}} = 0, \\ m_2 \cdot \ddot{X}_2 + c_1 \cdot \left(X_2 - X_1\right) + k_1 \cdot \left(\dot{X}_2 - \dot{X}_1\right) + c_z \cdot \left(X_2 - w\right) + \\ \qquad + m_2 \cdot \mu_2 \cdot g \cdot \left(1 - H\left(l_0 - \left|X_2 - w\right|\right)\right) \cdot sing\left(\dot{X}_2 - \dot{w}\right) + \\ \qquad + sing\left(\dot{X}_2 - \dot{w}\right) \cdot m_2 \cdot \mu i \cdot g = 0. \end{cases} \tag{1}$$

The fourth equation describes the forces acting on the valve housing in the considered case. Further on this equation will be modified to describe the characteristics of the proposed vibration insulation elements. Some simplifying assumptions to Equations (1):

- working liquid does not change its properties,
- Coulomb friction is neglected in pair: spool-muff inside directional control valve,
- Coulomb friction represents cooperation between valve body and valve holder,
- after play (between valve body and securing catches) is cancelled Coulomb friction represents cooperation between valve body and securing catches,
- springs characteristics are linear and described by stiffness coefficient c,
- description of hydraulic system is based on concentrated parameter model,
- the model does not represent influence pipes on valve body vibrations.

List of major symbols:

Symbol	Parameter	Dimension in SI
a_{p1}	leakage coefficient	[m^{4}s/kg]
A_a	throttle valve gap area	[m^2]
c_1	equivalent stiffness of valve centring springs	[N/m]
c_z	equivalent stiffness of springs fixing valve in holder	[N/m]
C_{q1}	throttle valve flow ratio	[–]
d_t	piston diameter	[m]
f	frequency	[Hz]
g	Earth's acceleration	[m/s^2]
h	valve-sleeve pair gap thickness	[m]
H	Heaviside step function	[–]
k_1, k_2	damping in respectively valve-sleeve pair and housing-holder pair	[Ns/m]
l	piston length	[m]
l_0	gap of valve body and securing catches	[m]
m_1	mass of piston valve and 1/3 of spring mass	[kg]
m_2	mass of distributor housing	[kg]
p_1	pressure before distributor	[Pa]
p_2	pressure after distributor	[Pa]
p_z	sink line pressure	[Pa]
Δp_2	throttle valve pressure drop	[Pa]
s_s	maximum gap width	[m]
t	time	[s]
w	excitation vibration amplitude	[m]
Q_p	theoretical pump delivery	[m^3/s]
x_m	gap length	[m]
x_p	mutual shift of valve and housing edges	[m]
X_1	displacement of piston valve	[m]
X_2	displacement of distributor housing	[m]
$\mu 2$	coefficient of friction of valve housing against securing catches	[–]
μi	coefficient of friction of valve housing against holder base	[–]
ρ	working liquid density	[kg/m^3]
ω	angular frequency	[rad/s]

Model (1) also takes into account the interaction between the valve housing and securing catches 5 (Figure 1). A numerical solution in the form of a "transmission function", understood as a ratio of valve housing vibration acceleration amplitude a_2 to excitation vibration acceleration amplitude a_0, is shown in Figure 3.

