Coincidence of pressure pulsations with excitation of mechanical vibrations of hydraulic system components. An experimental study

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Abstract

This paper discusses certain excitations that affect the components of hydraulic systems composed of pipes and valves. The negative effects of low-frequency noise and vibration on humans are also pointed out. Particular attention is given to the effect of pulsatile flow on the structure of hydraulic components. Pressure pulsations in a hydraulic system are shown to generate and transmit mechanical vibrations across a wide spectrum of frequencies, and the negative consequences of this phenomenon are pointed out. Based on the latest generation of proportional directional control valves, a special stand was built to generate pressure pulsations over a wide frequency range (up to 350 Hz). Amplitude-frequency spectra of mechanical vibrations and pressure pulsations were used instead of time courses in the considerations. The paper concludes that effort must be made to reduce the amplitudes of pressure pulsations in hydraulic systems, particularly in the low-frequency spectrum.

Keywords
vibrations, pressure pulsation, hydraulic systems, signals coincidence

1 Introduction

A particular property of hydraulic systems is that their operation depends on ambient conditions and the instantaneous values of flow rate and pressure. An operating hydraulic component, e.g. a directional or pressure control valve, is constantly subjected to complex excitations of various origins, including external vibrations coming from the ground, flow rate pulsation and pressure pulsation. It bears noting that any excitation applied at any point of the hydraulic system is transferred to the components of that system through viscous or elastic components. Meanwhile, the effects of this phenomenon depend on the location affected by the excitation, its amplitude and frequency, the physical properties of the components transferring these excitations, and the configuration of the hydraulic system.

Those elements of a hydraulic system whose essential part is a control component in the form of a spool, ball, cone, plate, etc. are subjected to variable loads resulting from changes in operating parameters: the flow rate of the medium, pressure caused by inconsistent positive displacement pump efficiency, and variable external load (Stryczek, 2014). These loads are frequently stochastic and may be harmonic in specific cases. Generally speaking,

excitations affecting hydraulic valves can be divided into intentional and interferential groups. Intentional excitations include signals controlling the operation of valves forming part of a control or regulation system (Glanowski, 2001; Tomasiak, 2001). Excitations that interfere with the operation of valves include vibrations affecting the valve (Kollek et al., 2008; Stosiak, 2006). The issue of the impact of these excitations on the operation of the valve control component applies to all hydraulic systems operating in real conditions, and, in particular, those subject to increased requirements as to the manufacture of precision elements and response to control signals. Due to the above, a significant impact of external signals on the operation of modern proportional elements or hydraulic microvalves is to be expected, as interferential excitation forces in these components may be of the same size as the controlling forces, leading to many detrimental effects that include stability loss, lack of positional precision, sealing damage, and increased noise generation (Kollek et al., 2010). At present, there is a strong tendency for proportional control technologies to be developed in various types of hydraulic components (Jesionek et al., 2004b; Tomasiak, 2001), replacing the previously used conventional components and opening up new possibilities with regard to time of response to control signals (Cichoń and Stosiak, 2011) and bandwidth frequency, and completely new opportunities for performing pre-programmed work cycles, reducing dynamic surplus and mitigating transitional states (Kudźma and Stosiak, 2013; Bury et al., 2022). Modern medical devices and industrial robots are equipped with proportional components such as proportional directional valves, proportional overflow valves, electrohydraulic boosters, and increasingly commonly, hydraulic microvalves (Jesionek et al., 2004a; Kollek (ed.), 2011).

