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Original Research Article

Ways of reducing the impact of mechanical vibrations on hydraulic valves

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ABSTRACT

This paper deals with the impact of mechanical vibrations on the environment, particularly on hydraulic valves. The main sources of such vibrations and their effects on hydraulic systems are indicated. Some documents setting down standard requirements for resistance to vibrations and to the noise generated by vibrations are cited. Two ways of reducing the impact of mechanical vibrations on the valve are proposed and a theoretical analysis, constituting the basis for selecting a material for an effective vibration isolator for the valve, is carried out.

1. Introduction

A running engineering machine is a source of mechanical vibrations with a wide spectrum of frequencies, including low frequencies [1–3]. The vibrations act on the operator inside the machine [4], on all the machine subassemblies and subsystems and indirectly, on the surrounding environment. For the sake of the health of the machine's valves, it is essential to identify the mechanical vibrations to which they are subjected. Such vibrations often may disturb the operation of the entire hydraulic system of a mobile machine. A disturbance in the operation of such a system is reflected in a change in the pressure fluctuation spectrum. The disturbance may lead to a deterioration in the accuracy of positioning the actuators, to uneven operation, shortening of the machine's life and sometimes to a higher level of low-frequency noise emitted [5]. Low-frequency vibrations and noise have a particularly adverse effect on hydraulic valves and the human being. In hydraulic valves they may excite the vibration of their control elements (such as the slide and the head) [6,7]. This occurs when the frequency of the external mechanical vibrations is close to that of the free vibrations of the valve control element. In the case of a human being, the vibrations via the skin mechanoreceptors transmit specific information to the central nervous system, causing reflex reactions of the human body [3,8,9]. The vibrations are accompanied by noise [10], also with low-frequency components. The noise is the subject of EU standard regulations. Hydraulic equipment producers, however, rather seldom specify the operating requirements concerning the resistance of their products (e.g. valves) to mechanical vibrations. One of such rare examples is the proportional distribution valve Parker–Hannifin D1FP, whose product data sheet [11] specifies the vibration value (about

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250 m/s^2) permissible with regard to its resistance to mechanical vibrations. The resistance of hydraulic valves to mechanical vibrations is usually tested in accordance with the relevant standards, such as, e.g. DIN-IEC68, part 2–6 [12] in the case of the D1FP distribution valve. Valve mechanical vibration resistance tests can also be conducted in accordance with other procedures described in Polish and international standards. Standard PN-IEC 68-2-59 1996 [13] specifies a method of testing electrotechnical subassemblies, equipment and other products (including hydraulic valves used in electrical applications) which during operation may be exposed to short-duration pulsating or oscillating forces generated by, e.g. seismic phenomena, an explosion or the vibrations of the machine in which they are installed. The tested product is excited by a certain number of constant-frequency sinusoidal beats. Standard PN-EN 60068-2-6:2008 [14] specifies a method of testing subassemblies, equipment and other products, which during transport or operation may be exposed to harmonic vibrations generated mainly by rotating, pulsating or oscillating masses. Such excitations occur in ships, planes, terrestrial vehicles and space vehicles. Resistance to mechanical vibrations in a frequency range of 5–3000 Hz is tested. A critical frequency identified by the test is a frequency at which faulty operation of the product or a deterioration in its properties due to vibration manifests itself or mechanical resonances (e.g. a valve control element) occur. Polish standard PN-EN ISO 4413:2011 [15] includes, among other things, requirements for the assembly of hydraulic valves (also pumps, servomotors, filters, etc.), but limited to a general statement that one should consider the effect of gravitation and vibrations on the valve.

In general, the vibrational resistance standards specify the permissible level of external mechanical vibrations which may adversely affect a machine, a piece of equipment or their components. In the precision industry, the experimentally determined maximum acceleration of about 0.981 m/s^2 has been adopted as the norm ensuring the vibrational resistance of measuring instruments and industrial equipment [16]. As regards rotor machines, they can be exposed to external mechanical vibrations below 9.81 m/s^2, without any adverse effect on their operation.

In this paper, the possibility of reducing the vibrations of the hydraulic distribution valve is studied. The experimental studies have been broadened with a general theoretical analysis of the problem of reducing the vibrations of a valve or a control element, which can be helpful in selecting the characteristics of antivibration materials and vibration damping systems for this purpose.

2. Experimental studies

Two ways of reducing the effect of external mechanical vibrations on hydraulic valve operation were investigated. One way consisted in flexibly fixing the distribution valve housing to a vibrating foundation while the other way consisted in introducing specially designed damping washers made of oil-resistant rubber into the distribution valve.

A linear hydrostatic drive simulator, capable of generating mechanical vibrations up to a frequency of 100 Hz, was used as the source of external vibrations. The simulator is described in more detail in [17]. As part of the experimental studies aimed at reducing distribution valve vibrations through flexible fixing, the acceleration of simulator table vibrations (excitation) and that of distribution valve housing vibrations (response) were measured (Fig. 1). In the case of the experiments aimed at reducing the vibrations of the distribution valve slide through the introduction of oil-resistant rubber washers into the distribution valve, the accelerations of both distribution valve vibrations (excitation) and slide vibrations (response) were measured (Fig. 2). The distribution valve was rigidly fixed in the simulator clamps.