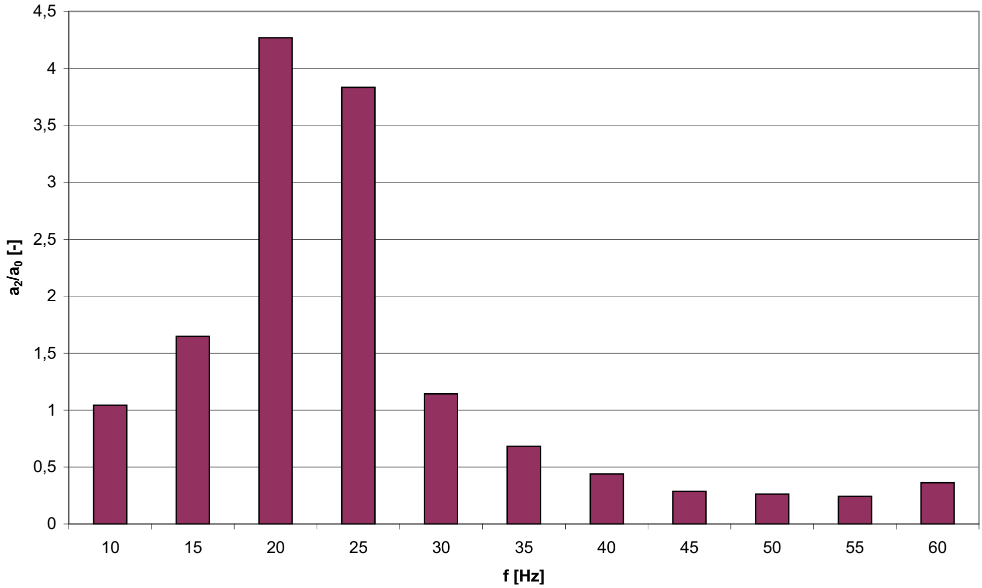

Fig. 3. Proportional distributor housing vibration acceleration amplitude a_2 relative to excitation vibration acceleration amplitude a_0 for f = 10–60 Hz

An analysis of the simulation results shows a considerable gain in housing vibration amplitude at a frequency of about 20 Hz. This is due to resonance since the mass of the vibrating valve amounts to about 4.5 kg and the equivalent stiffness of the holder springs is 86 000 N/m. Hence a gain in distributor housing vibration amplitude is observed in the range of 10–30 Hz (ineffective vibration insulation).

This means that valve insulation which will widen the insulation zone and reduce the resonance zone should be proposed. The black-box approach (Figure 4) was adopted to solve the problem.

Different forms of the insulating element can be assumed. The introduction of a vibration insulator with quasi-zero stiffness significantly contributes to the minimization of valve housing vibrations. The ideal characteristic of the vibration insulator with quasi-zero stiffness is described by the following Equation [13]:

$$F(x) = P_{1H} + c_{1w} l_H \sin\alpha_H + (c_{1w} + 2c_{2w})x - 2(P_{2H} + c_{2w} l_H)\frac{x}{\sqrt{x^2 + l_H^2 \cos^2\alpha_H}}, \tag{2}$$

where:

c_{1w}, c_{2w} – stiffness of respectively the main spring and the compensation spring,
α_H – angle of initial, original inclination of the side arm to axis y,
P_{1H}, P_{2H} – initial spring tensions in position α_H [N],
l_H – length of the side arm in position α_H.

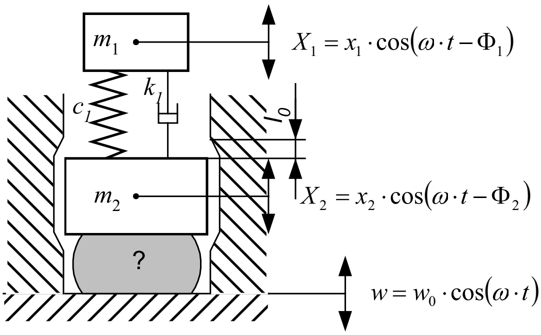

Fig. 4. Black-box approach to valve vibration insulation

The total stiffness of such a vibration insulator in the excitation direction (the direction of the external mechanical vibration) is:

$$c(x) = c_{1w} + 2c_{2w} - \frac{2(P_{2H} + c_{2w} l_H)}{\sqrt{x^2 + l_H^2 \cos^2 \alpha_H}} \left(\frac{l_H^2 \cos^2 \alpha_H}{x^2 + l_H^2 \cos^2 \alpha_H} \right). \tag{3}$$

Thus the fourth equation of model (1) can be written as:

$$\begin{gathered} m_2 \cdot \ddot{X}_2 + c_1 \cdot (X_2 - X_1) + k_1 \cdot (\dot{X}_2 - \dot{X}_1) + \\ + \left(c_{1w} + 2c_{2w} - \frac{2(P_{2H} + c_{2w} l_H)}{\sqrt{X_2^2 + l_H^2 \cos^2 \alpha_H}} \left(\frac{l_H^2 \cos^2 \alpha_H}{X_2^2 + l_H^2 \cos^2 \alpha_H} \right) \right) \cdot (X_2 - w) = 0. \end{gathered} \tag{4}$$

Exemplary solutions of model (1) supplemented with Equation (4) are shown in the figures below for excitation frequency f = 10–60 Hz.

An analysis of the simulation results shows that thanks to the use of the vibration insulator with quasi-zero stiffness the vibration of the valve housing can be considerably reduced. However, because of its dimensions such an insulator cannot be used in small spaces. Therefore materials with good vibration insulation properties and suitable for the use in small spaces should be sought. It seems that special pads (mats) for mounting hydraulic valves on them could meet the requirements. Such materials should also be resistant to hydraulic fluids and extreme ambient temperatures. Using the black-box approach one can select an insulator material characteristic ensuring effective vibration insulation in a wide excitation range.