Irrespective of how they are generated, the mechanical vibrations of hydraulic system components affect the human body. An initial analysis of the impact of vibration on humans may be performed after determining the scope of resonant vibrations of the human body treated as a multibody system. Figure 1 shows frequencies of normal mode vibrations of certain organs in the human body (Engel and Zawieska, 2010). Numerous experiments demonstrate that the nervous system and the circulatory system are the parts of the body that are most sensitive to vibrations.
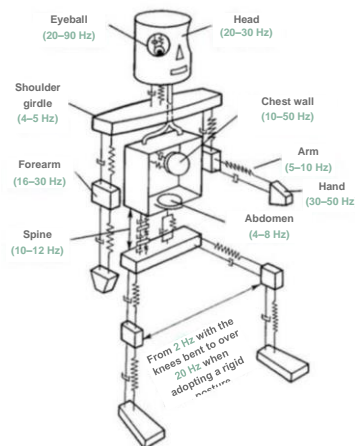

Figure 1 Simplified diagram of the elastic system in the human body (Engel and Zawieska, 2010: p168).

Through mechanoreceptors in the skin, vibrations transfer specific information to the central nervous system, causing reflexive reactions of the entire body. Long-term exposure to mechanical vibrations causes irreversible changes in various organs and systems. These changes can be divided into (Engel and Zawieska, 2010):

- acute changes that occur during exposure and for a short time afterwards; they involve specific changes in the behaviour of the entire body seen as a mechanical system;
- chronic changes that occur as a result of long-term exposure to intense vibrations.

The maximum exposure limits of the human body are often exceeded, causing a disorder known as vibration syndrome (Fig. 2) (Engel and Zawieska, 2010).
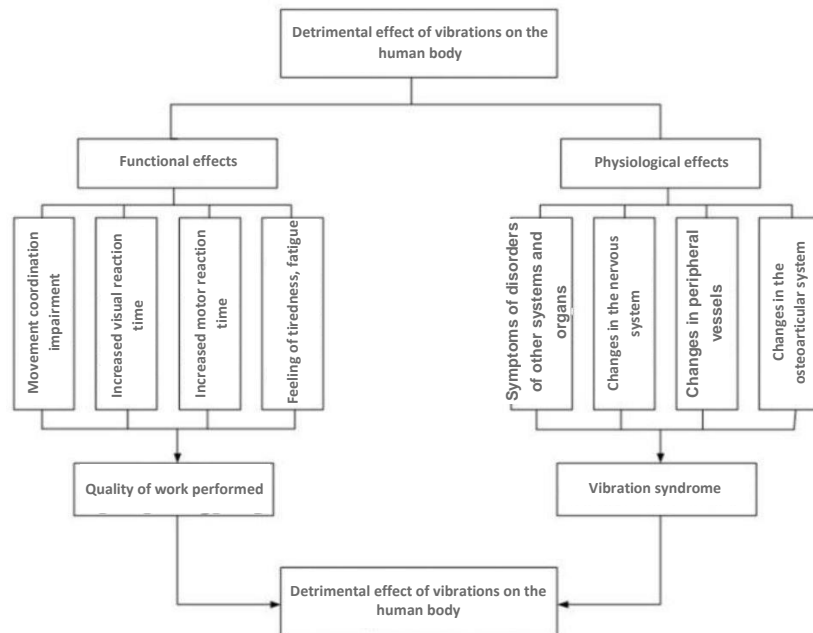
Figure 2 Effect of vibrations on the human body (Engel and Zawieska, 2010: 173).

The harmful effect of vibrations on machines and devices is determined by measuring the amplitude of the effective speed of vibrations in the place where the machine or device is installed and comparing it against the permissible values, shown in Table 1. For class II–V sensitivity, the permissible amplitude is compared against the maximum vibration value present in a given direction, whereas for class I sensitivity, it is compared against the modulus of the velocity vector $v_{dop} = \sqrt{v_x^2 + v_y^2 + v_z^2}$, where v_x, v_y, v_z are the components of the velocity vector in the x, y, z directions.

Table 1. Classes of sensitivity to external mechanical vibrations for groups of machines and devices according to (PN-ISO 9612:2004).

