ISSN 1648-7788 / eISSN 1822-4180

VILNIUS TECH
Vilnius Gediminas Technical University

AVIATION

2024
Volume 28
Issue 1
Pages 40–48
https://doi.org/10.3846/aviation.2024.20904

THE IMPACT OF MECHANICAL VIBRATIONS ON HYDRAULIC VALVES AND THE POSSIBILITY OF REDUCING THE EFFECTS

Michał STOSIAK [1], Paulius SKAČKAUSKAS [2]✉, Adam DEPTUŁA [3]

[1]*Faculty of Mechanical Engineering, Wrocław University of Science and Technology, Łukasiewicza 5, 50-371 Wrocław, Poland*
[2]*Faculty of Transport Engineering, Vilnius Gediminas Technical University, Plytinės g. 25, 10105 Vilnius, Lithuania*
[3]*Faculty of Engineering Production and Logistics, Opole University of Technology, ul. Sosnkowskiego 31, 45-272 Opole, Poland*

Article History:
- received 1 March 2023
- accepted 16 May 2023

Abstract. The paper shows that mechanical vibrations occur in a wide frequency range in the hydraulic systems operating in the real world. Hydraulic valves are also exposed to these vibrations. The paper gives examples of vibration sources and suggests that the influence of vibrations on hydraulic valves could be reduced. Particular attention was paid to the vibrating proportional distributor. The amplitude-frequency spectrum of pressure pulsation in a hydraulic system with a vibrating proportional distributor was analysed. During the tests, the frequency of external mechanical vibrations acting on the proportional distributor and their direction was changed.

Keywords: mechanical vibrations, hydraulic system, valve, aircraft, frequency analysis.

✉Corresponding author. E-mail: ***paulius.skackauskas@vilniustech.lt***

1. Introduction

When considering a hydraulic system, we must pay particular attention to the elements that control the basic parameters of the system, such as pressure, flow rate, and direction of flow of the working medium, which play a decisive role in the operation of the energy-receiving device in the system as well as the entire hydraulic system. A specific feature of hydraulic systems is the dependence of their work on external conditions and the above-mentioned parameters. A working hydraulic element, such as a valve that controls the direction of flow or pressure, is constantly exposed to complex, exciting forces of various origins, e.g., external ground vibrations, performance pulsation, and pressure pulsation.

In real conditions, hydraulic valves, including distributors, are exposed to mechanical vibrations (Gao et al., 2021; Zhang et al., 2022). The sources of these vibrations are multiple: imbalance of rotating machine parts, variable loads, machine movements on uneven ground, aeroplane flight etc. The aircraft vibration spectrum also includes low frequencies (Krause et al., 2023). The range of vibration frequencies in a typical civil aircraft depends on the aircraft type, engine type and phase of flight (Mansfield & Aggarwal, 2022). For turbine-powered aircraft in cruise flight, frequencies between 40 Hz and 90 Hz typically dominate the vibration spectrum. New aircraft engines operating at lower speeds will shift the vibration into the lower frequency region. Generally, the complex problem of the transmission of vibrations through a machine or device can be divided into three basic, interrelated stages:

- vibration sources;
- paths of vibration transmission;
- consequences.

An important source of vibrations is the drive system, e.g., an internal combustion engine performing an operational cycle with variable characteristics (Ahirrao et al., 2018; Bovsunovsky & Nosal, 2022). A working hydraulic system is also a source of mechanical vibrations mainly caused by surges of pressure and the periodic nature of the displacement pump operation vibrations (Gao et al., 2021; Zhang et al., 2022). The vibrations generated in this way are characterised by different frequencies, paths of their transmission being different as well. The mobile machine moving over uneven ground creates excitations in the frequency range from 0.5 to 250 Hz (Yupapin & Pornsuwancharoen, 2019; Abdelkareem et al., 2021), which also contributes to the generation and propagation of noise due to the mechanical connection of the elements of the hydraulic system, which is done through hoses and common fastenings (Wegener et al., 2021; Cao et al., 2019). In this frequency band, there are excitations from the driving engine (internal combustion engine) related to the kinematics of the displacement pump, which manifest themselves in the pressure pulsations in the machine's hydraulic system (Bach et al., 2017; Xu et al., 2023; Pang et al., 2021). A serious consequence of the resulting pressure pulsation is, among other things, vibrations of hydraulic lines (Gao et al., 2021; Czerwiński & Łuczko, 2015; Łuczko & Czerwiński, 2016). Vibrating hydraulic hoses can rub against the structural components of the machine on which they are installed or against each other. This can lead to abrasion of the hose walls. This applies to flexible hoses. In extreme cases, leakage of the hose may occur at the point of complete abrasion. In the case of high-pressure hydraulic lines, due to the large pressure difference, the flow of spray agent from such a gap can reach high velocities and cause further damage to the adjacent lines.

On the other hand, vibrations caused by the resistance of the flowing air are in the frequency range of 250–16,000 Hz and are caused by the detachment of the air stream from the machine elements (Sovardi et al., 2016; Savcı et al., 2022). Phenomena related to the flow of the working medium in the hydraulic system cause vibrations and noise (He et al., 2023; Song et al., 2022; Han et al., 2017). Occasional cavitation causes high-frequency noise to occur (Jia et al., 2022; Park et al., 2020; Zhao et al., 2022; Liu et al., 2022). Thus, time-varying forces excite hydraulic system components to vibrate, and noise is generated as a result of the surface vibrations of these components (Pan et al., 2022; Zhang et al., 2018; Pan et al., 2018).