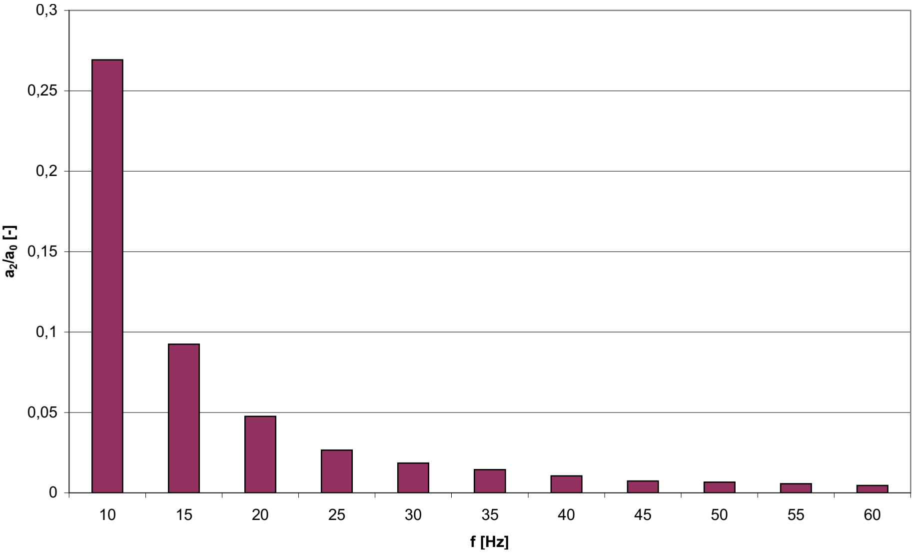

Fig. 5. Proportional distributor housing vibration acceleration amplitude a_2 relative to excitation vibration acceleration amplitude a_0 for f = 10–60 Hz

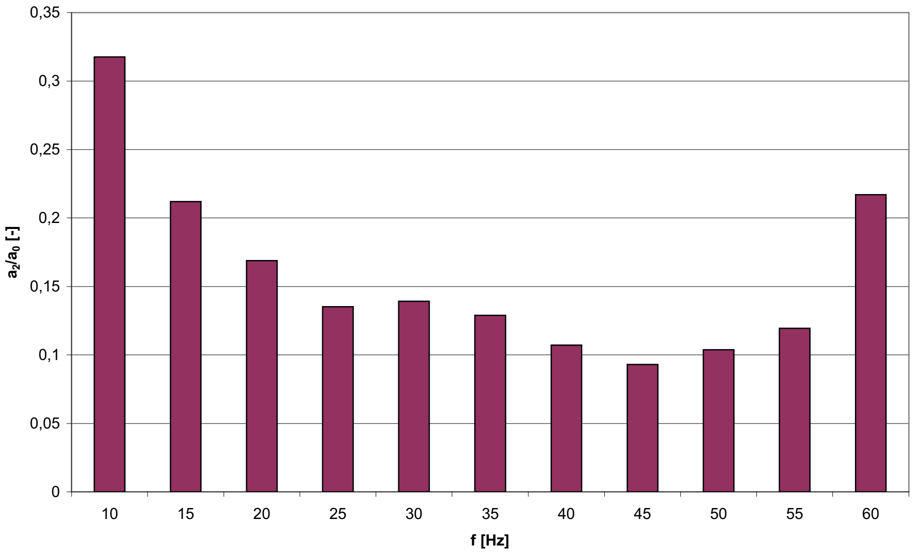

Fig. 6. Proportional distributor housing vibration acceleration amplitude a_2 relative to excitation vibration acceleration amplitude a_0 for f = 10–60 Hz

The results of the application of a vibration insulator with characteristic $c_2 \cdot x^2 + k_2 \cdot \dot{x}$ and c_2 = 20 000 N/m and k_2 = 50 Ns/m are shown in Figure 6. In this case, the fourth equation of model (1) should be supplemented with a nonlinear vibration insulator characteristic.

When a vibration insulator with a nonlinear damping characteristic (k_2 = 250 Ns/m) and linear stiffness (c_2 = 20 000 N/m) $c_2 \cdot x + k_2 \cdot \dot{x}^2$ is used to insulate base vibrations the valve housing vibrations are as shown in Figure 7.

