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Reduction of infrasounds in machines with hydrostatic drive

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Some hazards posed by the operation of hydraulic systems, connected with low-frequency vibrations and noise are presented. Special attention is focused on infrasounds. The sources of low-frequency vibrations and noise and ways of reducing them are indicated. An original solution ensuring the effective reduction of vibrations and noise in a wide frequency range, i.e., a wide-band damper of pressure fluctuations, also performing the function of an acoustic filter, is proposed. The effectiveness of the damper was confirmed by the results of laboratory tests and tests carried out on engineering machines working in real conditions.

Key words: *infrasounds, vibrations, pressure fluctuation, damper*

1. Introduction

This paper deals with the reduction of noise emitted by heavy engineering machines, generally with long-frequency noise and specifically with infrasonic noise [1]. Infrasounds are commonly defined as sounds or noise whose frequency spectrum is in a range of 1 to 20 Hz. Currently, the concept of low-frequency noise covering the frequency range from about 10 Hz to about 250 Hz is increasingly widely used. The lower frequency limit (10 Hz) has been adopted due to considerable difficulties in measuring this noise correctly and interpreting the results for frequencies below 8–10 Hz. Low-frequency noise, including infrasonic noise, is received by the human being through both the auditory canal and the vibration receptors distributed over the whole body. Its harmful impact manifests itself in subjectively perceived excessive fatigue, discomfort, drowsiness, balance and psychomotor disturbances and disturbances in the physiological functions. Such states result in changes in the central nervous system, characteristic of diminished wakefulness, which is especially dangerous for the operators of machines and equipment [1].

Figure 1 shows the free vibrations of some of the human being's organs [2]. Numerous experiments have demonstrated that the nervous system and the cardiovascular system are most sensitive to the vibration of the whole human body.

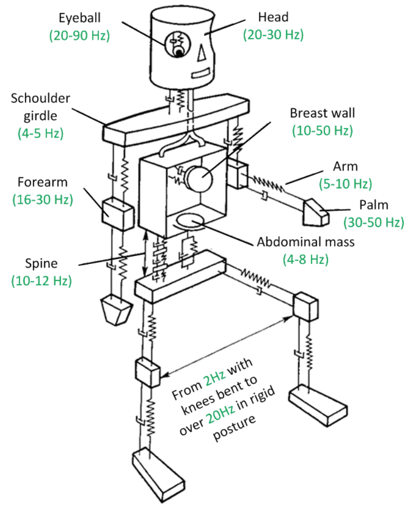

Fig. 1. Schematic of human elastic system [2]

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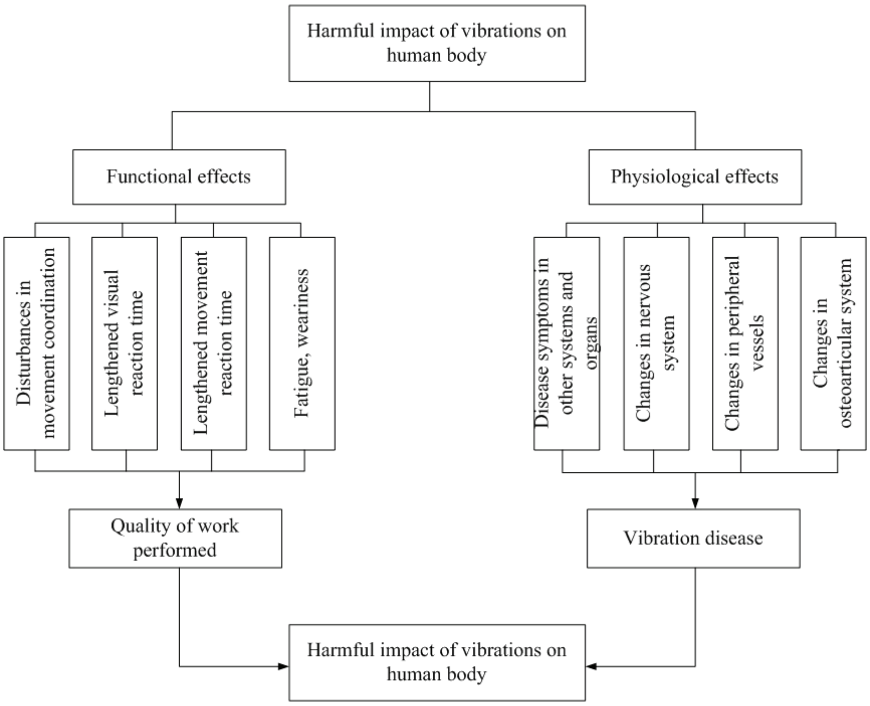

Fig. 2. Effects of vibrations on human body [2]

Table 1. Breakdown of disease symptoms observed for infrasound impact on human body

f [Hz]	L_m [dB]	t_{exp} [min]	N	Disease symptoms	Source
2–15	105			Visual reaction time lengthening in 50% of tested persons, balance disturbances in 10% of tested persons	[3]
1–2	150			Shift in auditory threshold, shifting eardrums sensation	[3]
2–15	110–120		7	Reaction time lengthening by 4%	[5]
3–15	115	30		Alcohol albuse symptoms	[5]
10	135	15	6	Sensation of internal organs vibration, feeling tympanic membrane vibrate, middle ear ache, rapid pulse, increased arterial blood pressure, respiration rate increased by 4 or more breaths per minute, auditory threshold shifted by 15–20 dB during test and by 8–10 dB after test	[6]
7	90	35	30	Drop in blood pressure, lowered pulse rate, cardiac murmurs occurred in half of tested persons	[4]
2–22	119–144	3	30	Auditory threshold shifted by 10 dB in 30% of tested persons	[4]
1–100	154	0.4–2		Sensation of swaying walls, headache and dizziness, breathlessness, ear buzzing, lockjaw	[4]

Legend: f – frequency, L_m – acoustic pressure level, t_{exp} – exposure time, N – number of tested persons, compiled from Refs. [3]–[6].

Through the mechanoreceptors in the skin vibrations transmit specific information to the central nervous system, causing reflex reactions of the whole body. As a result of the sustained action of mechanical vibrations irreversible changes take place in the different organs and systems. The changes can be divided into [2]:

- **acute changes**, occurring during and immediately after the exposure, consisting in peculiar changes in the behaviour of the whole body treated as a mechanical system;
- **chronic changes**, occurring as a result of the sustained action of high intensity vibrations.

Often the load capacity of the body is exceeded, resulting in health disorders referred to as vibration disease (Fig. 2) [2].

Infrasounds have an adverse effect on the performance of simple actions, resulting in the lengthening of the reaction time, the lowering of the perception threshold and the disturbances of balance. Noise of 2–15 Hz whose level amounts to 105 dB causes visual reaction time lengthening in 50% of the persons being tested [3]. In addition, infrasounds induce a false sense of well-being or euphoria. According to the available test results [4], 50% of the persons subjected for 35 minutes to the action of a tone with a frequency of 7 Hz and an acoustic pressure of 90 dB suffered from cardiac murmurs which would disappear after 10–25 minutes after the source of infrasounds was switched off. Also in [4] it is reported that the persons subjected twenty five times, each time for 2 minutes, to the action of a tone with a frequency of 7 Hz and an acoustic pressure of 90 dB complained of dizziness, breathlessness, the sensation of swaying walls, ringing in the ears, lockjaw and general fatigue. It has also been found that persons suffering from heart diseases, cardiovascular system diseases and otosclerosis are more susceptible to the action of infrasounds than healthy persons [4]. However, it should be mentioned that for humanitarian reasons there are constraints on tests aimed at determining noise levels causing disorders since lower intensity sounds cause disorders at the cellular level, changes in the blood, etc., which may contribute to diseases. Many of the diseases begin in the nervous system and in the vascular-coronary system and their early symptoms go unnoticed [4].