PCB-ICP accelerometers, a signal conditioner VibAmp PA16000D, a Tektronix four-channel digital oscilloscope with

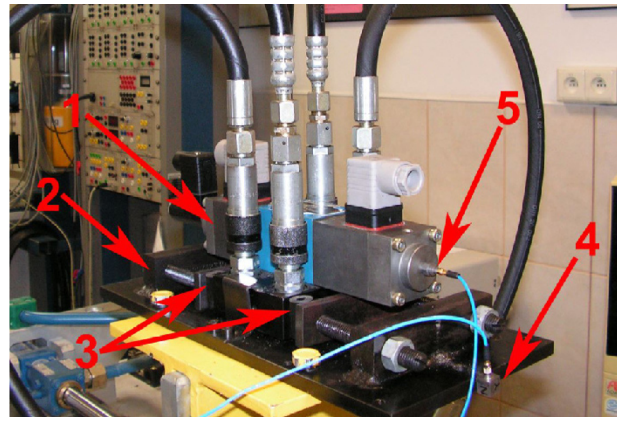

Fig. 1 – Way of fixing distribution valve and arrangement of accelerometers: 1 – hydraulic distribution valve, 2 – simulator table clamp, 3 – set of elastomer washers, 4 – accelerometer for measuring simulator table vibration acceleration (excitation), 5 – accelerometer for measuring distribution valve housing vibration acceleration.

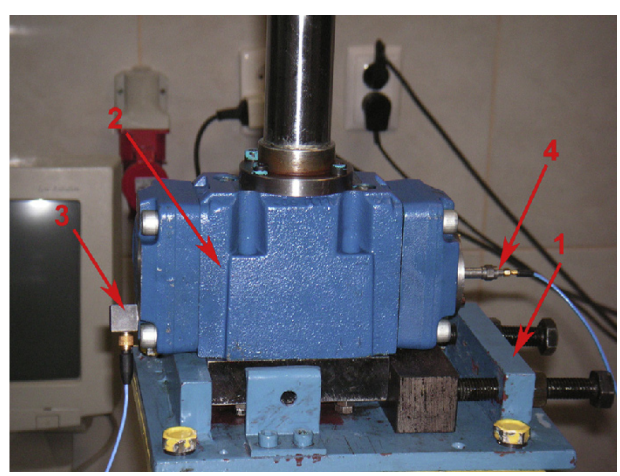

Fig. 2 – Way of fixing distribution valve and arrangement of accelerometers: 1 – simulator table clamp, 2 – hydraulic distribution valve, 3 – accelerometer for measuring distribution valve housing vibration acceleration (excitation), 4 – accelerometer for measuring distribution valve slide vibration acceleration (response).

dedicated software supplied by the producer and a PC for measurement data acquisition were used in the experiments.

2.1. *Identification of vibration isolator parameters*

Prior to the experimental studies aimed at determining the possibilities of reducing the impact of external mechanical vibrations on the hydraulic distribution valve, the parameters (stiffness and damping) of the materials later used to reduce the vibrations of both the distribution valve housing and the slide were experimentally identified using cylindrical and cuboidal specimens and modal analysis equipment [18,19]. A load would be applied by means of a modal hammer incorporating a force gauge while the system response would be registered by an acceleration gauge. The measurement signals were recorded and processed by a dedicated analyser. An analysis of the signals was carried out using a PC with special software.

The aim of the tests was to determine the elasto-dissipative properties of the selected elastomer elements. The tests were carried out on a special test rig as shown in Fig. 3. The test rig consisted of:

- a seismic mass of 49 kg,
- a vibrating mass of 12.9 kg,
- shock damping washers,
- and measuring equipment.

The measuring equipment consisted of a modal hammer (type SP 205) made by PCP Piezoelectronics, incorporating a force gauge for measuring the amplitude of the applied excitation, a PCB piezoelectric acceleration gauge for measuring the system response to the applied excitation and an HP analyser (type 35665A) recording the data from the gauges and converting them to transition characteristics.

The test rig was designed to minimize the influence of external vibrations on the motion of the vibrating mass. For this purpose a seismic mass characterized by great stiffness and weight and isolated from the foundation by means of damping washers was used.

Fig. 3 – Test rig: 1 – tested specimens; 2 – vibrating mass; 3 – seismic mass, 4 – damping washers.

The aim of the tests was to determine the frequency characteristics for each of the types of specimens. The characteristics were used to calculate the damping and dynamic stiffness parameters. The specimens would be placed on the seismic mass and a steel element constituting the vibrating mass would be placed on them. All the specimens of the same type would be arranged in the same precisely specified way. Because of the small size of the cylindrical specimens, four such specimens would be tested simultaneously in order to determine their elasto-dissipative properties. The specimens would be arranged to form a square with a side of 14 cm. This ensured the uniform loading of the specimens during the dynamic tests. The cuboidal specimens would be arranged in pairs parallel to one another so that the vibrating mass oscillated along the vertical axis.

In the case of the cylindrical elastomer washers, measurements would be conducted for two arrangements of the specimens. In the first arrangement the long axis (symmetry axis) of the specimen was oriented along the vertical axis, i.e. parallel to the direction of motion of the vibrating mass. In the second arrangement, the long axis of the specimen lay in the horizontal plane, i.e. it was oriented perpendicularly to the direction of vibrations. The specimens and their designations are shown in Fig. 4. The letter W represents a cylindrical specimen while the letter C or Z (in the second position) stands for the kind of material: Adipol 70 ShA and Ultraflex 64 ShA, respectively. The letters T and L indicate the direction of specimen loading: L – along the specimen's axis of symmetry, T – transversely to the symmetry axis. Digit 1 at the end stands for a 25 mm long specimen while digit 2 indicates a specimen length of 16 mm. For all the cylindrical elastomers the outside diameter was 16 mm and the inside diameter amounted to 6 mm.