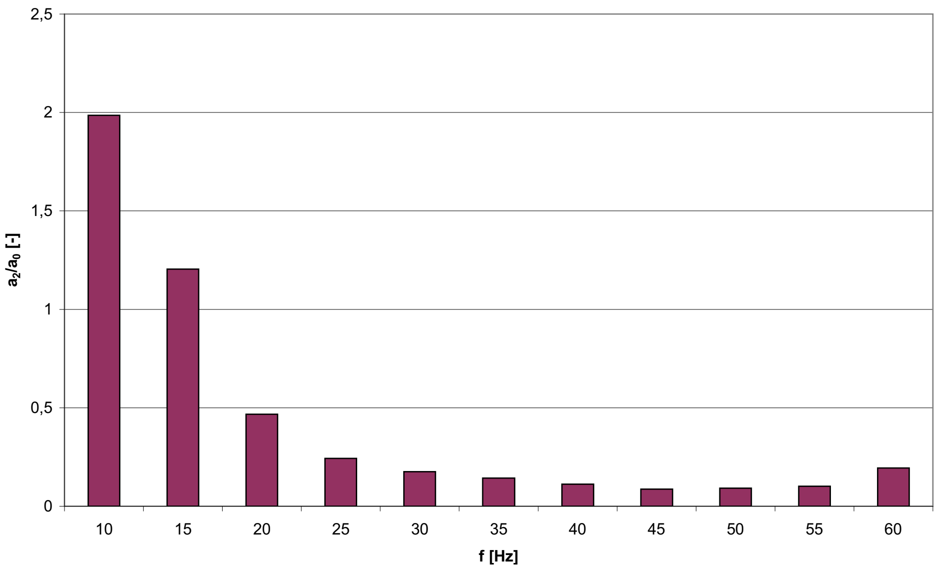

Fig. 7. Proportional distributor housing vibration acceleration amplitude a_2 relative to excitation vibration acceleration amplitude a_0 for f = 10–60 Hz

Figures 5 and 6 show that such a nonlinear vibration insulator characteristic can be selected that the insulation will be effective in the whole considered excitation frequency range.

The problem of influence of mechanical vibrations on valve was considered in theoretical and experimental way. Theoretical considerations were based on numerical calculations according to mathematical model. For some theoretical considerations experimental tests were done using test stand (hydraulic simulator, valve holder, spring set).

3. Experimental tests

A test rig enabling the generation of mechanical vibrations characterized by a prescribed frequency was built to experimentally verify the theoretical results and conclusions. The investigated valve – Mannesmann-Rexroth proportional distributor type 4WRE 6 E08-12/24Z4/M – fixed in the holder was mounted on the test rig and subjected to external mechanical vibrations (photo 1). Tests were done without pipes connected to valve.

A linear hydrostatic drive simulator Hydropax ZY25 made by Mannesmann-Rexroth, capable of generating vibrations up to 100 Hz, was the source of external mechanical vibrations. Main component of simulator of linear hydrostatic drive is servo valve which controls hydraulic cylinder. The simulator consists three main parts: hydraulic part, control part and control software. Displacement of simulator table is controlled by displacement transducer and its acceleration is controlled by accelerometer. On simulator table the tested valve was mounted. Electrical control signal for simulator was supported by external harmonic signal generator. The simulator is described in more detail in [4]. The proportional distributor was placed in the special holder and bilaterally supported with springs (there were two springs connected in parallel on each of the sides). Preliminary tests were carried out for springs with an equivalent stiffness of 86 000 N/m and a pre-deflection of 2 mm. The external excitation parameters are shown in Table 2.

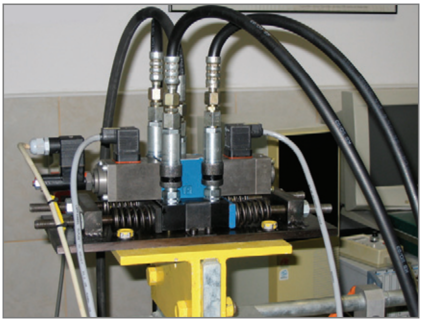

Photo 1. Proportional distributor placed in special holder and bilaterally supported with springs, during testing

Table 2. Amplitude of vibrations acting on tested hydraulic distributor

f [Hz]	w_0 [m]
30	0.000483
35	0.000406
40	0.000366
45	0.000269
50	0.000214
55	0.000145
60	0.0000522

Figure 8 shows an overall valve vibration diagram for the external excitation, i.e. a ratio of proportional distributor housing acceleration amplitude a_2 to excitation vibration amplitude a_0 versus a frequency of 25–60 Hz.