It should be mentioned that the infrasonic noise problem was noticed only in recent years and methods of measuring the quantities characterizing infrasonic noise were defined in an ultrasonic noise testing procedure published in the quarterly "Podstawy i Metody Oceny Środowiska Pracy" (The Foundations and Methods of Work Environment Assessment) [7] as late as 2001 and in the standards: PN-ISO 7196:2002 [8] and PN-ISO 9612:2004 [9]. The highest permissible infrasonic noise levels (HPNL values) in the engineering machine operator workplace were specified in the Labour and Social Policy Minister's ordinance at the end of 2002 [10]. The levels are shown in Table 2.

Table 2. Highest permissible infrasonic noise levels in operator workplace

Evaluated quantity	Permissible value
Equivalent acoustic pressure level adjusted by frequency characteristic G, for 8-hour, 24-hour or average weekly work time specified in emplyment code	$L(G)_{,eq\ 8h}$ 102 dB
Unadjusted peak acoustic pressure level	145 dB

Preliminary studies carried out by The Research Institute of the Central Work Protection Institute indicate that the highest permissible infrasonic noise levels ($L(G)_{eq8h}$ = 102 dB) are exceeded in various vehicles of Polish and foreign make [1]. The permissible infrasonic noise levels have been found to be exceeded by up to 7 HPNL in the driver's cabs of crawler dozers and wheel loaders working in, e.g., the Turów mines.

The same requirements and principles as the ones for audible range (20 Hz–20 kHz) of noise, e.g., caused by impact pressure variations or by factors connected with working liquid flow (cavitation – cavitation noise is high-frequency noise) are applied in the prevention of the harmful effects of infrasonic noise [11], [12]. Because of the considerable infrasound wavelengths (17–340 m) protection against infrasounds poses difficulty since conventional walls, partitions, screens and sound absorbers are rather ineffective in this case. In some cases, infrasonic waves are amplified due to the resonance of rooms, the structural elements of cabs, and the whole constructions. The best protection against the harmful impact of infrasounds is their elimination at the source, i.e., by eliminating the causes of infrasonic noise generation.

In order to eliminate sources of low-frequency noise, especially infrasonic noise, in the hydraulic system it is proposed to use an active pressure fluctuation damper.

2. Materials and methods

2.1. Disturbances in operation of hydraulic systems – their sources and effects

In real operating conditions hydraulic valves (also valve blocks) are subjected to mechanical vibrations [13]–[15]. There are different sources of the vibrations, such as: the unbalance of the rotating parts of machines, variable loads, travel on an uneven surface, etc., producing many adverse effects, such as the vibration of the control elements in hydraulic valves. The elements are responsible for setting gaps in hydraulic valves. When the valve's control element (e.g., the slide) begins vibrating its throttling gap area changes, which results in output and pressure fluctuations in the hydraulic system. The spectral compo-

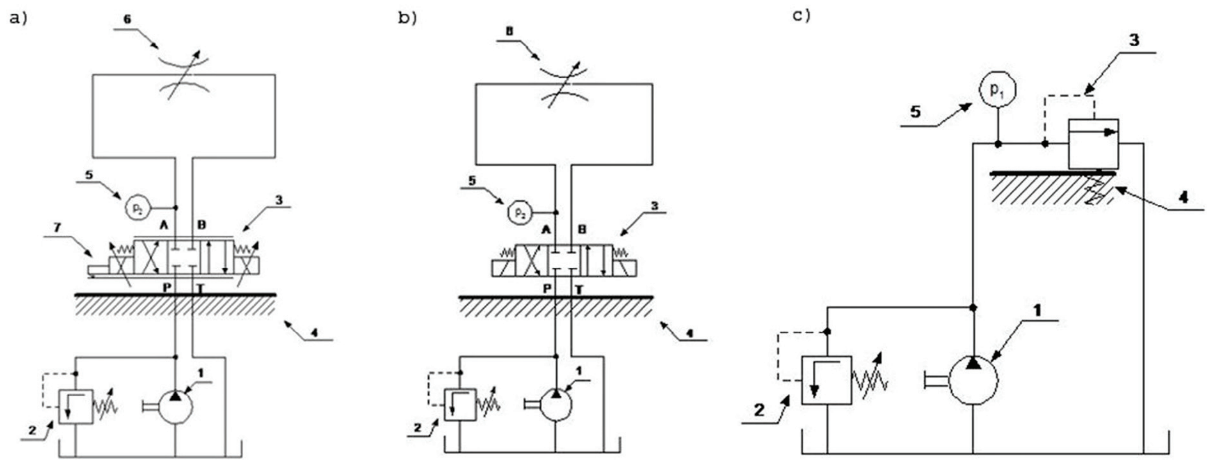

Fig. 3. Diagram of hydraulic system used for valve testing: 1 – displacement pump, 2 – safety valve, 4 – vibrating simulator table, 5 – pressure measurement, 6 – adjustable throttle valve; (a) 3 – proportional distributor, 7 – slide displacement sensor, (b) 3 – conventionally electrically controlled valve, (c) 3 – overflow valve

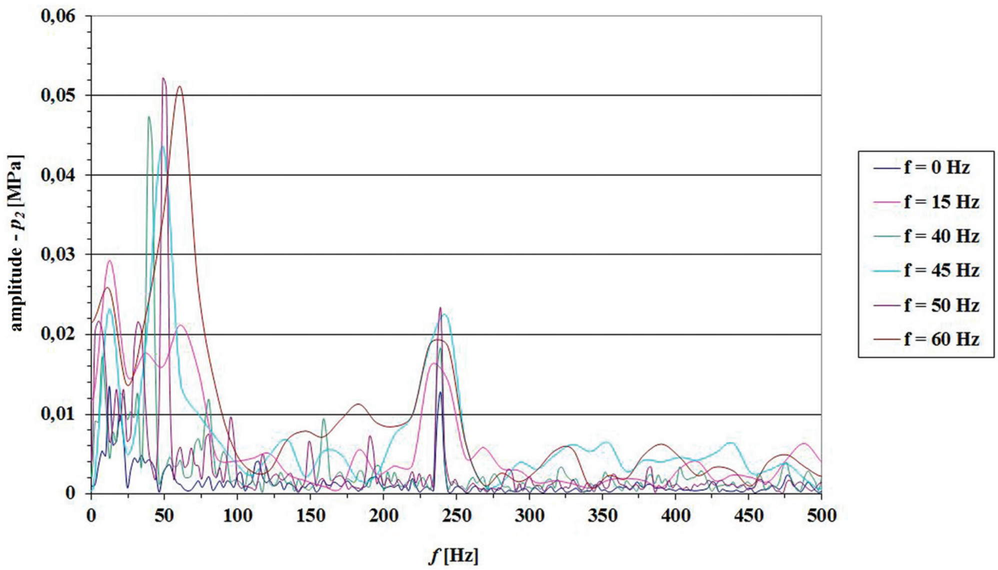

Fig. 4. Amplitude-frequency spectrum of pressure fluctuation in hydraulic system with proportional distributor excited with frequency $f = 0$, 15, 40, 45, 50 and 60 Hz. Average pressure – 2 MPa, average flow rate – 6.5 dm^3/min

nents of this vibration correspond to the spectral components of the excited element [16]. The authors' own research and literature reports [17]–[19] indicate that the resonance frequencies of valve control elements (conical heads, slides) are below 100 Hz [16, 20]. This frequency range is particularly dangerous for people since the resonance frequencies of major internal organs are below 100 Hz [2], [4]. The fluctuations may produce infrasounds. Pressure fluctuations may result in the unstable operation of hydraulic receivers, contributing to, among other things, their imprecise work.