Sensitivity class	Sensitivity description	Name of machine or device group	permissible velocity $v_{dop} \times 10^{-3}$ [m/s]
I	high sensitivity	devices used to balance and adjust instruments, microscopes, interferometers and other precision tools, computers, precision machining tools	0.1
II	moderate sensitivity	cog and bearing sanders; precision milling machines and lathes	1
III	low sensitivity	regular milling machines and lathes, power drills, sanders, textile weaving and printing machines	3
IV	nearly no sensitivity	motors, mortisers, sewing machines, metal and wood machining tools, presses, cutters	6
V	no sensitivity	fans, crushers, mills, agitators, vibration tables and sieves, screening machines, hammers	> 6

Table 1 shows that all sensitivity classes include groups of machines equipped with hydraulic components and systems. Class I machines are expected to be particularly susceptible to mechanical vibrations, and these include precision instruments and machine tools equipped with hydrotronic and microhydraulic components which are subject to increased requirements with regard to, for example, precision and repeatability of operation (Stosiak et al., 2020).

The mechanical vibrations of machines and system components also generate noise in a wide spectrum of frequencies. Low-frequency vibrations and noise, called infrasounds, have a particularly adverse effect on the human body; see Table 2.

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Table 2. List of pathological symptoms caused by the impact of infrasounds on the human body (Chaban et al., 2021; Tonin et al., 2016; Gužas and Viršilas, 2009; Stosiak, 2015): f – frequency, L_m – acoustic pressure level, t_{exp} – time of exposition, N – number of appearances.

f [Hz]	L_m [dB]	t_{exp} [s]		Symptoms
2–15	105			Increased visual response time in 50% of subjects, balance disorders in 10%
1–2	150			Change of threshold of hearing, feeling of eardrum displacement
2–15	110–120		7	Reaction time increased by 4%
3–15	115	1800		Symptoms similar to alcohol abuse
10	135	900	6	Sensation of internal organs vibrating, feeling of eardrum membrane vibrations, pain in the middle ear, increased pulse, increased arterial pressure, quickened breath by 4 or more breaths per minute, change of threshold of hearing by 15–20 dB during testing and by 8–10 dB immediately after the end of the test
7	90	2100	30	Reduced blood pressure, reduced heart rate, heart murmurs in half of subjects
2–22	119–144	180	30	Change of threshold of hearing by 10 dB in 30% of subjects
1–100	154	24–120		Vertigo, headache and dizziness, dyspnea, tingling in the ears, trismus

The above considerations lead to the conclusion that vibrations of machine and device components result in clearly adverse effects, which include:

- incorrect operation of the machine (including the irregular operation of hydraulic receivers, generation of pressure pulsations resulting from the excitation of vibrations of control components in hydraulic valves);
- reduced durability of the machine or device;
- harmful effect of vibrations (Chenxiao and Xushe, 2012) on the surroundings (humans, hydraulic valves, other machines, devices and processes);
- noise emissions (particularly low-frequency noise) (Hubballi and Sondur, 2017).

The analysis of vibration and or acoustic diagnostic signals can provide information about phenomena and processes in machines. The machines are often equipped with hydrostatic systems whose main source of noise is the working positive displacement pump or maximum valve. In this case, a diagnostic acoustic signal is used to detect the presence of cavitation in the suction port of the positive displacement pump or in the maximum valve (Kollek et al., 2007). The main objective of this paper is to demonstrate the coincidence of pulsatile fluid flow with mechanical vibrations of hydrostatic system components. The paper focuses on the excitation of mechanical vibrations of selected elements of hydrostatic systems by pulsatile flow of working fluid. It was pointed out that the phenomenon of excitation of mechanical vibrations of hydraulic micropipes and bodies of hydraulic valves occurs. The author's own test stands, measuring instruments used and examples of experimental results are presented. Finally, the presented results are discussed.

2. Pulsating fluid flow and vibrations. A theoretical background.

The coincidence of mechanical vibrations and pressure pulsations in hydraulic systems can be approached in two ways. The first approach relates to the vibrations of the hydraulic system components that are caused by a pulsating flow (e.g. vibrations of pipes and valves), while the other approach relates to pressure pulsation arising due to the vibrations of the hydraulic system components, and in particular, of valve control components (e.g. directional valve spools or relief valve discs).