Tests were also carried out on cuboidal specimens made of oil-resistant rubber that later would be placed in the clamps for reducing distribution valve vibrations. All the specimens had the same length and width, i.e. 101 and 26 mm, respectively, whereas their thickness amounted to 4, 12 and 15 mm. Their designations include digits indicating specimen thickness while the symbol TR represents a material with enhanced stiffness. The identification tests carried out on the test rig described above and shown in Fig. 3 yielded force–time and

Fig. 4 – Designations of cylindrical specimens – top view.

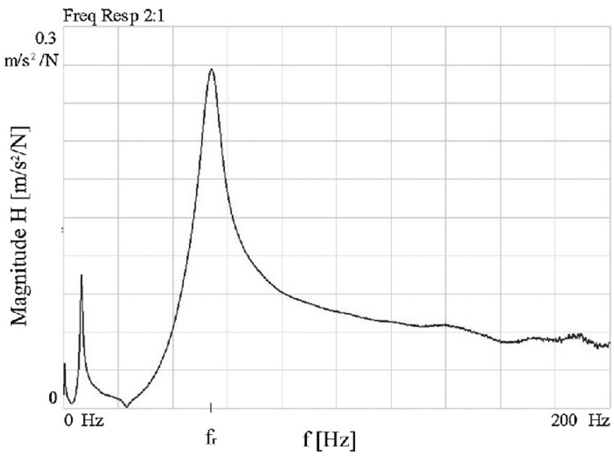

Fig. 5 – Frequency characteristic for cuboidal specimen 12 made of oil-resistant rubber; *f* – vibration frequency of vibrating mass (item 2, Fig. 3), f_r – vibrating mass natural frequency.

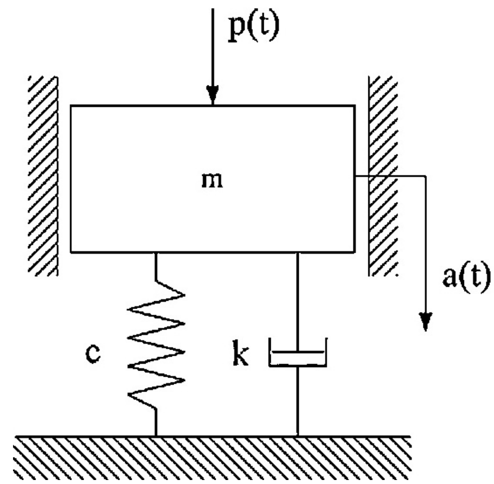

Fig. 6 – Schematic of dynamic model: *c* – stiffness parameter, *k* – damping parameter, *m* – vibrating mass.

acceleration–time diagrams which were used to determine the transition characteristic by means of the HP 3566 analyser. An exemplary experimental transition characteristic for the specimen designated with symbol 12 is shown in Fig. 5.

From the frequency characteristics the natural frequency of the elastomer-vibrating mass system and the corresponding modal damping were determined. Then the damping and dynamic stiffness of the tested material were calculated.

The modal damping was calculated, using the half-power method [19], from the formula:

$$\xi_r = \frac{\omega_2 - \omega_1}{2\omega_r} \tag{1}$$

where ω_1 and ω_2 were calculated from the relation:

$$|H(j\omega_1)| = |H(j\omega_2)| = \frac{|H(j\omega_r)|}{\sqrt{2}} \tag{2}$$

where $H(j\omega_r)$ – an amplitude value [m/s^2/N] corresponding to natural angular frequency ω_r, $H(j\omega_1)$, $H(j\omega_2)$ – amplitudes [m/s^2/N] corresponding to angular frequency ω_1 and ω_2, lying on both sides of natural angular frequency ω_r, consistently with the condition: (2), $\omega_r = 2\pi f_r$, f_r – natural frequency.

In order to describe the dynamic properties of the tested materials one can use a dynamic model with one degree of freedom (Fig. 6) [19], consisting of linear elasto-dissipative elements connected in parallel (a Kelvin–Voigt body) where mass m corresponds to the vibrating mass used in the test.

Considering that the tested elements were so arranged that they could be treated as connected in parallel, the unit stiffness and damping for the cylindrical specimens can be calculated from the formulas:

$$c = \frac{c_z}{4}; \quad k = \frac{k_z}{4} \tag{3}$$

where k_z and c_z are respectively the equivalent damping value and the equivalent stiffness value for the set of tested specimens, and for the cuboidal specimens from:

$$c = \frac{c_z}{2}; \quad k = \frac{k_z}{2} \tag{4}$$

The calculated parameters are presented in Tables 1 and 2.

The stiffness c and damping k parameter values experimentally determined for the materials were then used in studies aimed at reducing the impact of external vibrations on the distribution valve and the slide. The obtained results also show, for all the types and dimensions of the elastomers, that when the long axis of the specimen was oriented parallel to the direction of motion of the vibrating mass, the tested elements were characterized by greater damping and stiffness than when the long axis was oriented perpendicularly to the direction of load application. Moreover, the elastomers made of Ultraflex 64 ShA are characterized by much higher damping and stiffness values than the elastomers made of Adipol 70 ShA. In the case of the cuboidal specimens, damping and stiffness were found to decrease with increasing specimen thickness.