Vibrations have many negative effects and often cause vibrations of control elements in hydraulic valves (Wang et al., 2022; Bouzidi et al., 2018; Awad & Parrondo, 2020). These elements are responsible for regulating the gap size in the hydraulic valves. Since the valve control element (e.g., spool) vibrates, the area of the valve's throttling gap changes, which in the hydraulic system is manifested by pulsation of performance and, consequently, of pressure, with the components of the spectrum of this pulsation corresponding to the components of the vibration spectrum of the excited element. Our investigations and literature studies (Wang et al., 2021; Josifovic et al., 2016; Stosiak et al., 2023) show that the resonant frequencies of hydraulic valve control elements (conical poppets, spools) are below 100 Hz (Stosiak et al., 2023). The range of these frequencies is also particularly dangerous for humans since the resonant frequencies of vital internal organs are also below 100 Hz (Zheng et al., 2019; Arnold et al., 2018; Govindan et al., 2020). These pulsations can also be excited by infrasound. Low-frequency pressure pulsations can cause unstable operation of hydraulic receivers, which contributes to their inaccurate operation.

The influence of vibration on the environment, including people, machines and devices, is covered by the 2016 and 2017 standards, still in force, with the numbers: PN-B-02170:2016-12 (Polski Komitet Normalizacyjny, 2016) and PN-B-02171:2017-06, respectively (Polski Komitet Normalizacyjny, 2017). These standards provide methods for the assessment and degrees of sensitivity of machines, devices, buildings, and people to external mechanical vibrations (Engel & Zawieska, 2010). The harmful effect of such vibrations is assessed by measuring the amplitude of the effective vibration velocity at the place of installation of the machine or device and comparing it with the permissible values given in Table 1. For sensitivity classes II-V, the permissible amplitude is compared with the maximum value of vibrations occurring in the direction. For class I, on the other hand, it is compared with the velocity vector modulus $v_{dop} = \sqrt{v_x^2 + v_y^2 + v_z^2}$, where v_x, v_y, v_z – components of the velocity vector in the directions x, y, z.

Table 1 shows that in all sensitivity classes, there are groups of machines equipped with hydraulic elements and systems. A particularly strong influence of mechanical vibrations is to be expected in class I, in which there are precision instruments and machine tools equipped with hydrotronic and micro-hydraulic elements, which are subject to increased requirements, including those relating to the accuracy and repeatability of work.

In order to effectively counteract the negative effects of vibrations, one must not only know the causes of vibrations but also find paths where these vibrations distribute.

Table 1. Classes of sensitivity of groups of machines and devices to external mechanical vibrations according to (Polski Komitet Normalizacyjny, 2009).

Sensitivity class	Sensitivity description	The name of the group of machines, devices	Permissible speed $v_{perm} \times 10^{-3}$, m/s
I	very sensitive	balancing and adjustment devices instruments, microscopes, interferometers and other precision instruments, computers, precision machine tools	0.1
II	medium sensitive	grinders for threads, gears, bearings; precise milling machines, lathes	1
III	not very sensitive	ordinary lathes, milling machines, drills, grinders, textile and weaving machines, typographic machines	3
IV	almost insensitive	motors, chisel mortisers, sewing machines, metalworking and woodworking machines, presses, trimmers	6
V	completely insensitive	fans, crushers, grinders, shakers, vibrating tables and sieves, screens, hammers	>6

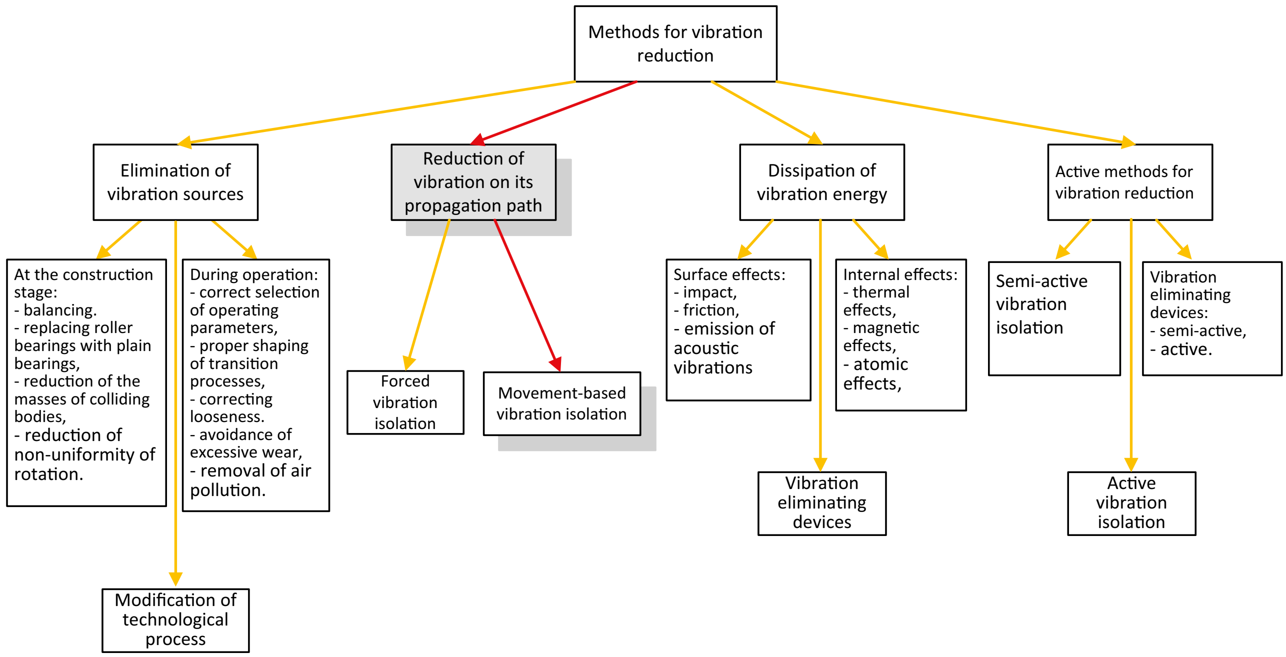

Figure 1. The main methods for reducing vibration

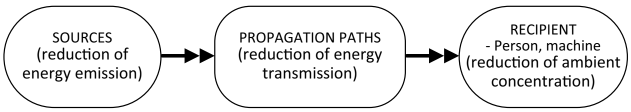

Figure 2. General scheme of action to reduce vibroacoustic hazards

Generally, this is a very difficult task due to the complexity of the machine or device design. Typically, machines are complex multi-mass systems with many degrees of freedom, complicated structure with holonomic and non-holonomic constraints, contain elastic and damping elements, sometimes with non-linear characteristics (Karpenko & Nugaras, 2022), with clearances in kinematic pairs.