Tests in which a simulator of the linear hydrostatic drive was used as a source of external low-frequency mechanical vibrations were carried out. The simulator is described in more detail in [13], [21].

A selected hydraulic valve was mounted in a specially designed grip of the simulator. The valve was then subjected to mechanical vibrations generated by the simulator and the pressure fluctuations in this hydraulic system were recorded by a piezoelectric ICP transducer. The direction of the external mechanical vibrations coincided with the direction of motion of the valve's control element tested. A diagram of the hydraulic system in which the selected valve was tested is shown in Fig. 3a–c.

An overall diagram showing the amplitude-frequency spectrum of the pressure fluctuation in the hydraulic system with the vibrating proportional distributor, consistent with Fig. 3a, is shown in Fig. 4.

In Fig. 4 one can see spectral components the frequencies of which correspond to the frequencies of the external mechanical vibrations below 100 Hz. Moreover, there is a component originating from the fluctuations in the delivery of the pump feeding the distributor system. The rotational speed of the pump drive shaft and the number of the pump displacement parts indicate that the resulting spectral component will appear at a frequency of about 242 Hz.

Then a conventional distributor was tested in the hydraulic system shown in Fig. 3b. The direction of the external mechanical vibrations was consistent with the direction of motion of the slide (Fig. 5).

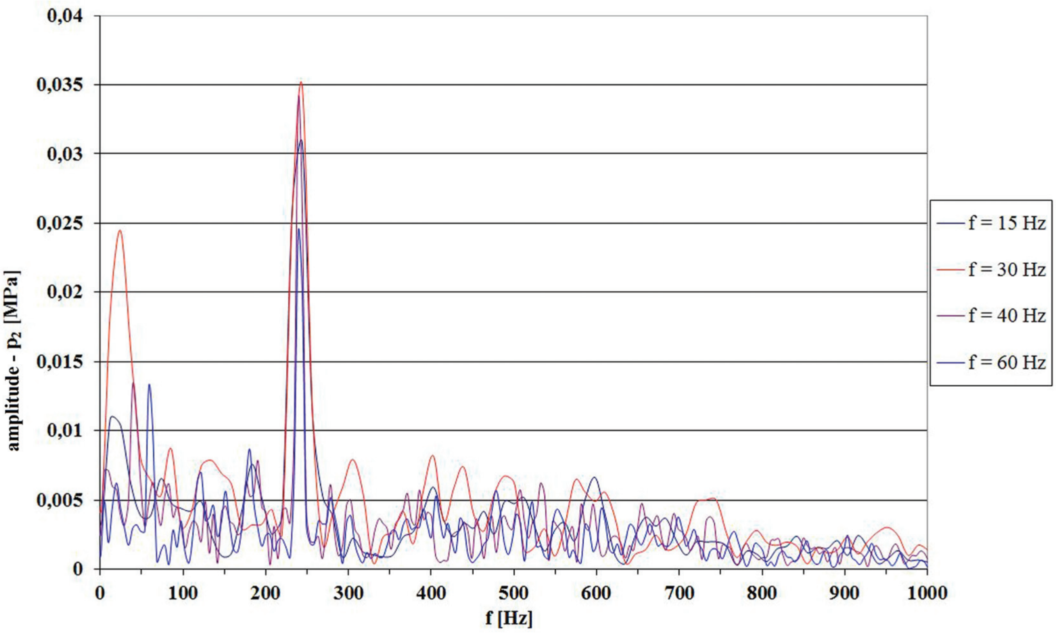

Fig. 5. Amplitude-frequency spectrum of pressure fluctuation in hydraulic system with conventionally electrically controlled single-stage slide distributor excited with frequency *f* = 15, 30, 40 and 60 Hz. Average pressure – 2 MPa, average flow rate – 6.5 dm^3/min

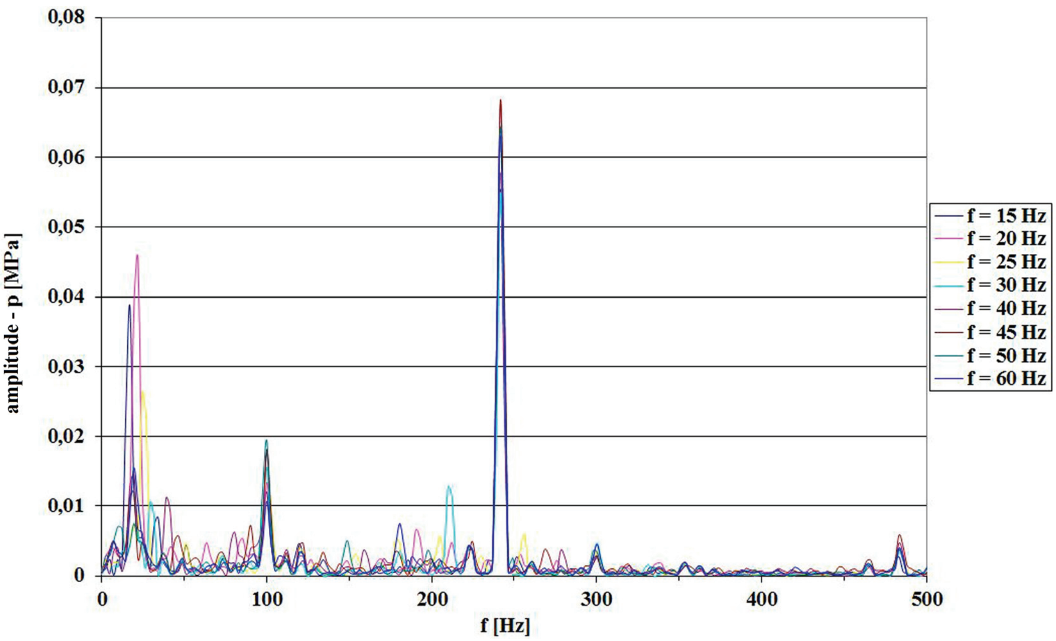

Fig. 6. Amplitude-frequency spectrum of pressure fluctuation in hydraulic system with single-stage overflow valve excited with frequency *f* = 15, 20, 25, 30, 40, 45, 50 and 60 Hz. Average pressure – 2 MPa, average flow rate – 6.5 dm^3/min

Figure 6 presents the results of experiments in which a single-stage valve was placed on the vibrating table of the hydraulic simulator in the hydraulic system shown in Fig. 3c.

An analysis of the results presented in Figs. 4, 5 and 6 shows that there is a correlation between some components of the pressure fluctuation spectrum and the external mechanical vibrations acting on the selected hydraulic valves. This manifests itself in the presence of harmonic components corresponding to the frequency of the external mechanical vibrations, in the spectrum. The fluctuations, in turn, can excite low-frequency vibrations of the hydraulic conduits, propagating for considerable distances.

In order to minimize the transfer of the external mechanical vibrations from the foundation to the hydraulic valves one can use, e.g., vibration insulators [20]. Another effective way of reducing pressure fluctuations at both low frequencies (<100 Hz) and higher frequencies (150–350 H) is to additionally use a compact pressure fluctuation damper [22].