Table 1 – Damping *k* and stiffness *c* parameters for cylindrical specimens.

	Damping k [kg/s]	Stiffness c [kg/s^2]
WCL1	10.97	9.70E+4
WCT1	7.26	4.30E+4
WCL2	9.27	8.04E+4
WCT2	8.47	6.49E+4
WZL1	7.01	8.41E+4
WZT1	3.55	2.79E+4
WZL2	4.69	5.55E+4
WZT2	4.43	4.30E+4

Table 2 – Damping *k* and stiffness *c* parameters for cuboidal specimens.

	Damping k [kg/s]	Stiffness c [kg/s^2]
4	112.88	2.16E+6
12	59.02	7.50E+5
15	52.57	7.16E+5
TR4	149.96	4.08E+6
TR12	91.27	1.44E+6
TR15	66.11	1.16E+6

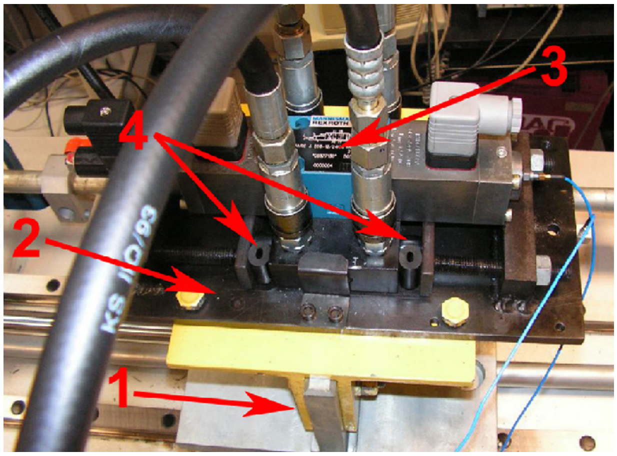

Fig. 7 – Hydraulic distribution valve fixed in clamps with flexible antivibration insulation elements: 1 – simulator table, 2 – clamp, 3 – hydraulic distribution valve, 4 – set of flexible antivibration insulation elements.

2.2. *Reduction of distribution valve housing vibrations*

A series of experiments were carried out in which the distribution valve would be fixed in the simulator table clamps, using sets of antivibration insulation elements (whose elasto-dissipative properties had been experimentally identified – Tables 1 and 2) in different configurations (Figs. 7 and 8). The distribution valve would be excited with harmonic simulator table vibrations with a frequency of 10–100 Hz and a known amplitude.

Different sets of elastomer damping washers would be placed on both sides of the distribution valve in the clamps. In order to assess the effectiveness of the antivibration insulation provided by the elastomer elements the ratio of the distribution valve vibration acceleration amplitude (response) to the simulator table vibration acceleration amplitude (excitation) was determined. It was assumed that antivibration insulation was effective when the following inequality was satisfied:

$$\frac{a_2}{a_0} < 1 \tag{5}$$

where a_2 – the amplitude of distribution valve housing vibration acceleration [m/s^2], a_0 – the amplitude of simulator table vibration acceleration [m/s^2].

Five different sets of vibration isolators were used in the tests. The particular sets differed from each other in the number of elastomer elements which they consisted of, and, for the cylindrical elements, in the direction of load application relative to the long axis. Hence the sets were characterized by different equivalent stiffness c_z values and different equivalent damping k_z values, as shown in Table 3.

The experimental results in the form of a diagram showing the distribution valve acceleration amplitude/simulator table vibration acceleration amplitude ratio depending on the excitation frequency are presented in Fig. 9.

It appears from the results presented in Fig. 9 that such a set of washers can be designed which will ensure effective antivibration insulation in nearly the whole considered range of frequencies.

2.3. *Reduction of slide vibrations*

In order to explore the possibilities of reducing slide vibrations, experiments were carried out in which specially shaped washers made of oil-resistant rubber would be installed inside the distribution valve, i.e. between the valve housing and the slide centring springs. The distribution valve would be fixed directly in the clamps of the simulator table. The simulator table vibrations ranged from 10 Hz to 100 Hz. The arrangement of the washers inside the distribution valve housing and their shape are shown in Fig. 10.

Because of the distribution valve design, washers with an outside diameter of 26 mm, an inside diameter of 22 mm and a height of 4 mm were used. The dynamic model of the slide inside the valve housing is shown in Fig. 11. The slide having