Efforts should be made to reduce vibration. The main methods for reducing vibration are presented in an illustrative way in Figure 1.

Results of vibration reduction are the best when several methods are used simultaneously (Figure 2), and this is through e.g., (Engel & Zawieska, 2010):

- reducing the vibroacoustic energy generated by the source;
- reducing the transmission of vibroacoustic energy along the path of its transmission;
- reducing the ambient concentrations in specific areas of workstations.

Pathways of vibroacoustic energy transmission should be identified in a way that will enable taking further steps, e.g., changing the location of vibration sources, changing their location on foundations, using active or passive methods of vibration isolation, using dampers, cabins, partitions, sets of shims to isolate the valve body from the vibrating ground or to isolate the valve control element from the vibrating body. The susceptible mounting of hydraulic valve bodies on a vibrating machine frame can be considered. The parameters of the vibration isolator must be selected so as to limit the vibrations of the valve body over a wide range of excitation frequencies. Limiting the vibration of the valve body is at the same time limiting the excitations acting on the valve control element. In simplified analyses, a vibration-isolated hydraulic valve can be regarded as a single-mass system – due to the fact that the mass of the valve body is usually much greater than that of the valve control element. However, in more detailed considerations, such a system should be treated as a two-mass system, and its characteristics should be selected based on the expected range of excitations. The selection of passive vibration isolators should take into account their characteristics, dimensions and oil resistance. Vibration isolators with non-linear characteristics are worth investigating.

The topic discussed is also particularly important with regard to high-power hydraulic systems mounted on decks aboard seagoing ships, where a serious source of external vibrations is the ship's main internal combustion engine and hull subjected to cyclic deformation in the sea wave (Banaszek & Petrovic, 2019, Banaszek et al., 2018).

2. Vibrations of the valve control element caused by external excitations

In order to determine the influence of external mechanical vibrations on hydraulic valves, experimental tests were conducted. The testing process used the HYDROPAX ZY25 simulator of the linear hydrostatic drive as the source of mechanical vibrations with a fixed amplitude and frequency. The hydraulic diagram of the modified simulator system is shown in Figure 3. The ZY25 simulator consists of three main parts:

- hydraulic part;
- SYHCE1 control device;
- HCE1 control program.

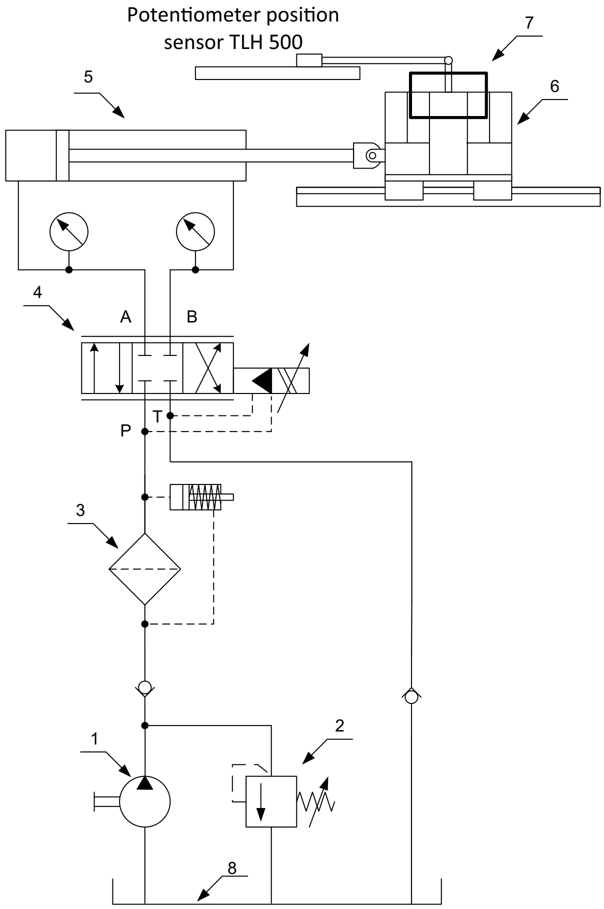

Figure 3. Hydraulic diagram of the linear hydrostatic drive simulator that generated mechanical vibrations: 1 – feeding pump; 2 – adjustable maximum valve; 3 – oil filter; 4 – electrohydraulic amplifier 4WSE2EM10-45; 5 – working actuator; 6 – simulator table; 7 – tested element: hydraulic lift valve, proportional spool valve; 8 – hydraulic oil tank