2.2. Identification of noise sources in hydraulic system

Noise can be generated in a hydraulic system in two ways:

- directly – the source of noise causes changes in ambient air pressure. An example of such a noise source is the rotor of the fan in the electric motor driving a pump;
- indirectly – time-variable forces excite vibrations in the hydraulic system components. The vibration of the surface of the components results in noise emission [23].

Indirectly generated noise dominates in the hydraulic system. Variable forces acting on the parts of the hydraulic system are produced as a result of pressure fluctuations and the mechanical connection of the hydraulic system components through conduits and the common mounting. The vibration of a single component, e.g., a valve, is produced by the impact of the liquid and induces vibrations in the other elements connected with this component. Pressure fluctuation is on one hand the result of the periodically variable rate of flow of the working medium (due to cyclical operation of pump displacement components) and on the other hand, it is the result of external excitations in the form of mechanical vibrations acting on the hydraulic system parts fixed to all kinds of load-bearing structures, e.g., to the loader's frame.

Another source of pressure fluctuations in the hydraulic system is the variable character of hydraulic engine loads and engine starting and braking (nonstationary states). Pressure fluctuations in drive systems caused by the variable loading of the work performing systems occur in the low frequency band, i.e., from 0.5 to 10 Hz (the infrasonic range), whereas the fluctuations originating from the pump are in a range of 50–1500 Hz (and above).

2.3. Effective method of reducing low-frequency noise

It is necessary to reduce low-frequency pressure fluctuations for ergonomic reasons and also because the reduction of the low-frequency excitations leads to the minimization of the resonant vibration of the hydraulic system elements, such as conduits, valves, controllers and distributors, whose free vibration frequencies are in this frequency range. In order to minimize pressure fluctuation amplitudes in the low-frequency range an active damper described in patent specification [24] was adopted. Figure 2 shows this damper in its version adapted to laboratory testing. The adaptation consisted in the introduction of an additional piston rod making it possible to change the total mass of the vibrating system in a simple way.

The low-frequency fluctuation damper shown in Fig. 7 works as follows. After cylinder 1 is filled with hydraulic oil and the hydropneumatic accumulator mounted in back cover 4 is pre-charged with nitrogen up to pressure p_0, the damper cylinder is connected in parallel through a threaded hole in back cover 5 to the pump pressure conduit. The damper operation comes down to the taking over of delivery fluctuations. The latter are the cause of the excitation of pressure fluctuations in the system. The excitations due to fluctuations in delivery and consequently in pressure are taken over by the moving piston & piston rod–hydropneumatic spring system. An analysis of the mathematical model shows that the damper parameters are matched to ensure the equality of resonant frequency f_r of the piston–hydropneumatic spring system and frequency f_w of the excitation of pressure fluctuations in the system. The damper design enables the proper choice of its resonant frequency by changing reduced M of the vibrating system, being the sum of piston mass M_t, mass M_c of the liquid in the accumulator, the reduced-to-piston-surface mass of the liquid in the connector and a corrective mass mounted on a specially led out piston rod of the damper, in accordance with the relation

$$f_r = \frac{1}{2\pi}\sqrt{\frac{nF_0^2 p_k^2}{p_0 V_0 M}} \tag{1}$$

where: F_0 – the area of the piston active surface, p_k – the system's average operating pressure, p_0 – the initial pressure of the gas in the accumulator, V_0 – the initial volume of the accumulator, n – a polytropic exponent.

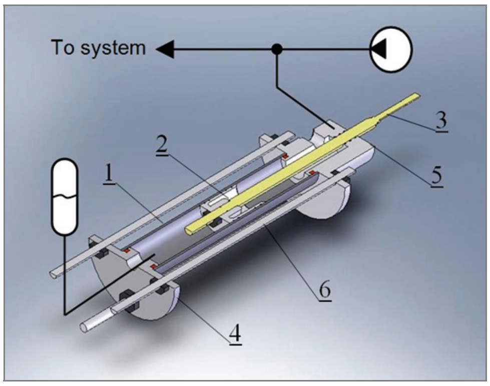

Fig. 7. Low-frequency active damper of pressure fluctuation, in its laboratory version: 1 – cylinder, 2 – piston, 3 – piston rod, 4 – back cover with hydropneumatic accumulator, 5 – front cover with connector to system, 6 – pin

Then the damper effectiveness was experimentally verified by determining the amplitude-frequency characteristic. The pressure fluctuation amplitude and the piston travel were respectively the input quantity and the output quantity. In the course of the tests the resonant frequency corresponding to the maximum transmittance was determined whereby an additional confirmation of the effectiveness of the active damper at the resonant frequency was obtained (Fig. 8).

3. Results

3.1. Experimental verification of damper effectiveness

Hydraulic studies of pressure fluctuation, involving the low-frequency damper, were carried out using test rig designed by the authors. The amplitude-frequency characteristics of the tested damper were determined. A pressure pulse exciter was used to generate harmonic pressure fluctuations in a frequency range of 7–60 Hz. The design and principle of operation of the exciter is presented in the authors' paper [25]. The test rig consisted of two main systems: a system generating harmonic pressure fluctuations and a feed system. The working medium, fed at a constant rate of delivery from the feed system, was subjected to harmonic pressure fluctuation excitations.

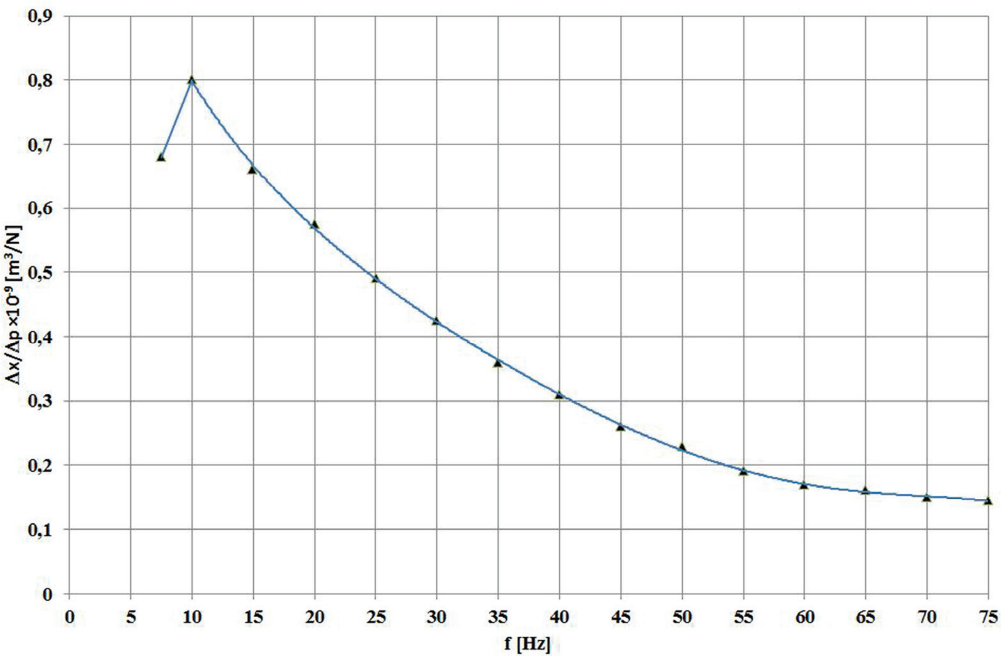

Fig. 8. Amplitude-frequency characteristic of active damper with piston rod and HYDAC, accumulator, forcing pressure 12 MPa