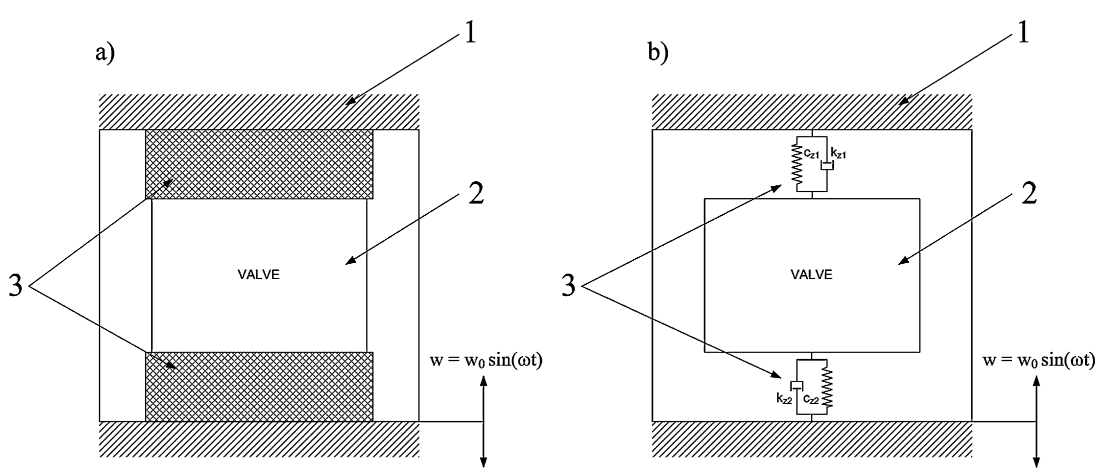

Fig. 8 – Schematic of flexible fixing of distribution valve to foundation: (a) location of flexible elements, (b) dynamic model of flexible elements: 1 – vibrating simulator table clamp, 2 – hydraulic distribution valve, 3 – set of flexible antivibration insulation elements.

Table 3 – Equivalent stiffness c_z and equivalent damping k_z values for selected sets of washers.

Washer set number	Equivalent stiffness c_z [N/m]	Equivalent damping k_z [kg/s]
1	28.36E+04	43.24
2	2.32E+06	132.22
3	54.36E+04	55.84
4	65.52E+04	70.04
5	71.56E+04	84.88

mass m_1 is centred by springs each with known stiffness c_{s1}. Moreover, damping k_{s1} occurs on both sides of the slide.

The washers are represented by a model consisting of linear elasto-dissipative elements connected in parallel (the Kelvin–Voigt model), with one degree of freedom, where stiffness is denoted with c_{s1} and damping with k_{k1}, for each of the two washers.

The centring spring stiffness in the tested distribution valve was c_{s1} = 2900 N/m. The washer stiffness was c_{c1} = 1.41E+06 N/m. Therefore, assuming the connection to be linear, the equivalent stiffness on both sides of the slide amounts to 2894 N/m. Whereas the equivalent stiffness on both sides of the slide for the parallel connection amounts to 5788 N/m. In a similar way the equivalent damping for one side was found to amount to 8.66 kg/s while the equivalent damping on both sides of the slide for the parallel connection amounted to 17.32 kg/s. The experimental results showing the effect of the use of the washers in reducing the slide vibrations are presented in Fig. 12.

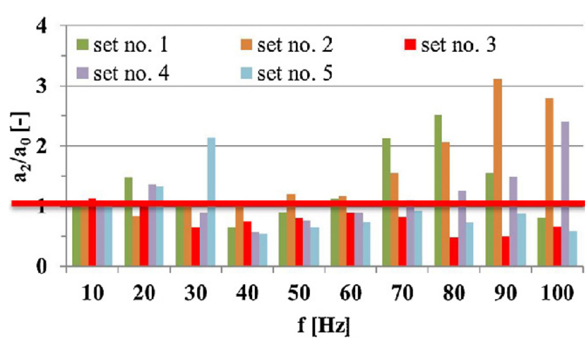

Fig. 9 – Effectiveness of damping valve vibrations excited by foundation vibrations for different sets of flexible washers: a_2 – valve housing vibration acceleration amplitude, a_0 – simulator table vibration acceleration amplitude.

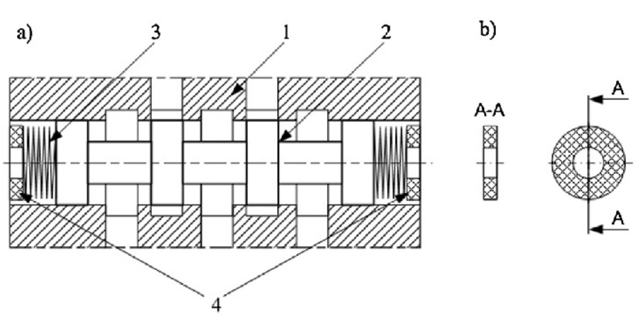

Fig. 10 – Oil-resistant rubber washers installed inside distribution valve housing: (a) location inside valve housing: 1 – vibrating valve housing, 2 – valve slide, 3 – centring springs, 4 – flexible washers, (b) washer shape.

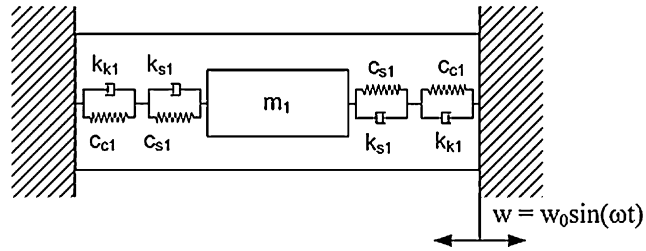

Fig. 11 – Dynamic model of slide in vibrating distribution valve housing.

An analysis of the equivalent stiffness on each side of the slide indicates that it is decisively influenced by the lower of the values, i.e. the spring stiffness in this case. This is due to the way in which the stiffness is connected, i.e. in series. The results presented in Fig. 12 show that the adopted solution does not result in a significant reduction of slide vibrations. Therefore the effect of a change in stiffness and damping on the amplitude of the vibration of the slide excited into vibration by the vibrations of the valve housing should be examined. In the case of distribution valves controlled by proportional electromagnets, one should take into account the maximum forces (typically 15–20 N [20,21]) generated by such electromagnets with an adjusted step, and the size of the latter (typically about 4.5 mm [20,21]) required by the slide pair design. This puts constraints on the washer stiffness value since the controlling force generated by the electromagnet must be greater than the sum of the other forces, i.e. the friction force in the slide pair, the inertial force of the slide and the associated fluid, the equivalent stiffness force of the centring springs and washers and the hydrodynamic force.