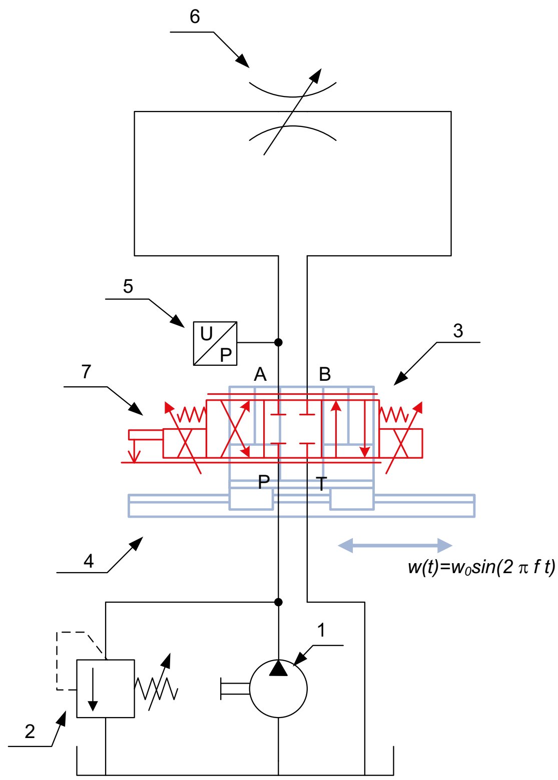

Figure 4. Distribution of measurement points in the test system of the proportional distributor: 1 – pump supplying the test system of the considered element; 2 – overflow valve; 3 – tested proportional distributor 4WRE 6 E08-12/24Z4/M from Mannesmann-Rexroth placed on the simulator table; 4 – hydraulic simulator table HYDROPAX ZY25, which kinematically excites the tested element according to a harmonic function and has the table position measuring point; 5 – pressure change measurement point with the piezoelectric sensor M101A04 from Piezotronics; 6 – adjustable throttle valve as a load of the test system; 7 – measurement of the distributor spool position with an inductive sensor

During the tests, a measuring system was used that enabled the measurement and recording of the following quantities:

- changing the position of the hydraulic simulator table – potentiometer position sensor TLH-500;
- position of the controller of the tested element (in the case of a proportional spool distributor) – inductive spool position sensor;
- pressure pulsation at the point of the presence of the tested element – piezoelectric pressure transducer from Piezotronics.

The measurement system enabled measurement, recording, processing in real-time, and saving the values on a hard disc of a connected PC. As a result, measurement files were obtained in ASCII and graphic format. The measurement data were then processed using comprehensive analysis and visualisation software ORIGIN Professional 7.5 and an EXCEL spreadsheet. The arrangement of the measurement points is shown in Figure 4.

A proportional distributor was installed in a custom-made holder of the simulator table in such a way that the direction of external vibrations was parallel to the direction of movement of the spool in the sleeve (first series of tests) and in such a way that the direction of the external vibrations was perpendicular to the movement of the spool in the sleeve (series second test), by rotating the tested distributor by 90°. A constant electrical control signal was applied to the coils of the proportional solenoids of the distributor, causing the spool to deflect 2 mm from its neutral position. The flow rate of the working medium through the tested manifold was 1×10^{-4} m^3/s (6 dm^3/min). The average pressure value at measurement point 5 (according to Figure 4) was 2 MPa. During the tests, valve 2 (according to Figure 4) remained closed. For clarity, a comparison of the test results for three selected external excitation frequencies (table vibrations) is presented: 40, 50, and 60 Hz in the form of amplitude-frequency spectra of pressure pulsation (measured at point 5 according to Figure 4) and displacement of the spool in the body of the tested proportional valve (p. 7 according to Figure 4) – Figures 5–10.

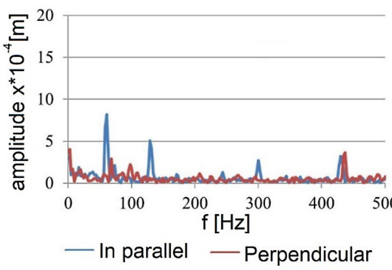

Figure 5. The amplitude-and-frequency spectrum of pressure pulsation in a system with a vibrating proportional distributor – vibration frequency of the simulator table is 40 Hz

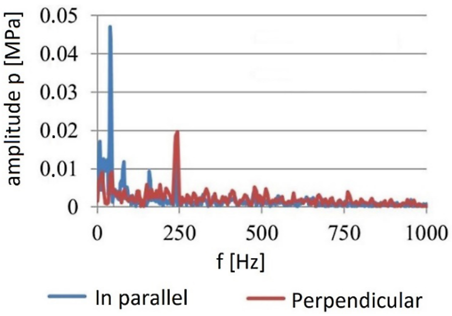

Figure 6. The amplitude-and-frequency spectrum of the proportional distributor spool vibrations – the vibration frequency of the simulator table is 40 Hz

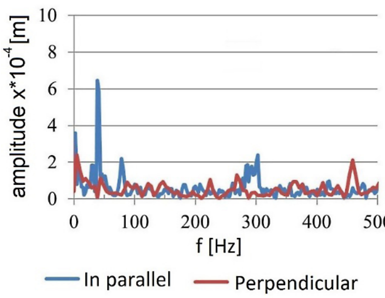

Figure 7. The amplitude-and-frequency spectrum of pressure pulsation in a system with a vibrating proportional distributor – vibration frequency of the simulator table 50 Hz

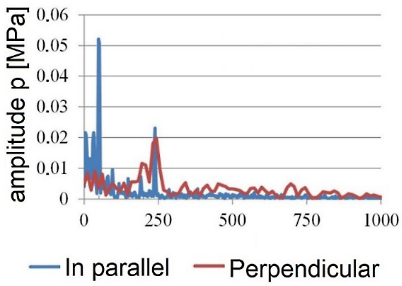

Figure 8. The amplitude-and-frequency spectrum of the proportional distributor spool vibrations – the vibration frequency of the simulator table is 50 Hz

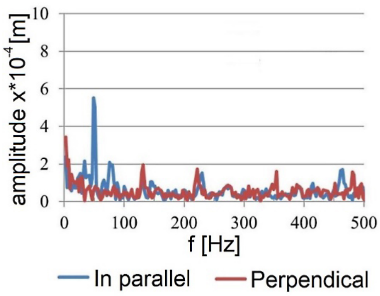

Figure 9. The amplitude-and-frequency spectrum of pressure pulsation in a system with a vibrating proportional distributor – vibration frequency of the simulator table 60 Hz