According to the test rig diagram (Fig. 9), the low-frequency active damper (8) was installed shunt-wise relative to the main system. The damper was supplied from the system via a connector situated under the piston, whereas above the piston the damper was supplied from the accumulator (p_0 = 2.2 MPa,

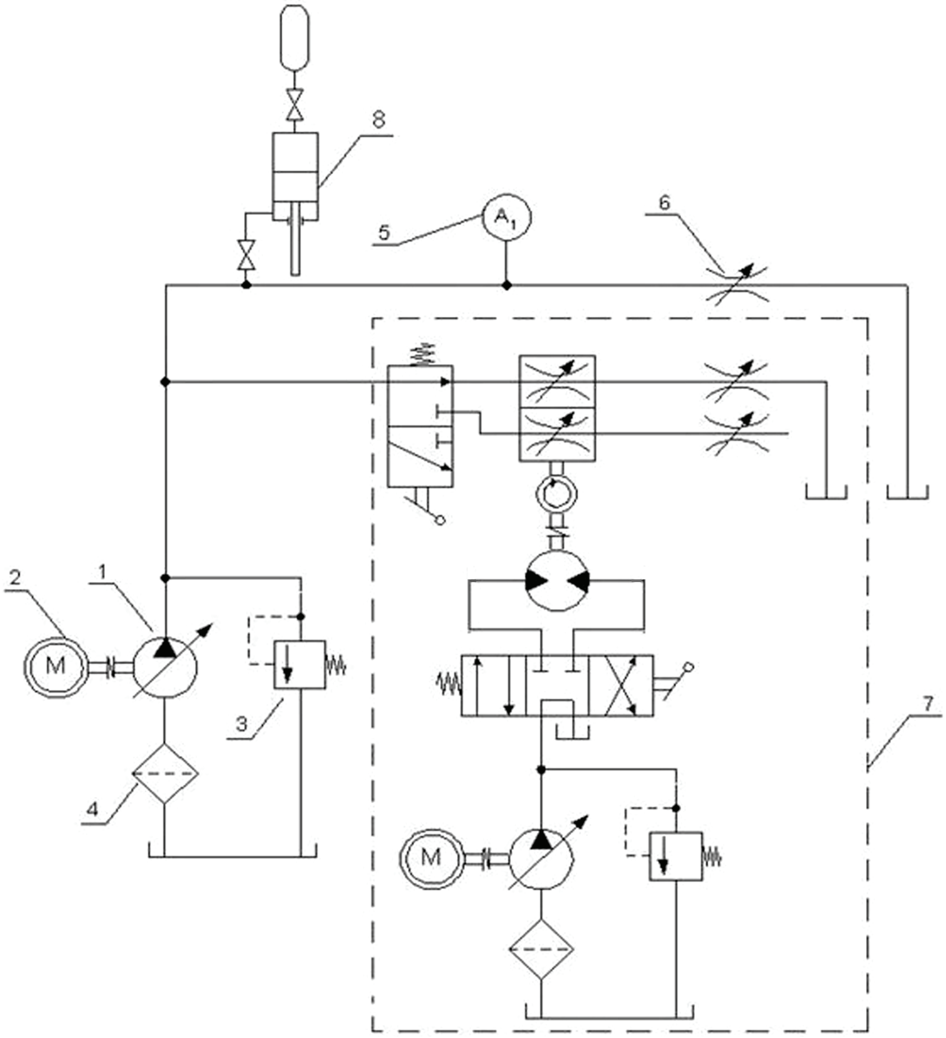

Fig. 9. Hydraulic diagram for determining frequency characteristics of low-frequency damper: 1 – variable delivery pump, 2 – electric motor, 3 – pressure relief valve, 4 – suction filter, 5 –pressure sensor, 6 – throttle valve, 7 – pressure pulse exciter, 8 – low-frequency pressure fluctuation damper

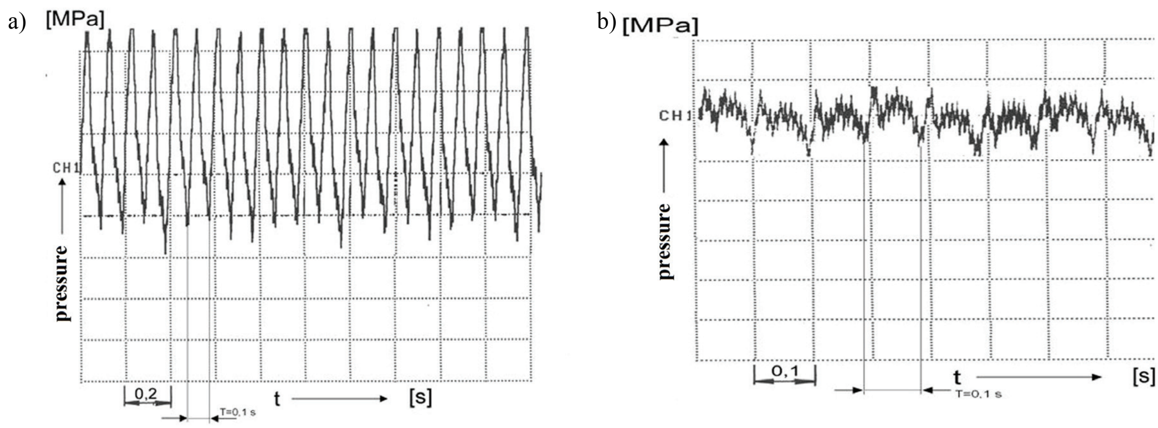

Fig. 10. Pressure fluctuation in hydraulic system. Frequency of pressure fluctuation excitation by pulse exciter f_w = 10 Hz. Forcing pressure p_t = 16 MPa, (a) – system without damper, (b) – system with damper

V_0 = 0.7 dm^3). Examples of the results of the pressure fluctuation measurements are shown in Figs. 10–12.

The above experimental results (Figs. 10a, b, 11 and 12) demonstrate the effectiveness of the low-frequency pressure fluctuation active damper in the infrasonic range.