Variable fluid pressure in a hydraulic system may be caused by factors such as the pulsating efficiency of the working fluid, resulting from the kinematics of operation of displacement components in the pump. The instantaneous capacity of an external gear pump can be represented as (Stryczek, 2014, p66):

(1) $$Q = \frac{b_z \cdot \omega_l}{2} \cdot \left[r_{w1}^2 + \frac{r_{t1}}{r_{t2}} \cdot r_{w2}^2 - r_{t1} \cdot (r_{t1} + r_{t2}) - \left(1 + \frac{r_{t1}}{r_{t2}}\right) \cdot u^2 \right]$$

where,

b_z – width of the wheels, ω_l – angular velocity of the gear, r_w – radius of the apex wheel, r_t – radius of the rolling wheel, u – momentary distance of the point of contact of the teeth, moving along the line of the buttress, from the mesh pole, r_z – radius of the base wheel, φ_z – angle of rotation of the gears.

In general, the flow pulsation of a positive displacement pump is described by the flow irregularity coefficient δ (Stryczek, 2014: 71):

(2) $\delta = \frac{Q_{max} - Q_{min}}{Q_{av}}$

where,

Q_{max}, Q_{min} – maximum and minimum flow rate respectively, Q_{av} – mean flow rate.

The pulsating efficiency of the pump creates periodically variable pressure values in the hydraulic system, in the form of a harmonic function. The frequency of this pressure pulsation corresponds to the pulsation of efficiency (Kudźma et al., 2014). It also causes mechanical vibrations, arising in hydraulic pipes and bodies of hydraulic valves.

Fluid flowing in a tube can cause flexural vibrations if a certain velocity value is exceeded, resulting in a loss of stability of the tube. The value of the velocity at which this occurs is usually called the *critical velocity of flow*. The critical velocity depends on the boundary conditions adopted, i.e. the way the tube is supported. The description of the flexural vibrations of a hydraulic tube with a flowing fluid assumes treating the tube as a Bernoulli–Euler beam. Therefore, the dynamic equilibrium of an elementary tube section d_x, on which forces and corresponding moments act, must be considered. Without going into the details of the analytical considerations, the final form of the mathematical relation describing the flexural vibrations of the microtube caused by the pulsating fluid flow can be presented (Gryboś, 2005: 54) as follows.

(3) $EJ\frac{\partial^4 w}{\partial x^4} - \frac{\partial}{\partial x}\left[(S - pA)\frac{\partial w}{\partial x}\right] + b\frac{\partial w}{\partial t} + m_c\frac{D^2 w}{Dt^2} + m_r\frac{\partial^2 w}{\partial t^2} + (m_c + m_r)gcos\alpha = 0$

where,

m_r –mass of a tube section of unit length;

m_c –mass of a stream section of unit length;

$w_{(x,t)}$ –tube deflection in the section with coordinate x and time t;

S – external tensile force;

p – pressure inside the tube;

A – surface area of the tube;

b – coefficient of external resistance of the medium in which the tube is located.

The value *(S-pA)* depends on the pressure p and on the type of ties limiting the conductor's freedom of axial displacement.

The deflection function of the tube which is to satisfy the above equation should consist of two terms: one corresponding to the static deflection w_s, which is the result of the weight of the tube and the fluid, and the dynamic deflection w_d. It can be formulated as follows:

(4) $w(x,t) = w_s(x,t) + w_d(x,t)$

In order to obtain the final solution of the above equations, treating the pipe as a Bernoulli–Euler beam, it is necessary to formulate the boundary conditions – the way the tube is supported. Obtaining a more accurate description requires taking into account wave phenomena in the tube and flow during non-stationary processes, and the tube must be treated as a Timoshenko beam. This leads to partial differential equations of the hyperbolic type, which will not be presented further in this paper, as the focus here is on the experimental research aspect.