It appears from the diagram shown in Figure 8 that for a system of springs with equivalent stiffness c_z = 86 000 N/m and a proportional distributor with a mass of 4.5 kg the vibration insulation is effective (transmission function $a_2/a_0 < 1$) in the given external vibration frequency range. As a result of the insulation, the distributor housing vibration amplitude and the distributor slide-valve vibration amplitude decrease [5]. Consequently, the amplitude of the pressure fluctuations due to the excitation of distributor slide-valve vibrations also decreases. However, in the case of so simple vibration insulation, resonance may be generated at external vibration frequencies other than the ones used in the test. Therefore, as Figures 5 and 6 indicate, a vibration insulation element with other properties and characteristics, e.g. with nonlinear stiffness and with damping, should be used.

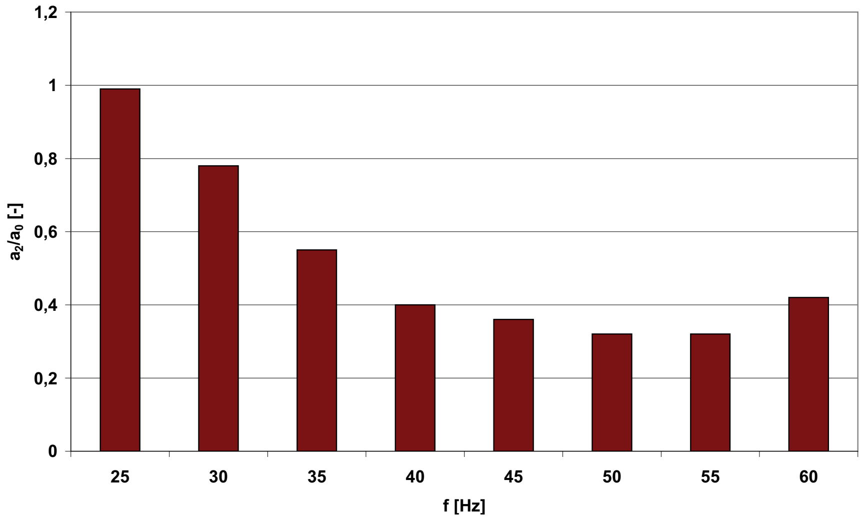

Fig. 8. Proportional distributor housing vibration acceleration amplitude a_2 relative to excitation vibration acceleration amplitude a_0 for f = 25–60 Hz

4. Conclusion

It has been shown that there is a need to reduce the vibration of the hydraulic valves with which machine tools and mobile machines are commonly equipped. The use of vibration insulators in the form of springs whose characteristics are linear results in a reduction in valve housing vibration acceleration amplitude at certain external vibration frequencies, but it may be conducive to resonance at other frequencies. Comparison of results presented on Figure 3 and Figure 8 shows, that differences between model and test are not great for frequency range 35–60 Hz. The biggest differences are observed in resonant area (25 Hz). It follows from the presented cases of vibration insulation (Figures 5–8) that materials with linear characteristics should be used in order to extend the range of effective vibration insulation. Thanks to the use of a vibration insulator with a nonlinear characteristic the valve housing vibration acceleration amplitude was reduced by a few tens of percent: by over 90% for the vibration insulator with quasi-zero stiffness and by about 80% for the vibration insulator whose stiffness or damping was proportional to displacement or velocity to the second power. A reduction in valve housing vibration will lead to a reduction in slide-valve vibration, particularly in the resonant vibration range. As a result, the pressure fluctuations and the emitted noise (particularly in a low frequency range) will decrease and the precision of the motions of the hydraulic receivers will increase. Vibration insulators in such applications should also satisfy other criteria, such as: resistance to changes in ambient temperature, resistance to hydraulic fluids, and small geometric dimensions. Therefore, besides having proper physicochemical properties, a vibration insulator should have a standardized design suitable for typical connection plates for hydraulic valves.

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Wibroizolacja elementów sterujących układów hydraulicznych

W artykule skupiono się na problemie oddziaływania zewnętrznych drgań mechanicznych na zawory hydrauliczne. Omówiono skutki tych oddziaływań. Przeprowadzono analizę teoretyczną wpływu charakterystyki wybranych izolatorów na redukcję drgań korpusu zaworu hydraulicznego. Przedstawiono wstępne badania eksperymentalne dla prostych przykładów izolatorów.