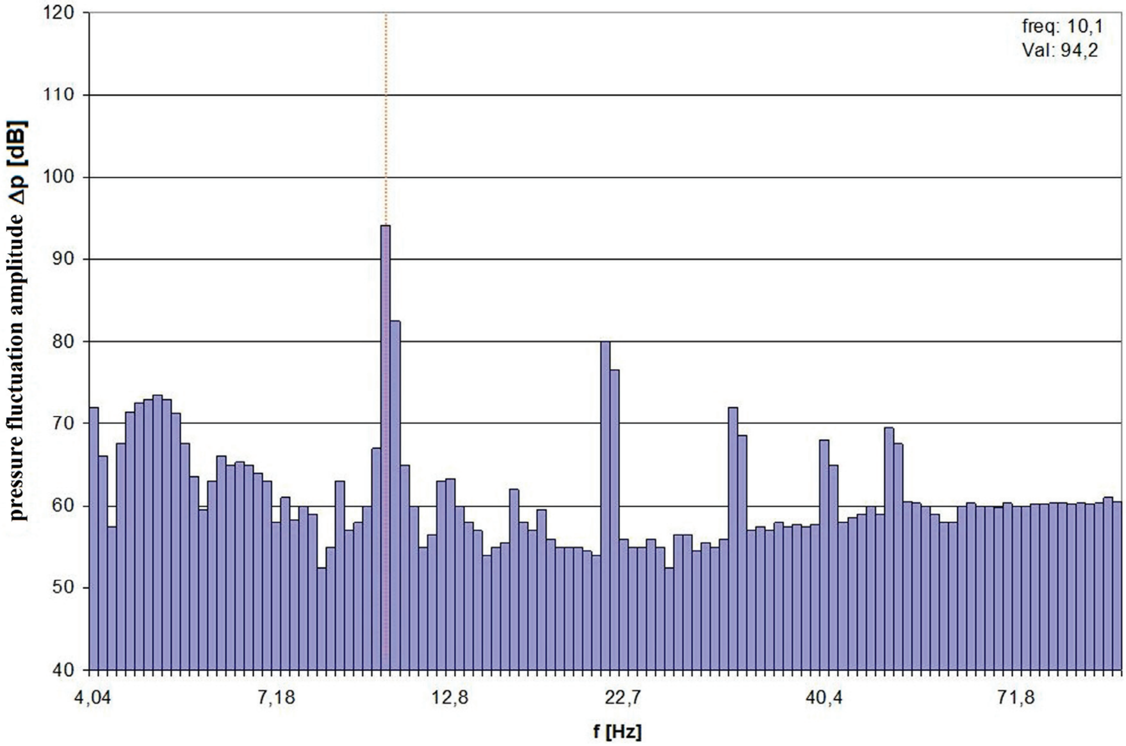

Fig. 11. Narrow-band analysis of pressure fluctuations in hydraulic system without active damper. Frequency of pressure fluctuation excitation by pulse exciter f_w = 10 Hz. Forcing pressure p_t = 16 MPa

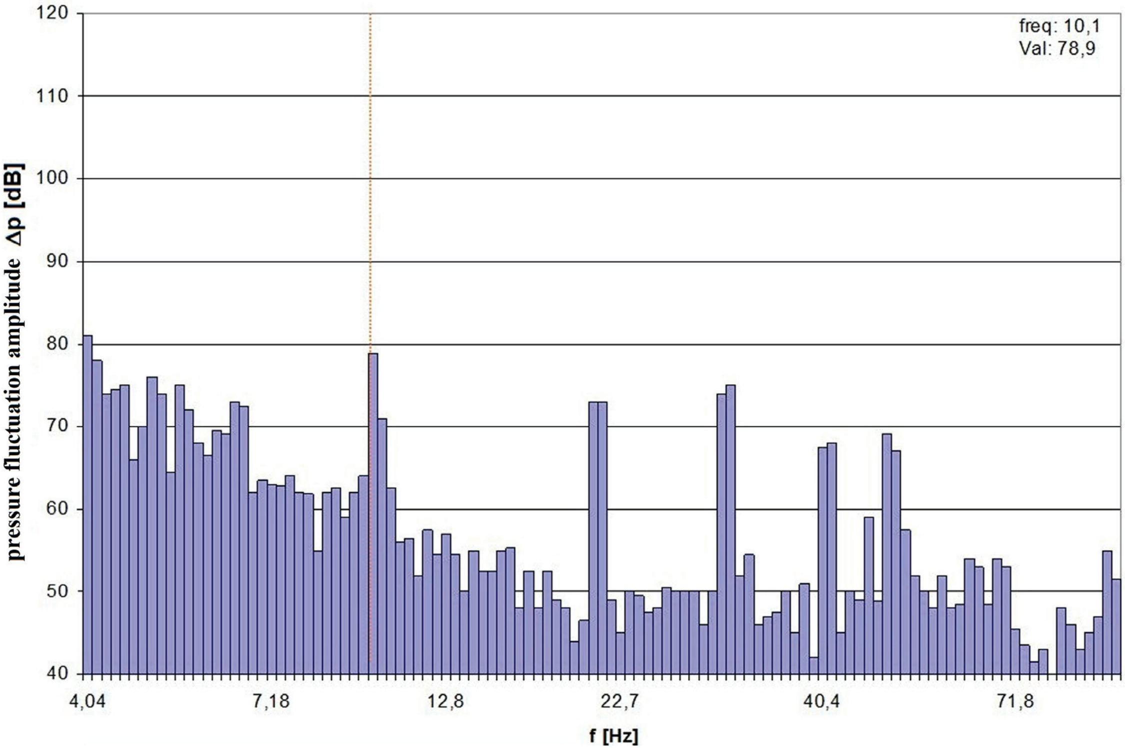

Fig. 12. Narrow-band analysis of pressure fluctuations in hydraulic system with active damper. Frequency of pressure fluctuation excitation by pulse exciter f_w = 10 Hz. Forcing pressure p_t = 16 MPa

4. Discussion

During the operation of a hydrostatic drive system a wide spectrum of excitations is generated [22], [26]. The latter, in turn, generate pressure fluctuations and consequently, audible and infrasonic noise. Protection against infrasounds is difficult because of the considerable lengths of infrasonic waves (λ_f = 17–340 m) in the case of which traditional walls, partitions, screens and sound absorbers are rather ineffective [27]. Often infrasonic noise is amplified due to the resonance of the structural members of cabins and whole objects [1], [28].

It is commonly assumed that the principal source of pressure fluctuations is the output fluctuation due to displacement pump kinematics. The dominant frequency in the output spectrum and consequently, in the pressure fluctuation spectrum, is given by the relation [29]

$$f_i = \frac{n_p z_t K}{60}\ \text{Hz} \tag{2}$$

where: n_p – pump shaft rotational speed [rpm], z_t – the number of pump displacement parts, K – the next number of the harmonic component.

For example, in the case of a 2110-type gear pump, f_1 amounts to 250 Hz (z_t = 10 teeth, pump shaft rotational speed n_p = 1500 rpm).

It is known from the measurement and operational practice relating to machines with hydrostatic drives that in the pressure fluctuation spectrum, lower frequency (also infrasonic) components appear besides the components arising from pump delivery fluctuations. This is due to external excitations and to the dynamic properties of hydraulic system connected with, among other things, resonance phenomena in the hydraulic conduits. One of the ways of reducing the impact of external low-frequency mechanical vibrations on the hydraulic valves is their flexible mounting. By mounting the valve housing flexibly (e.g., via a pack of springs) one can obtain, by matching the spring rates to the valve mass and to the frequency which is to be reduced, a reduction in pump casing vibrations in the range of low frequencies close to the resonance of the control element, e.g., a distributor slide (Fig. 13). The possibilities of insulating valves from the low-frequency vibrations of the foundation are described in more detail in [20].