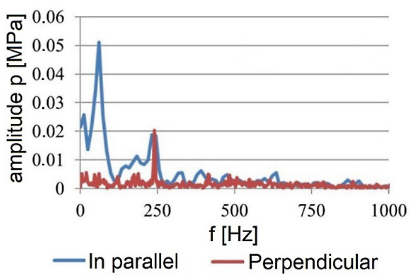

Figure 10. The amplitude-and-frequency spectrum of the proportional distributor spool vibrations – the vibration frequency of the simulator table is 60 Hz

From the depicted vibration waveforms (in the form of an amplitude-and-frequency spectrum) of the kinematically excited spool, it can be seen that at the external excitation frequency of approx. 60 Hz, the amplitude of these vibrations reaches its maximum in the analysed frequency range. At this external excitation frequency, the amplitude of the spool vibration increases by almost 45% compared to the amplitude observed for the excitation frequency f = 50 Hz. The presented results prove that the spool is excited to vibrate by external mechanical vibrations. The spool vibrations induced in this way cause changes in the amplitude-and-frequency spectrum of pressure pulsations in the system. There appear harmonic components with frequencies corresponding to the frequencies of the spool vibrations and the external excitation. In addition, the presented results of experiments indicate that the transmission of vibrations to the spool is significantly influenced by the angle α contained between the direction of spool movement and the direction of external excitation (external vibrations). Two extreme cases were considered, i.e., the direction of the spool movement is consistent with the direction of external vibrations (angle α = 0°) and the direction of spool movement perpendicular to the direction of external vibrations (angle α = 90°). In Figures 6, 8, and 10, excitation of the spool vibrations is observed. The dominant components of the spectrum of these vibrations correspond to the vibration frequency of the simulator table (frequencies of external excitations). The excited vibrations of the spool also cause periodic changes in the size of the throttling gaps of the proportional distributor, which results in the appearance of components in the spectrum of pressure pulsations with frequencies corresponding to the excited vibrations of the spool – Figures 5, 7 and 9. In the pressure pulsation spectrum, there was a component with a frequency of approx. 240 Hz, and caused by the performance pulsation of the positive displacement pump of the hydraulic system in which the tested distributor operated (Figure 4).

3. Theoretical analysis of the possibility of reducing vibration

When selecting the method of reducing the influence of external mechanical vibrations on the hydraulic distributor and its spool, the limitations concerning the dimensions of vibration isolators, their rigidity and damping should be taken into account. Geometric limitations result from the building conditions, i.e., the available space in which the insulator can be installed (e.g., set of elastomer shims). Theoretical considerations were carried out using a single-mass model with one degree of freedom, Figure 11. In this model, m represents the mass of the distributor, and c and k are the equivalent stiffness and equivalent damping of the flexible elements (shims) between the distributor body and the vibrating floor.

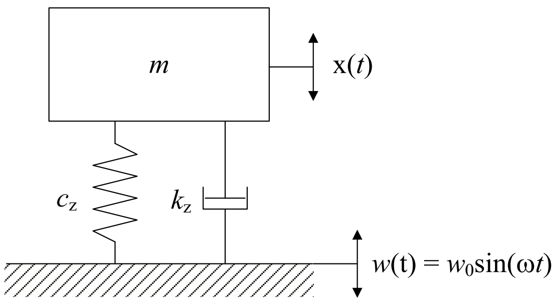

Figure 11. Model of a vibrating system with one degree of freedom

A body of mass m is excited to vibrate by a kinematic excitation in the form of a harmonic function given in the following form:

$$w = w_0 \sin(\omega t), \tag{1}$$

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 (distributor body or spool) is described by the equation:

$$m\ddot{x} + k_z(\dot{x} - \dot{w}) + c_z(x - w) = 0. \tag{2}$$

In order to limit the influence of vibrations of the distributor body, it is necessary to minimize the value of the expression describing the amplitude of absolute vibrations of the distributor body:

$$x_0 = w_0 \cdot \sqrt{\frac{1 + \left(2\gamma\frac{\omega}{\omega_0}\right)^2}{\left(1 - \left(\frac{\omega}{\omega_0}\right)^2\right)^2 + \left(2\gamma\frac{\omega}{\omega_0}\right)^2}} \to \min, \tag{3}$$

where: w_0 – excitation amplitude [m]; ω – excitation frequency [rad/s]; ω_0 – natural frequency of the system [rad/s]; γ – dimensionless coefficient of attenuation.

For steady motion, the phase shift angle Φ varies from 0 to π, and its fastest changes can be observed near the point where the excitation frequency equals the natural frequency of the vibrating mass – Figure 12. In this case, this angle is expressed by the formula (Harris & Piersol, 2009):

$$\Phi = arctg\frac{2\gamma\frac{\omega}{\omega_0}}{\frac{1}{\left(\frac{\omega}{\omega_0}\right)^2} - 1 + 4\gamma^2}. \tag{4}$$

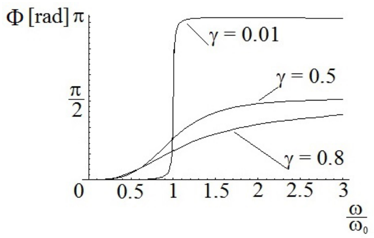

Figure 12. Phase shift angle δ as a function of the ratio between frequency and different values of dimensionless damping coefficient γ

In assessing the effectiveness of the isolation of the distributor body from the vibrating ground, the ratio between the absolute displacement amplitude of the manifold body x_0 and the ground vibration amplitude w_0 can be used. This coefficient can be called the amplification factor T_a (Harris & Piersol, 2009; Schmitz, 2012):

$$T_a = \frac{x_0}{w_0} = \sqrt{\frac{1 + \left(2\gamma\frac{\omega}{\omega_0}\right)^2}{\left(1 - \left(\frac{\omega}{\omega_0}\right)^2\right)^2 + \left(2\gamma\frac{\omega}{\omega_0}\right)^2}}. \tag{5}$$

The expected properties of isolating materials for distributor body vibrations can be interpreted graphically using the relationship between the coefficient T_a and the frequency ratio for a fixed dimensionless damping γ (Figure 13) or in the case of varying the dimensionless damping coefficient, Figure 14.