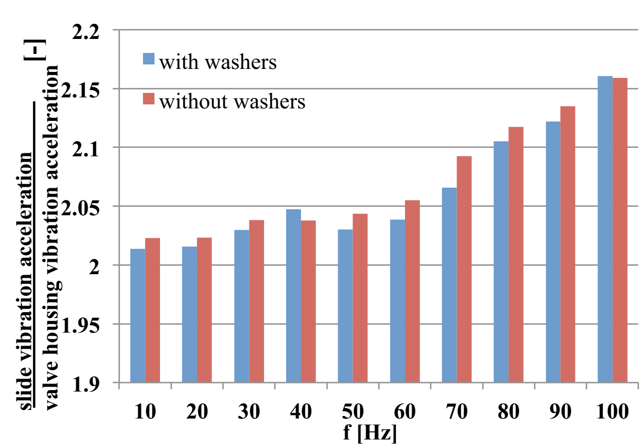

Fig. 12 – Relative slide vibration acceleration/valve housing vibration acceleration ratio versus valve housing vibration frequency.

3. Theoretical analysis of vibration reduction possibilities

The experimental studies indicate that when selecting a way of reducing the impact of external mechanical vibrations on the hydraulic distribution valve and its slide one should take into consideration the constraints relating to the dimensions of vibration isolators and their stiffness c and damping k. Geometric constraints stem from the incorporation conditions, i.e. the space available for installing a vibration isolator (e.g. a set of elastomer washers). Isolator stiffness c constraints stem from the maximum power generated by the slide controlling element (e.g. a proportional electromagnet).

The theoretical analysis can be based on the same mathematical model used in two ways: one can consider a reduction in valve housing vibration as an excitation acting on the valve slide or a reduction in the relative vibrations of the slide inside the valve housing. In both cases one can use a single mass model with one degree of freedom, in which m represents, depending on the considered case, the distribution valve mass or the slide mass, where c and k are respectively the equivalent stiffness and the equivalent damping of the flexible washers between the distribution valve housing and the clamps or the equivalent stiffness of the flexible washers and the centring springs and the equivalent damping of the flexible washers of the slide pair (Fig. 13).

A body having mass m is excited into vibration by a kinematic excitation in the form of a harmonic function expressed by the equation:

$$w = w_0 \sin(\omega t) \tag{6}$$

where w_0 – vibration amplitude [m], $\omega = 2\pi f t$ [rad/s], f – frequency [Hz], t – time [s].

The absolute motion of the body (the distribution valve housing or the slide) is described by the equation:

$$m \cdot \ddot{x} + k(\dot{x} - \dot{w}) + c(x - w) = 0 \tag{7}$$

The relative displacement of the body with mass m and the vibrating foundation (e.g. a distribution valve housing) can be written as:

$$y = x - w \tag{8}$$

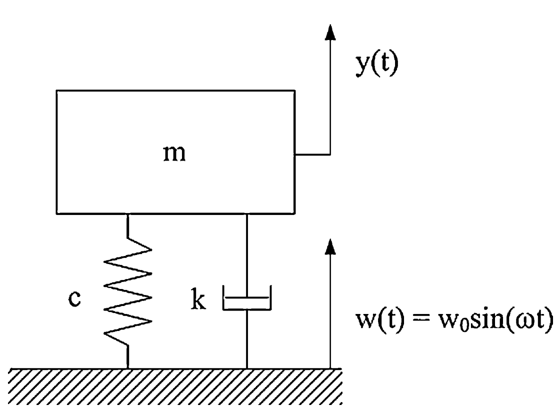

Fig. 13 – Model of vibrating system with one degree of freedom.

Thus the relative motion equation assumes the form:

$$m\ddot{y} + m\ddot{w} + k\dot{y} + cy = 0 \tag{9}$$

Considering that w is a known time function, Eq. (9) can be written as:

$$\ddot{y} + 2h\dot{y} + \omega_0^2 y = -\ddot{w}(t) \tag{10}$$

and after the harmonic form of the kinematic excitation is taken into account, as:

$$\ddot{y} + 2h\dot{y} + \omega_0^2 y = w_0 \omega^2 \sin(\omega t) \tag{11}$$

where

$$h = \frac{k}{2m} \tag{12}$$

$$\omega_0^2 = \frac{c}{m} \tag{13}$$

The relative vibrations of the distribution valve slide are described by the equation:

$$y(t) = B_a w_0 \sin(\omega t + \delta) = B_a w(t + \tau) \tag{14}$$

The slide vibrations are proportional to the housing vibrations, but they are shifted by time $\tau = \delta/\omega$. Coefficient B_a, which can be called a transfer factor, amounts to [22]:

$$B_a = \frac{(\omega/\omega_0)^2}{\sqrt{(1 - (\omega^2/\omega_0^2))^2 + 4(h^2/\omega_0^2)(\omega^2/\omega_0^2)}} \tag{15}$$

or

$$B_a = \frac{y_0}{w_0} \tag{16}$$

where y_0 – the amplitude of relative slider displacement, w_0 – the amplitude of valve housing vibrations.

For given kinematic excitation parameters one can plot a spatial diagram of the dependence between transfer factor B_a and the equivalent stiffness c (of the centring springs and the flexible washers) and the equivalent damping k inside the distribution valve housing. Fig. 14 shows such a dependence for the assumed parameters: slider mass $m = 0.18$ kg and excitation frequency $f = 80$ Hz.