The amplitude-frequency spectrum for pressure fluctuations during the starting of the hydrostatic gear of the truck-mounted crane slewing mechanism,

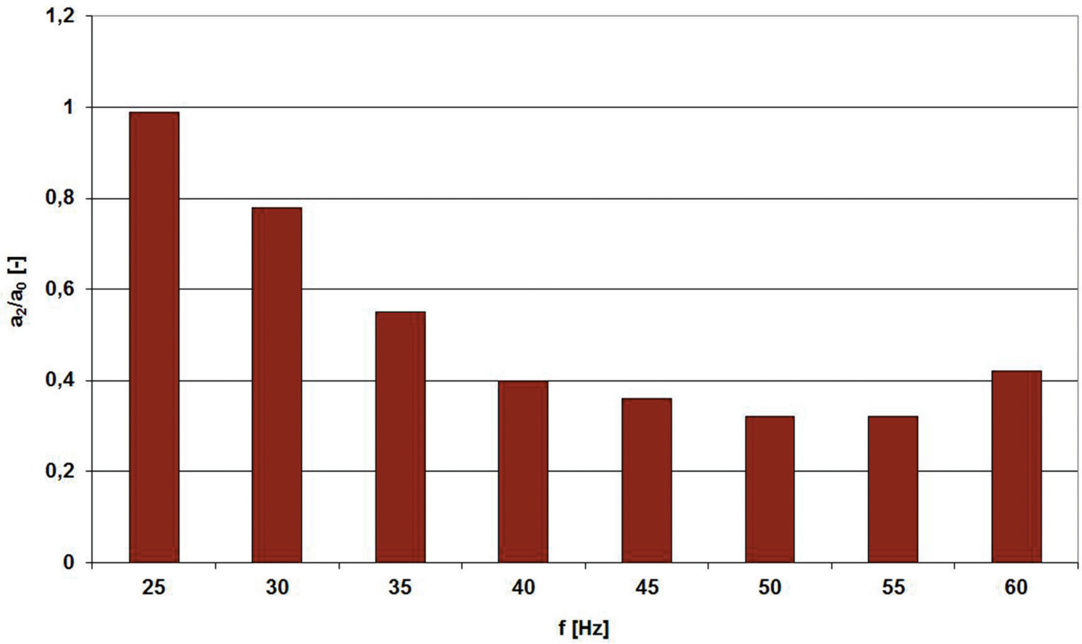

Fig. 13. Proportional distributor housing vibration acceleration amplitude a_2 relative to excitation vibration acceleration amplitude a_0 for f = 25–60 Hz [20]

shown in Fig. 14, represents a case in which lower frequency components occur.

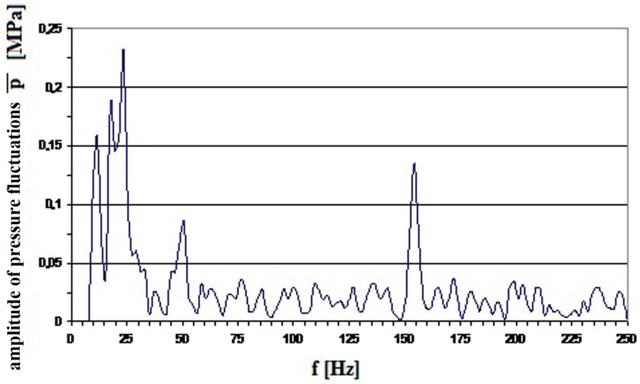

Fig. 14. Amplitude-frequency spectrum for pressure fluctuations during starting of hydrostatic gear of truck-mounted crane slewing mechanism with pressure relief valve participation

A narrow-band analysis of pressure fluctuations during the starting of the hydraulic system of the truck-mounted crane's slewing gear showed that significant pressure fluctuation amplitudes occur in a range of 5–25 Hz and at 160 Hz. The former range stems from the resonant properties of the overflow valve and the natural frequency of the slewing gear, calculated from the relation [30]

$$f = \frac{1}{2\pi} \frac{q_s}{\sqrt{\frac{V_u}{B_z} I_{zr}}} \tag{3}$$

where: q_s – specific absorbing capacity of the hydraulic engine, V_u – initial volume of the liquid in the conduits, B_z – an equivalent volume elasticity modulus taking into account the deformability of the liquid and the conduits, I_{zr} – a reduced solid moment of inertia.

The frequency of 160 Hz corresponds to the basic harmonic of displacement pump delivery fluctuations, determined by the number of displacement elements and the rotational speed of the pump shaft. The amplitudes of pressure fluctuations generated by displacement pumps at excitation frequencies higher than 150 Hz are reduced by means of passive dampers. Passive dampers operate on the principle of the interference of the pressure wave originating from the pump with the pressure wave reflected from the damper and propagating in the opposite direction.

Owing to their high effectiveness in reducing pressure fluctuation amplitudes, chamber dampers are often used in practice. A chamber damper has the form of a cylinder with length H and diameter D_T larger than diameter D of the pressure conduit, as shown in Fig. 15. The way of determining volume V_{ot} and the location of the chamber damper in the hydraulic circuit is described in detail in [31].

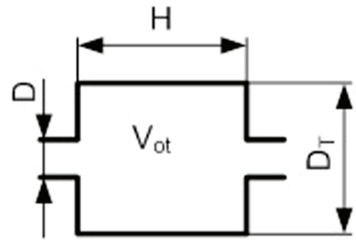

Fig. 15. Schematic of chamber damper with characteristic dimensions

By introducing the notion of damper effectiveness B_{to} one can evaluate damping effectiveness from the following definitional relation [32], [33]

$$B_{to} = 20 \log \frac{\overline{p}_1}{\overline{p}_2} \tag{4}$$

where: $\overline{p}_1$ – amplitude of pressure fluctuations ahead of the damper, $\overline{p}_2$ – amplitude of pressure fluctuations behind the damper.

The reduction in pressure fluctuation amplitudes by the chamber damper for the ideal liquid, taking into account geometric parameters of the damper, is described by the relation [33]

$$B_{tk} = 20 \log \sqrt{1 + \left[\left(\frac{D_T}{D} \right)^2 - 1 \right] \cdot \sin^2 \left(\frac{H 2\pi f_w}{c_o} \right)} \tag{5}$$

where: D_T – diameter of the chamber damper, D – diameter of the pressure conduit, H – length of the chamber damper, f_w – frequency of excitations.

The use of chamber dampers for low-frequency damping is limited by the fact that their acceptable geometric dimensions are the main determinant of their effectiveness. An example of undesirable effects (Figs. 16 and 17) of the use of a chamber damper for low-frequency damping are the results presented in [34].

The dangers connected with the occurrence and impact of low-frequency vibrations and noise in hydraulic systems have been presented. The problem is important because of the particularly harmful effects of such excitations on the human body. This is reflected in the standards concerning the introduction of machines and equipment incorporating hydraulic systems into the EU markets.

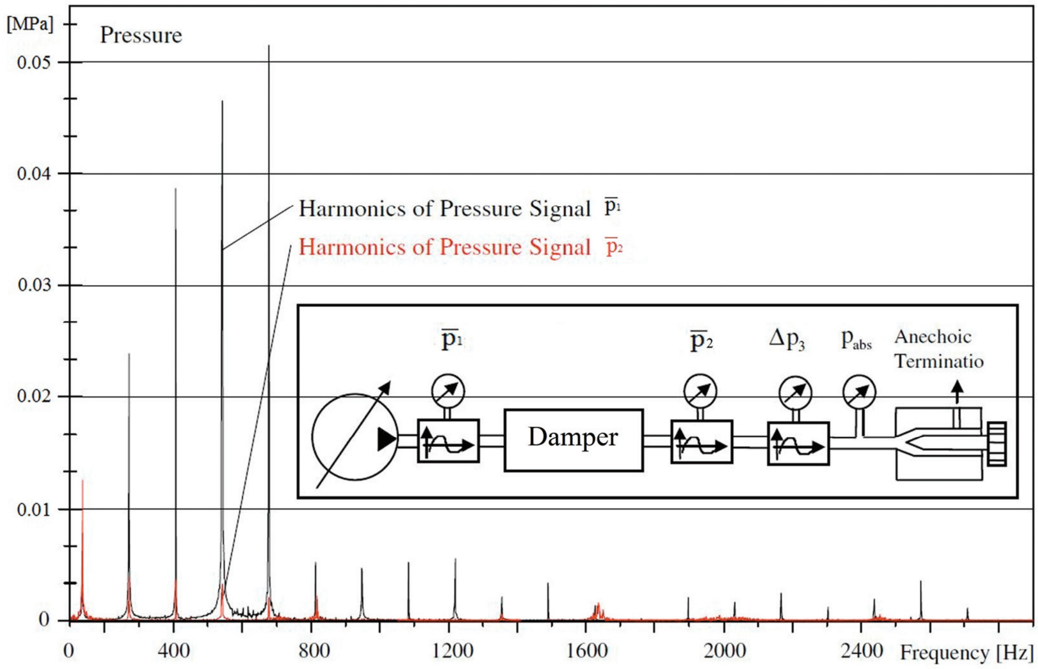

Fig. 16. Comparison between amplitudes of pressure fluctuations $\overline{p}_1$ ahead of chamber damper and those of pressure fluctuations $\overline{p}_2$ behind chamber damper, [34]