In addition, the effectiveness of the vibration isolation can be used as defined by the relationship (Engel & Zawieska, 2010):

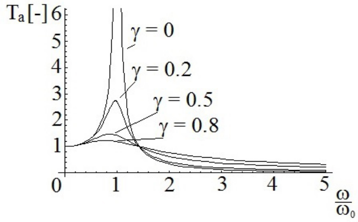

Figure 13. Dependence of coefficient T_a on the ratio of excitation frequency ω and the natural frequency ω_0 for a fixed value of the dimensionless attenuation coefficient γ

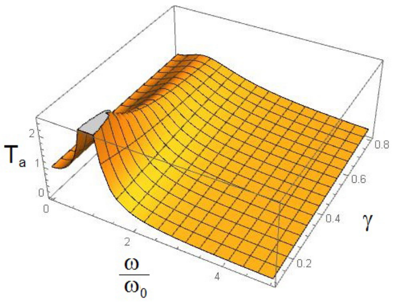

Figure 14. Dependence of coefficient T_a on the ratio of excitation frequency ω and the natural frequency ω_0 and dimensionless damping coefficient γ

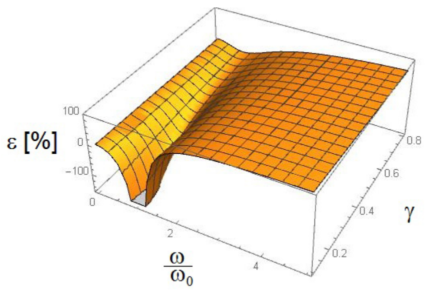

Figure 15. Dependence of coefficient ε on the ratio of excitation frequency ω and the natural frequency ω_0 and dimensionless damping coefficient γ

$$\varepsilon = \left(1 - \frac{x_0}{w_0}\right) 100\%. \quad (6)$$

In this case, the parameters c and k should be selected so that the efficiency of vibration isolation, determined by the Equation (6), is as high as possible, i.e., close to 100% (Figure 15).

Figure 15 shows that the vibration isolation will be more effective with a greater ratio of the excitation frequency to the natural frequency of the distributor body.

4. Conclusions

The paper presents some effects of mechanical vibrations on the hydraulic distributor. The most unfavourable effects of vibrations are observed at a frequency close to the resonance frequency of the distributor spool and when the direction of vibrations coincides with the direction of the spool movement. The oscillating spool causes a change in the area of the throttling surface in the spool pair of the distributor. Variable throttling causes pressure pulsations with frequencies corresponding to the vibration frequencies of the spool. The theoretical considerations presented in this paper may be helpful in designing a passive vibration isolation system with suitable properties. The following general conclusions can be drawn about the effectiveness of the vibration isolation system:

If $\omega \ll \omega_0$, then for force excitation, the ratio between the amplitude of the force transferred to the ground to the amplitude of the excitation force is close to one, similar to kinematic excitation, the value of the ratio between the amplitude of fixed vibrations to the amplitude of the excitation is also close to one, the use of vibrators has no effect;

If $\omega \approx \omega_0$, then the value of the ratio between the amplitude of the force transferred to the ground to the amplitude of the excitation force increases and reaches significant values at low damping. The elastic support should prevent operation in this range because the forces transmitted by the system, in this case, may have greater amplitudes than the excitation forces or absolute displacements with amplitudes greater than the displacement of the excitation-amplification factor T_a, given by the relationship (5), greater than 1;

If $\frac{\omega}{\omega_0} > \sqrt{2}$, then vibration isolation is effective, i.e., the value of the amplification coefficient T_a is less than 1, regardless of the type of damping and the values of coefficients defining it. However, the effectiveness of vibration isolation is greater the lower the damping is;

If the ratio ω / ω_0 continues to increase, the efficiency of vibration isolation, determined by the relationship (6), approaches 100%; however, starting from the value $\frac{\omega}{\omega_0} = 5$, the rate of increase in the effectiveness of vibration isolation ε is significantly slowed down, and for a dimensionless damping coefficient γ = 0, it practically does not occur. For $\frac{\omega}{\omega_0} \gg \sqrt{2}$ it can be noticed that the influence of the dimensionless damping coefficient on the effectiveness of vibration isolation decreases;

In the resonance range, the value of the amplification coefficient T_a can be reduced by using high damping. Therefore, if a machine or device equipped with a hydraulic valve operates for a long time in the resonance range, a material with high damping should be used to prevent excessive vibration amplitudes of the valve body.

Disclosure statement

The authors do not have any competing financial, professional, or personal interests from other parties.

References

Abdelkareem, A. A. M., Xu, L., Jing, X., Eldaly, B. M. A., Zou, J., & Ali, M. K. A. (2021). Field measurements of the harvestable power potentiality of an off-road sport-utility vehicle. *Measurement*, *179*, Article 109381. https://doi.org/10.1016/j.measurement.2021.109381

Ahirrao, N. S., Bhosle, S. P., & Nehete, D. V. (2018). Dynamics and vibration measurements in engines. *Procedia Manufacturing*, *20*, 434–439. https://doi.org/10.1016/j.promfg.2018.02.063

Arnold, J. J., & Griffin, M. J. (2018). Equivalent comfort contours for fore-and-aft, lateral, and vertical whole-body vibration in the frequency range 1.0 to 10 Hz. *Ergonomics*, *61*(11), 1545–1559. https://doi.org/10.1080/00140139.2018.1517900