In order to reduce relative slide vibrations one should select such an equivalent stiffness c and/or an equivalent damping k inside the distribution valve housing that $B_a < 1$. Assuming an

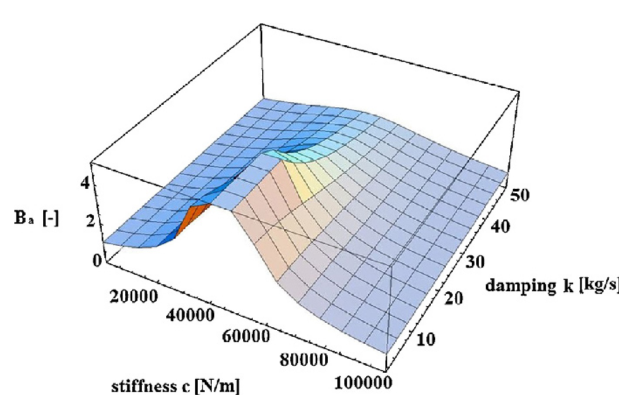

Fig. 14 – Dependence between transfer factor B_a and equivalent stiffness c and damping k inside distribution valve.

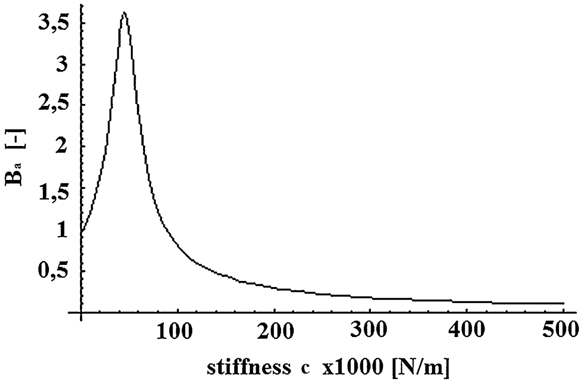

Fig. 15 – Factor B_a as function of equivalent stiffness c for adopted damping k and excitation parameters.

equivalent damping k inside the valve housing equal to 25 kg/s and $f = 80$ Hz, one can determine the equivalent stiffness c value for which $B_a < 1$ (Fig. 15).

It follows from Fig. 15 that equivalent stiffness c should be higher than 100,000 N/m in order to avoid slider resonance for the adopted excitation parameters. However, considering the controlling forces, this is not possible to achieve because of the forces (typically about 15 N) generated by the proportional electromagnets. Assuming a slide stroke of 4.5 mm, one gets the maximum equivalent stiffness of 3333 N/m (excluding the other forces which the controlling force generated by the electromagnets must overcome).

If the equivalent stiffness c equal to 3000 N/m and the same excitation parameters as above are assumed, one can examine the influence of equivalent damping k on factor B_a (Fig. 16).

According to Fig. 16, the equivalent damping k for the adopted equivalent stiffness c should be greater than 30 kg/s.

A similar analysis, aimed at minimizing the amplitude of absolute housing vibrations, can be carried out for the distribution valve housing [23]:

$$x_0 = w_0 \cdot \sqrt{\frac{1 + (2\gamma(\omega/\omega_0))^2}{(1-(\omega/\omega_0)^2)^2 + (2\gamma(\omega/\omega_0))^2}} \rightarrow \min \tag{17}$$

where w_0 – the excitation amplitude [m], ω – the excitation angular frequency [rad/s], ω_0 – the system natural angular frequency [rad/s], γ – a dimensionless damping coefficient, $\gamma = (h/\omega_0) = (k/2m\omega_0)$.

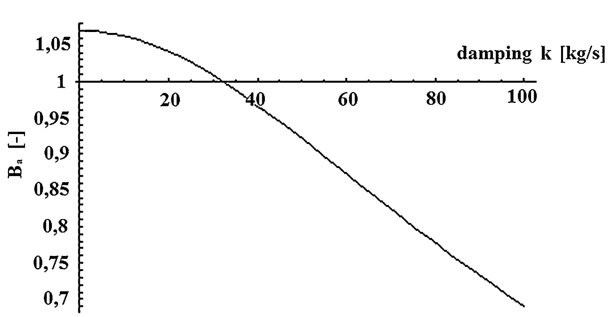

Fig. 16 – Factor B_a as function of equivalent damping k for adopted stiffness c and excitation parameters.

In addition, one can use the antivibration insulation effectiveness calculated from the relation [3]:

$$\varepsilon = \left(1 - \frac{x_0}{w_0}\right) \times 100\% \tag{18}$$

In this case, such parameters c and k should be selected that the antivibration insulation effectiveness, calculated from formula (18), is as high as possible.

4. Conclusions

The results of the experiments and the theoretical analysis indicate that in order to reduce slider vibrations one can introduce shock damping washers (made of a material characterized by high stiffness c and damping k) into the distribution valve housing, between the housing and the centring springs. In the case of distribution valves with single-step electric (e.g. proportional) control, this approach has constraints because of the maximum values of the controlling forces generated by the proportional electromagnets, and the required slide stroke. Another possible way of reducing distribution valve housing vibrations, and consequently of slider vibrations, is to mount the distribution valve on flexible washers whose equivalent stiffness and equivalent damping can be calculated from relation (17) for the set excitation parameters and the mass of the vibrating valve. The experimental results presented in Figs. 9 and 12 indicate that although the materials used reduce slider or housing vibrations, the antivibration insulation effectiveness is not satisfactory. Therefore, mainly for the purposes of isolating distribution valve housing vibrations, one should search for other materials, using the criteria defined by relations (17) and (18).