3. Methods and experimental setup.

Experiments have been carried out to verify the excitation of vibrations by pulsating fluid flow. The experiments used a steel hydraulic microtube with an internal diameter of 4 mm, an external diameter of 6 mm, a length of 194 cm, and a circular cross-section. The tube was supported at both ends. A diagram of the hydraulic system with

measurement points marked is illustrated in Fig. 3. Measured parameters included pressure pulsation in the tested microhydraulic system (points p_1 and p_2) and mechanical vibrations of the tube in two mutually perpendicular directions X and Z (points a_1 and a_2). The experimental system featured an external meshing gear micropump with a unit efficiency of 0.8 cm^3/rotation, powered by an electric motor with a speed of 1,380 rpm. The value of static pressure in the system was measured using a throttle microvalve.
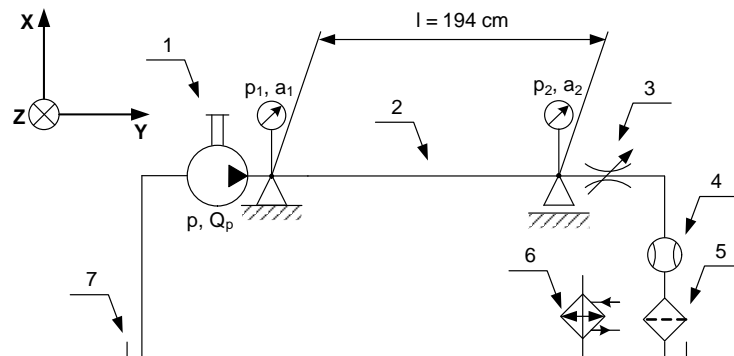
Figure 3. Diagram of the experimental system: p_1 – piezoelectric pump pressure pulsation sensor, a_1 – 3-axis accelerometer, p_2 – piezoelectric throttle valve pressure pulsation sensor, a_2 – 1-axis accelerometer, a_3 – 3-axis accelerometer, 1 – micropump, $q_p = 0.8$ cm^3/rotation, 2 – hydraulic microtube, internal diameter 4 mm, 3 – adjustable hydraulic microvalve, 4 – flow meter, 5 – hydraulic oil filter, 6 – hydraulic oil cooler, 7 – hydraulic oil tank.

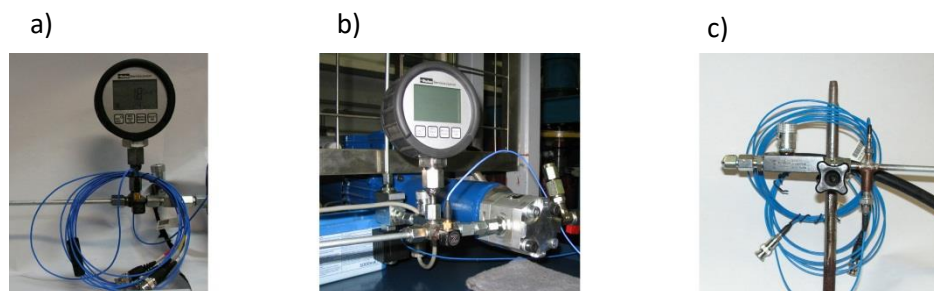Figure 4 shows the detailed locations of measurement sensors in the system.

Figure 4. Point of measurement for the acceleration of mechanical vibrations of the hydraulic microtube and pressure pulsation in the microhydraulic system: a), b) at the micropump, c) at the throttle microvalve

The acceleration of mechanical vibrations was measured using Piezotronics accelerometers: point a_1 – 3-axis ICP accelerometer, point a_2 – 3-axis miniature ICP accelerometer; pressure pulsation was measured using ICP 105C23 miniature piezoelectric pressure sensors.