Awad, H., & Parrondo, J. (2020). Hydrodynamic self-excited vibrations in leaking spherical valves with annular seal. *Alexandria Engineering Journal*, *59*(3), 1515–1524. https://doi.org/10.1016/j.aej.2020.03.033

Bach, D., Masselter, T., & Speck, T. (2017). Damping of pressure pulsations in mobile hydraulic applications by the use of closed cell cellular rubbers integrated into a vane pump. *Journal of Bionic Engineering*, *14*(4), 791–803. https://doi.org/10.1016/S1672-6529(16)60444-4

Banaszek, A., & Petrovic, R. (2019). Problem of non proportional flow of hydraulic pumps working with constant pressure regulators in big power multipump power pack unit in open system. *Technical Gazette*, *26*(2), 294–301. https://doi.org/10.17559/TV-20161119215558

Banaszek, A., Łosiewicz, Z., & Jurczak, W. (2018). Corrosion influence on safety of hydraulic pipelines installed on decks of contemporary product and chemical tankers. *Polish Maritime Research*, *98*(2), 71–77. https://doi.org/10.2478/pomr-2018-0056

Bouzidi, S. E., Hassan, M., & Ziada, S. (2018). Experimental characterisation of the self-excited vibrations of spring-loaded valves. *Journal of Fluids and Structures*, *76*, 558–572. https://doi.org/10.1016/j.jfluidstructs.2017.11.007

Bovsunovsky, A., & Nosal, O. (2022). Highly sensitive methods for vibration diagnostic of fatigue damage in structural elements of aircraft gas turbine. *Procedia Structural Integrity*, *35*, 74–81. https://doi.org/10.1016/j.prostr.2021.12.050

Cao, H., Kang, T., & Chen, X. (2019). Noise analysis and sources identification in machine tool spindles. *CIRP Journal of Manufacturing Science and Technology*, *25*, 26–35. https://doi.org/10.1016/j.cirpj.2019.04.001

Czerwiński, A., & Łuczko, J. (2015). Parametric vibrations of flexible hoses excited by a pulsating fluid flow, Part II: Experimental research. *Journal of Fluids and Structures*, *55*, 174–190. https://doi.org/10.1016/j.jfluidstructs.2015.03.007

Engel, Z., & Zawieska, M. (2010). *Noise and vibrations in working processes sources risk assessment* (1st ed.). CIOB PIB (in Polish).

Gao, P., Tao Yu, T., Zhang, Y., Wang, J., & Zhai, J. (2021). Vibration analysis and control technologies of hydraulic pipeline system in aircraft: A review. *Chinese Journal of Aeronautics*, *34*(4), 83–114. https://doi.org/10.1016/j.cja.2020.07.007

Govindan, R., Saran, V. H., & Harsha, S. P. (2020). Low-frequency vibration analysis of human body in semi-supine posture exposed to vertical excitation. *European Journal of Mechanics – A/Solids*, *80*, Article 103906. https://doi.org/10.1016/j.euromechsol.2019.103906

Han, M., Liu, Y., Wu, D., Zhao, X., & Tan, H. (2017). A numerical investigation in characteristics of flow force under cavitation state inside the water hydraulic poppet valves. *International Journal of Heat and Mass Transfer*, *111*, 1–16. https://doi.org/10.1016/j.ijheatmasstransfer.2017.03.100

Harris, C., & Piersol, A. (2009). *Harris' shock and vibration handbook* (6th ed.). McGraw Hill.

He, J., Zhang, Y., Liu, X., Li, B., Sun, S., Peng, J., & Liu, W. (2023). Experiment and simulation study on cavitation flow in pressure relief valve at different hydraulic oil temperatures. *Flow Measurement and Instrumentation*, *89*, Article 102289. https://doi.org/10.1016/j.flowmeasinst.2022.102289

Jia, J., Zhang, J., & Huang, Z. (2022). Cavitation flow and broadband noise source characteristics of NACA66 hydrofoil with a V groove on the suction surface. *Ocean Engineering*, *266*(2), Article 112889. https://doi.org/10.1016/j.oceaneng.2022.112889

Josifovic, A., Roberts, J. J., Corney, J., Davies, B., & Shipton, Z. K. (2016). Reducing the environmental impact of hydraulic fracturing through design optimisation of positive displacement pumps. *Energy*, *115*(1), 1216–1233. https://doi.org/10.1016/j.energy.2016.09.016

Karpenko, M., & Nugaras, J. (2022). Vibration damping characteristics of the cork-based composite material in line to frequency analysis. *Journal of Theoretical and Applied Mechanics*, *60*(4), 593–602. https://doi.org/10.15632/jtam-pl/152970

Krause, L., Par, S., & Töpken, S. (2023). Pleasantness ratings for vertical whole-body vibration on an aircraft seat and relevant body parts involved. *Applied Acoustics*, *207*, Article 109330. https://doi.org/10.1016/j.apacoust.2023.109330

Liu, J., Liu, Z., Wu, J., Li, Z., Chen, P., & Gu, X. (2022). Visualization experiment and numerical calculation of the cavitation evolution inside the injector ball valve. *Fuel*, *329*(1), Article 125500. https://doi.org/10.1016/j.fuel.2022.125500

Łuczko, J., & Czerwiński, A. (2016). Experimental and numerical investigation of parametric resonance of flexible hose conveying non-harmonic fluid flow. *Journal of Sound and Vibration*, *373*, 236–250. https://doi.org/10.1016/j.jsv.2016.03.029

Mansfield, N. J., & Aggarwal, G. (2022). Whole-body vibration experienced by pilots, passengers and crew in fixed-wing aircraft: A state-of-the-science review. *Vibration*, *5*(1), 110–120. https://doi.org/10.3390/vibration5010007

Pan, Y., Li, Y., Huang, M., Liao, Y., & Liang, D. (2018). Noise source identification and transmission path optimisation for noise reduction of an axial piston pump. *Applied Acoustics*, *130*, 283–292. https://doi.org/10.1016/j.apacoust.2017.10.009