On the basis of the theoretical analysis the following simplified and generalized procedure for selecting antivibration insulation is proposed [3]:

1. Determine the required antivibration insulation effectiveness; usually the effectiveness level of 70% is sufficient in engineering practice.
2. For the antivibration insulation effectiveness determined in pt. 1, specify the maximum value of ratio x_0/w_0, using Table 4 [3].
3. Determine the value of the ratio of excitation angular frequency ω to distribution valve natural angular frequency ω_0. For this purpose determine the lowest value of excitation angular frequency ω.

Table 4 – Effectiveness of antivibration insulation [3].

Antivibration insulation effectiveness [%]	Maximum antivibration insulation factor	Required ratio ω/ω_0
90	0.1	3.32
80	0.2	2.45
70	0.3	2.08
60	0.4	1.87
50	0.5	1.73

4. From the nomograph used for determining antivibration insulation parameters select natural angular frequency ω_0 of the distribution valve to be insulated, which is needed to ensure the antivibration insulation factor value determined in pt. 2. The nomograph can be found in the literature, e.g. [3,24].
5. From the nomograph determine the static deflection corresponding the natural angular frequency ω_0 specified in pt. 4.
6. Calculate equivalent stiffness c, from relation (13).
7. Calculate the stiffness of the particular insulating elements, using relation $c_i = c/n$, where n – the number of insulating elements;
8. Calculate the load per insulating element.
9. Select an antivibration insulation system from the typical commercial antivibration insulation products or design your own system satisfying the above requirements.

REFERENCES

[1] J. Sadowski, R. Rogiński, Noise Control in Public Transport, WKŁ, Warsaw, 1965 (in Polish).
[2] J. Grajnert, Vibration Insulation of Machines and Vehicles, Wrocław University of Technology Publishing House, Wrocław, 1997 (in Polish).
[3] Z. Engel, M. Zawieska, Noise and Vibration in Work Processes: Sources, Assessment, Hazards, Central Institute for Labour Protection – National Research Institute, Warsaw, 2010 (in Polish).
[4] M. Demic, J. Lukic, Investigation of the transmission of fore and aft vibration through the human body, Applied Ergonomics 40 (2009) 622–629.
[5] N. Chenxiao, Z. Xushe, Study on vibration and noise for the hydraulic system of hydraulic hoist, in: Proceedings of 2012 International Conference on Mechanical Engineering and Material Science (MEMS), 2012.
[6] M. Stosiak, The effect of ground mechanical vibrations on pressure fluctuation in the hydraulic system, Hydraulika i Pneumatyka 26 (3) (2006) 5–8 (in Polish).
[7] M. Stosiak, Vibration insulation of hydraulic system control components, Archives of Civil and Mechanical Engineering 11 (1) (2011) 237–248.
[8] J. Renowski, Noise: Indices and Evaluation Criteria, Wrocław University of Technology Publishing House, Wrocław, 1988 (in Polish).
[9] Z. Kudźma, M. Stosiak, Reduction of infrasounds in machines with hydrostatic drive, Acta of Bioengineering and Biomechanics 15 (2) (2013) 51–64.
[10] H. Ortwig, Experimental and analytical vibration analysis in fluid power systems, International Journal of Solids and Structures 42 (2005) 5821–5830.
[11] D1FP – Parker–Hannifin valve data sheet.
[12] DIN IEC 68 Part 2–6, Fundamental environmental test procedures, sinusoidal oscillation.
[13] PN-IEC 68-2-59:1996, Environmental research. Fe test: vibration – sinusoidal beat method (in Polish).
[14] PN-EN 60068-2-6:2008, Environmental research – part 2–6: tests – Fe test: vibration (in Polish).
[15] PN-EN ISO 4413:2011, Hydraulic drives and controls – general guidelines for systems (in Polish).
[16] R. Łączkowski, Vibroacoustics of Machines and Equipment, WNT, Warsaw, 1983 (in Polish).
[17] W. Kollek, Z. Kudźma, M. Stosiak, Simulator of linear hydrostatic drive as source of new research possibilities, in: Proceedings of Hydraulic and Pneumatic Drives and Controls 2005, Problems and Development Trends in the First Decade of the 21st Century, International Scientific-Technical Conference, 2005.
[18] T. Kucharski, System of Measuring Mechanical Vibrations, WNT, Warsaw, 2002 (in Polish).
[19] M. Kulisiewicz, S. Piesiak, Methodology for Modelling and Identifying Dynamic Mechanical Systems, Wrocław University of Technology Publishing House, Wrocław, 1994 (in Polish).
[20] E. Tomasiak, Hydraulic and Pneumatic Drives and Controls, Silesian Polytechnic Publishing House, Gliwice, 2001(in Polish).
[21] A. Osiecki, Hydrostatic Drive for Machines, WNT, Warsaw, 1998 (in Polish).
[22] Z. Osiński, Theory of Vibrations, PWN, Warsaw, 1980 (in Polish).
[23] J. Goliński, Vibro-Insulation of Machines and Equipment, WNT, Warsaw, 1979 (in Polish).
[24] C.M. Harris, A.G. Piersol, Shock and Vibration Handbook, McGraw-Hill, New York, 2002.