The excitation of mechanical vibrations caused by pulsating flow was also verified using a specially designed test stand, with a latest-generation DFPlus single-stage proportional directional valve as its major component. The test stand enabled the generation of pressure pulsation with frequencies up to 350 Hz (the limit frequency of the DFPlus directional valve). A diagram of the test stand hydraulic system is shown in Figure 5.
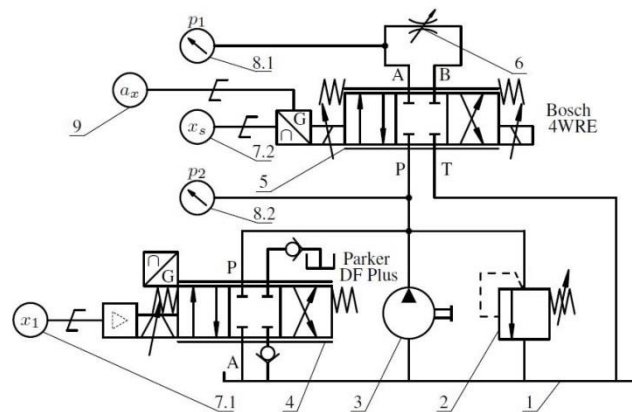
Figure 5. Diagram of the test stand hydraulic system used to test the impact of pressure pulsation on a hydraulic valve: 1 – oil tank, 2 – safety valve, 3 – displacement pump, 4 – DFPlus proportional directional valve (pulsation exciter), 5 – tested valve: a single-stage proportional directional valve, symbol 4WRE, 6 – adjustable throttle valve, 7.1, 7.2 – directional valve spool position sensors, 8.1, 8.2 – pressure sensors, 9 – accelerometer.

The following parameters were measured and recorded: DFPlus directional valve control signal waveform, DFPlus directional valve spool displacement, pressure changes upstream and downstream of the tested 4WRE directional valve, acceleration of vibrations of the body of the tested 4WRE directional valve, displacement of the spool of the tested 4WRE directional valve. A diagram of the control, measurement and data acquisition system is shown in Figure 6.
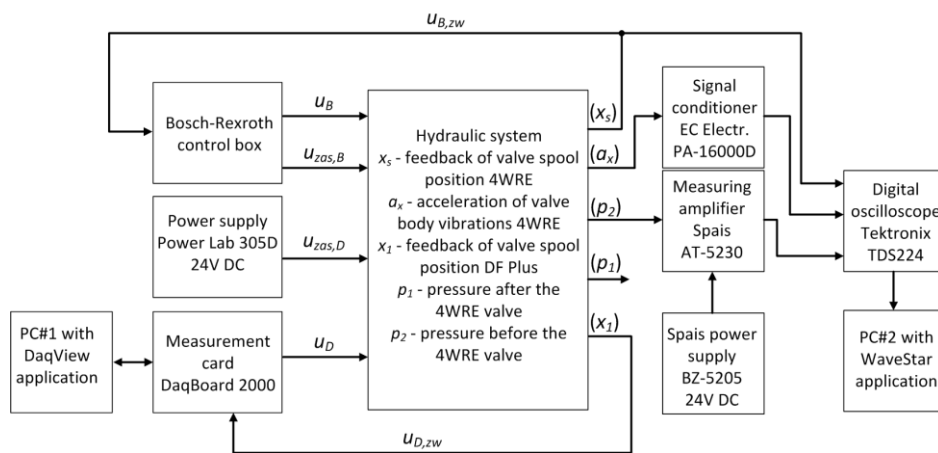

Figure 6. Diagram of the control, measurement and data acquisition system in the pressure pulsation generation test stand.

The software used with the test stand enabled the generation of changes in the control signal of the DFPlus valve over a required time. The procedure for controlling the valve using the software referred to above is described in detail in references (Herok et al., 2014) and (Kudźma, 2012). The harmonic signal used had a set frequency f and amplitude s_0, and a fixed zero offset value s_{01} of the form: $s = s_{01} + s_0 \cdot sin(2 \cdot \pi \cdot f \cdot t)$. The test results that indicate the excitation of mechanical vibrations of the body of the hydraulic directional valve as a result of pressure pulsation are shown in Figures 8 and 9, with the following parameters of the control signal: $s_{01} = 5$ V, $s_0 = 4$ V. The modular design of the test stand enables replacement of the 4WRE directional valve (item 5, Fig. 5) with a different hydraulic valve.