Pan, Y., Liu, R., Bin, G., & He, X. (2022). Vibration and noise reduction of phononic crystal structure laid on the noise transmission path of axial piston pump. *Applied Acoustics*, *200*, Article 109075. https://doi.org/10.1016/j.apacoust.2022.109075

Pang, H., Wu, D., Deng, Y., Cheng, Q., & Liu, Y. (2021). Effect of working medium on the noise and vibration characteristics of water hydraulic axial piston pump. *Applied Acoustics*, *183*, Article 108277. https://doi.org/10.1016/j.apacoust.2021.108277

Park, C., Kim, D. G., Yim, G. T., Park, Y., & Moon, I. (2020). A validation study of the model test method for propeller cavitation noise prediction. *Ocean Engineering*, *213*, Article 107655. https://doi.org/10.1016/j.oceaneng.2020.107655

Polski Komitet Normalizacyjny. (2009). *PN-EN ISO 9612:2009 Akustyka – Wyznaczanie zawodowej ekspozycji na hałas – Metoda techniczna*. https://sklep.pkn.pl/pn-en-iso-9612-2009e.html

Polski Komitet Normalizacyjny. (2016). *PN-B-02170:2016-12 Ocena szkodliwości drgań przekazywanych przez podłoże na budynki*. https://sklep.pkn.pl/pn-b-02170-2016-12p.html

Polski Komitet Normalizacyjny. (2017). *PN-B-02171:2017-06 Ocena wpływu drgań na ludzi w budynkach*. https://sklep.pkn.pl/pn-b-02171-2017-06p.html

Savcı, İ. H., Şener, R., & Duman, İ. (2022). A study of signal noise reduction of the mass air flow sensor using the flow conditioner on the air induction system of heavy-duty truck. *Flow*

Measurement, and Instrumentation, *83*, Article 102121. https://doi.org/10.1016/j.flowmeasinst.2022.102121

Schmitz, T. L. (2012). *Mechanical vibrations: Modeling and measurement* (1st ed.). Springer-Verlag. https://doi.org/10.1007/978-1-4614-0460-6

Song, P., Wei, Z., Zhen, H., Liu, M., & Ren, J. (2022). Effects of prewhirl and blade profile on the hydraulic and cavitation performance of a centrifugal pump. *International Journal of Multiphase Flow*, *157*, Article 104261. https://doi.org/10.1016/j.ijmultiphaseflow.2022.104261

Sovardi, C., Jaensch, S., & Polifke, W. (2016). Concurrent identification of aero-acoustic scattering and noise sources at a flow duct singularity in low Mach number flow. *Journal of Sound and Vibration*, *377*, 90–105. https://doi.org/10.1016/j.jsv.2016.05.025

Stosiak, M., Karpenko, M., Deptuła, A., Urbanowicz, K., Skačkauskas, P., Cieślicki, R., & Deptuła, A. M. (2023). Modelling and Experimental verification of the interaction in a hydraulic directional control valve spool pair. *Applied Sciences*, *13*(1), Article 458. https://doi.org/10.3390/app13010458

Wang, H., Chen, Z., & Huang, J. (2021). Improvement of vibration frequency and energy efficiency in the uniaxial electro-hydraulic shaking tables for sinusoidal vibration waveform. *Energy*, *218*, Article 119477. https://doi.org/10.1016/j.energy.2020.119477

Wang, H., Lai, Z., Wu, D., Zhang, K., & Zheng, M. (2022). Investigation of the friction-induced vibration of a novel four-way reversing valve using spectral kurtosis and number of peaks spectrum. *Mechanical Systems and Signal Processing*, *166*, Article 108425. https://doi.org/10.1016/j.ymssp.2021.108425

Wegener, K., Bleicher, F., Heisel, U., Hoffmeister, H. W., & Möhring, H. C. (2021). Noise and vibrations in machine tools. *CIRP Annals*, *70*(2), 611–633. https://doi.org/10.1016/j.cirp.2021.05.010

Xu, W., Wang, Z., Zhou, Z., Sun, C., Zhang, J., Yan, R., & Chen, X. (2023). An advanced pressure pulsation model for external gear pump. *Mechanical Systems and Signal Processing*, *187*, Article 109943. https://doi.org/10.1016/j.ymssp.2022.109943

Yupapin, P., & Pornsuwancharoen, N. (2019). Asphalt road surface vibration and force distribution generated by pickup truck braking. *Measurement*, *148*, Article 106871. https://doi.org/10.1016/j.measurement.2019.106871

Zhang, D., Juan, M., Zhang, Z., Gao, P., Jin, J., Wang, J., & Yu, T. (2022) A dynamic modeling approach for vibration analysis of hydraulic pipeline system with pipe fitting. *Applied Acoustics*, *197*, Article 108952. https://doi.org/10.1016/j.apacoust.2022.108952

Zhang, J. D., Kabir, K. M. M., Lee, H. E., & Donald, W. A. (2018). Chiral recognition of amino acid enantiomers using high-definition differential ion mobility mass spectrometry. *International Journal of Mass Spectrometry*, *428*, 1–7. https://doi.org/10.1016/j.ijms.2018.02.003

Zhao, L., Wu, J. Y., Jin, Z. J., & Qian, J. Y. (2022). Cavitation effect on flow resistance of sleeve regulating valve. *Flow Measurement and Instrumentation*, *88*, Article 102259. https://doi.org/10.1016/j.flowmeasinst.2022.102259

Zheng, G., Qiu, Y., & Griffin, M. J. (2019). Fore-and-aft and dual-axis vibration of the seated human body: Nonlinearity, cross-axis coupling, and associations between resonances in the transmissibility and apparent mass. *International Journal of Industrial Ergonomics*, *69*, 58–65. https://doi.org/10.1016/j.ergon.2018.08.007