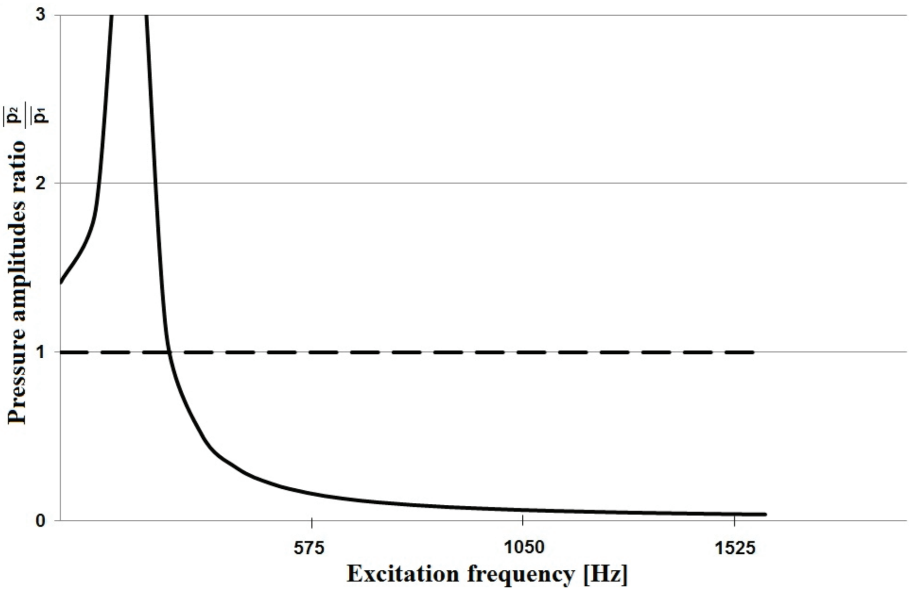

Fig. 17. Dependence between pressure amplitudes ratio and excitation frequency [34]

The special focus was on the reduction of the noise emitted by heavy engineering machines, taking into account specifically infrasonic noise and generally, low-frequency noise. It is indicated that the hydrostatic drive system is one of the main noise sources in engineering machines. Sounds are generated as a result of pressure fluctuations. An original concept of reducing pressure fluctuations in the range of low-frequency excitations through the use of a specially designed active damper has been presented. The effectiveness of the damper proposed was experimentally tested in the process of reducing pressure fluctuation amplitudes. Some of the results are shown in Fig. 18.

The damper was found to be most effective when its free frequency coincided with the excitation frequency to be reduced. The acoustic effect of the use of the active damper is illustrated in Fig. 19.

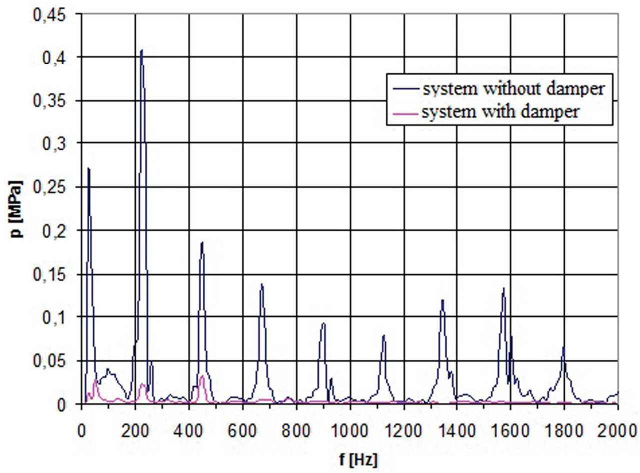

Fig. 18. Comparison of amplitude-frequency spectra for a system with and without chamber damper. Average pressure 15 MPa

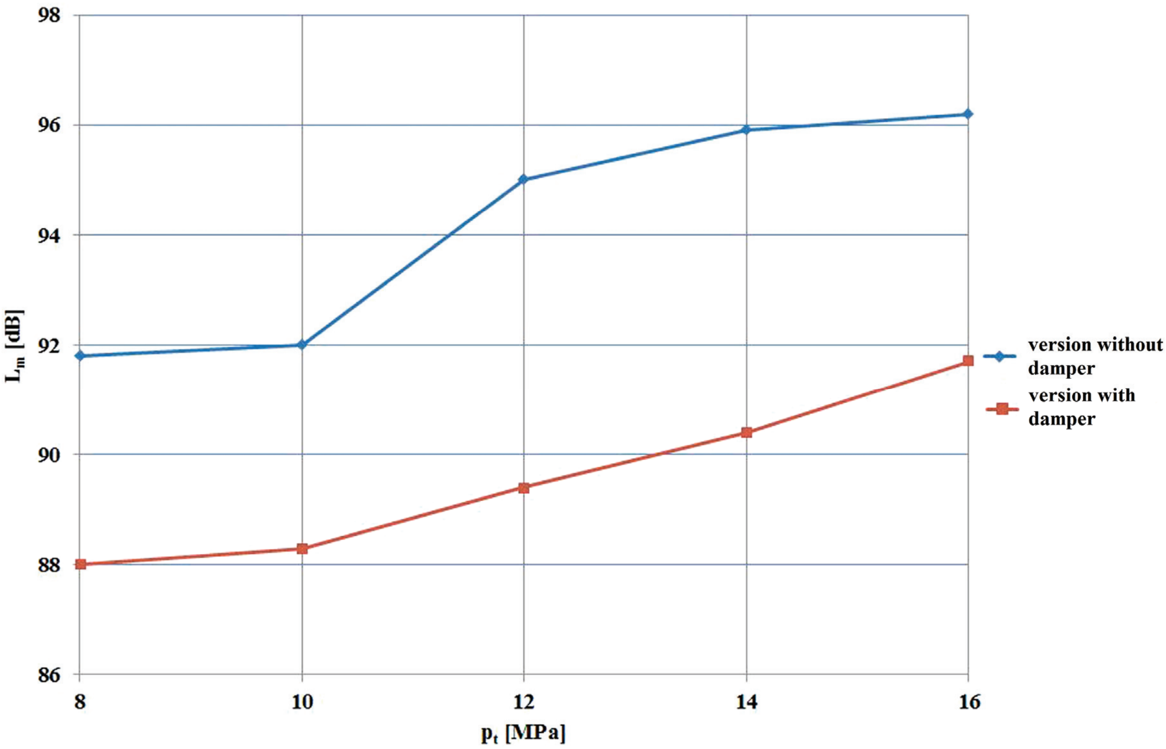

Fig. 19. Noisiness characteristic $L_m = f(p_t)$ of jib lifting hydraulic system in loader Ł-200

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