4. Results and discussion

The test stands shown in Figures 3 and 5 allowed results to be obtained, examples of which are shown in Figures 7, 8 and 9. The results of the test carried out on the test bench of Figure 3, in the form of an amplitude and frequency spectrum of pressure pulsation and vibrations of the microtube, horizontally and perpendicular to the axis of the tube, are shown in Figure 7.
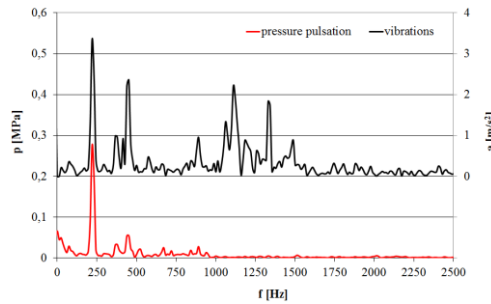

Figure 7. Amplitude and frequency spectrum of pressure pulsation and mechanical vibrations of the hydraulic tube; $p_{śr}$ = 5 MPa, Q = 1.83×10^{-5} m^3/s (1.1 dm^3/min).

Harmonic constituents of mechanical vibrations corresponding to pressure pulsation can be seen in the spectrum shown in Figure 7 – the pressure pulsation of the fluid flow generated mechanical vibrations in the hydraulic tube. The frequency value of the thus generated first constituent of the excitation spectrum (pressure pulsation) may be calculated using the following formula:

(5) $$f_1 = \frac{n \cdot z}{60} = \frac{1380 \cdot 10}{60} = 230 \text{ Hz}$$

where,

n – angular speed of the pump shaft [rpm],

z – number of teeth [–].

Figures 8 and 9 show the horizontal vibrations of the 4WRE directional valve body along the axis of movement of the directional valve spool. These results were obtained on a test stand, the hydraulic diagram of which is shown in Figure 5.
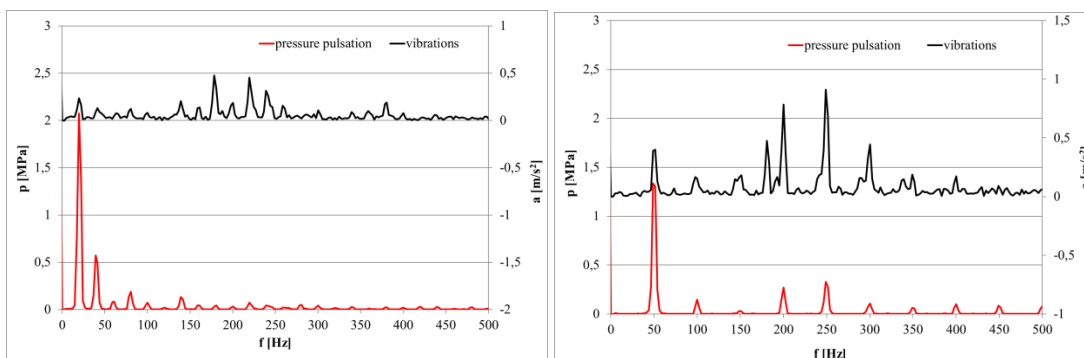

Figure 8. Amplitude and frequency spectra of pressure pulsation and excited mechanical vibrations of the body of the 4WRE directional valve: a) frequency of DFPlus directional valve control signal f = 20 Hz, b) frequency of DFPlus directional valve control signal f = 50 Hz, c) frequency of DFPlus directional valve control signal DFPlus f = 70 Hz, d) frequency of DFPlus directional valve control signal DFPlus f = 90 Hz; $p_{śr}$ = 2.5 MPa, Q = 1.083×10^{-4} m^3/s (6.5 dm^3/min).
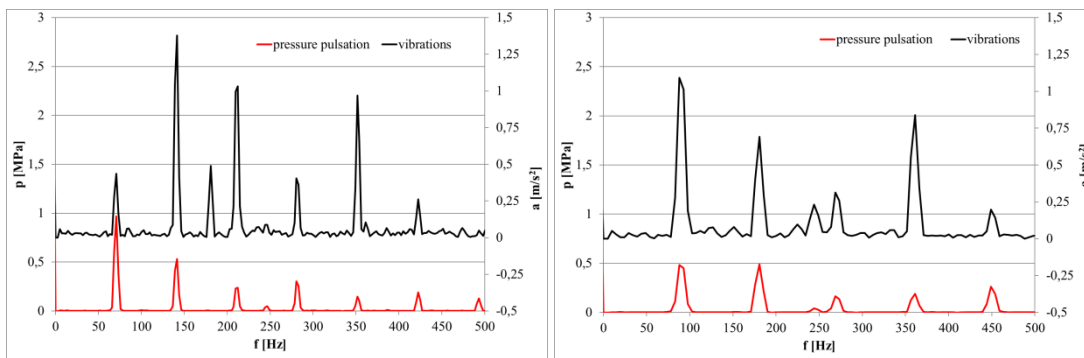
Figure 9. Amplitude and frequency spectra of pressure pulsation and excited mechanical vibrations of the body of the 4WRE directional valve: c) frequency of DFPlus directional valve control signal DFPlus f = 70 Hz, d) frequency of DFPlus directional valve control signal DFPlus f = 90 Hz; $p_{śr}$ = 2.5 MPa, Q = 1.083×10^{-4} m^3/s (6.5 dm^3/min).

The results shown in Figures 8 and 9 indicate the spectrum of pressure pulsation upstream of the tested 4WRE directional valve at point 8.2 in Figure 5, and the acceleration of vibrations of the body of the directional valve at point 9 along the spool axis. The results confirm that a correlation exists between pressure pulsation in a hydraulic system and the vibrations of valves forming part of the system.

3 Conclusions

Both our own research and literature reports indicate that machines equipped with hydraulic systems generate mechanical vibrations with broad frequency spectra and significant amplitudes. These vibrations affect the human body and, in particular, the components of hydraulic systems. Furthermore, mechanical vibrations may disrupt the normal operation of these components. An analysis of the results of tests involving exciting flexural vibrations of microtubes and vibrations of the body of the 4WRE directional valve indicates coincidence between the constituents of the pressure pulsation spectrum (excitation) and the constituents of acceleration of mechanical vibrations (system response), which results in the generation of mechanical vibrations at frequencies corresponding to the frequency of pressure pulsation in the tested system. For the results of directional valve body vibration tests presented in the study, the highest amplitudes of the first vibration constituent were observed for a pressure pulsation frequency of 90 Hz. Significantly higher constituents are evident in the spectra of vibrations of the valve body (180 Hz, 360 Hz). With an increase in the frequency of pressure pulsation generated by the DFPlus directional valve, its amplitude falls, while the amplitudes of the 4WRE directional valve body vibrations increase. This may indicate that the excitation frequency approaches one of the frequencies of normal mode vibrations of the 4WRE directional valve, treated as a two-body system. In Figures 8 and 9 we can also observe components of the vibration spectrum with frequencies below 100 Hz, which are particularly dangerous vibration frequencies for humans (as shown in Figure 1). Low-frequency vibrations are poorly attenuated by matter and propagate over long distances. It is therefore important to eliminate them at their source, i.e. using active methods.

Although the paper presents considerations for machines and equipment equipped with hydraulic systems, the scope of conclusions is not limited to this class of machines. Adverse effects of vibrations and noise on the environment (including human beings) may have different sources but identical effects. For example, in road passenger transport vehicles, vibrations and noise transmitted into the vehicle interior are caused by, among other things, unevenness of the ground, unbalance of rotating parts, inaccuracy of assembly, clearances, aerodynamic resistance of the medium, etc. In the analysis of the effects of such excitations, frequency ranges, signal amplitudes, place of application of the excitation, transmission paths and exposure times are important. Consequently, modern machinery or equipment must not only meet the new requirements with regard to efficiency and energy saving, but also with regard to the emission of harmful signals into the environment. Such an approach will meet the requirements for sustainable development of new machine designs.

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