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DISADVANTAGEOUS EFFECTS OF VIBRATIONS ON THE MICRO-HYDRAULIC RELIEF VALVE: EXPERIMENTAL APPROACH

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Abstract. The paper discusses the negative impact of external mechanical vibrations, which lead to the malfunction of drive systems, particularly affecting hydrostatic drives. The hydraulic system components feature a spring-supported control element that vibrates due to external mechanical vibrations, leading to pressure pulsation. The resulting pressure pulsation causes many unfavorable characteristics of hydraulic system operation. The positive displacement pump is shown as the main source of pressure pulsation in a hydraulic system. For selected frequencies of external mechanical vibration close to the natural frequency of the valve control element, the resulting pressure pulsation far exceeds the pressure pulsation resulting from the displacement pump. This paper presents selected results showing pressure pulsations as a consequence of the displacement pump and external mechanical vibrations acting on the pressure-relief valve.

Keywords: microhydraulics, valve, vibrations, pressure pulsation, experimental research, frequency spectrum, aircraft, aviation industry.

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1. Introduction

Hydraulic drives are widely used in the aviation industry, mainly because of their high force with relatively small dimensions. They are designed to control the control surfaces of the aileron, indicator, elevator and auxiliary rudder due to the high positioning accuracy of the actuators of hydraulic drives and low inertia allowing for fast movements. Hydraulic drives are also used to retract and extend the chassis, enabling its locking and amortization using the capacitance present in the system. Hydraulic actuators are also used for main, differential and parking braking systems. In larger aircraft, hydraulic drives are also used to open and close sliding doors and loading ramps, this is due to smooth speed regulation and controlled movements, especially important when loading and unloading goods. Such systems are found in transport, military and civilian cargo aircraft. Damage to the hydraulic drive will result in failure of the system containing the hydraulic drive. Hydraulic drives are required to be reliable, precise, resistant to high loads and temperature changes. This requires continuous monitoring and supervision by aviation authorities. Operating machinery and equipment with micro-hydraulic systems is a source of mechanical vibrations across a wide spectrum of frequencies (Stosiak et al., 2023b, 2023c, Stosiak & Karpenko, 2024). The sources of vibrations often include drives, such as an internal combustion engine with a periodic duty cycle and variable characteristics, imbalances in rotating machine parts and electric motor shafts, or variable loads (Stosiak, 2015; Bureika, 2024). The sources of vibration found in the environment are very diverse and can generally be divided into deterministic and random, external and internal (Kilikevicius et al., 2019). This also pertains to aircraft, which are subject to internal vibrations from the operation of propulsion components and external vibrations caused by the environment acting on the aircraft's skin (Karpenko et al., 2024). Machines and their components can also experience vibrations due to the movement of machines in their environment. An example of such self-excited vibrations is the flutter vibrations of aircraft wings. These are usually high-frequency vibrations. In many practical cases, forces cause vibrations to occur directly on machine components or may result from the interaction of other machine components or equipment equipment (Karpenko et al., 2024; Zhang et al., 2024). In addition to forces that periodically change their value over time, the source of vibration can also be forces whose value remains constant, but their direction or point of application changes (e.g., inertial centrifugal forces). The coincidence of mechanical vibration and pressure pulsation in micro-hydraulic systems can be considered in two ways. Firstly, the vibration of micro-hydraulic system components is induced by pulsating flow (e.g., vibration of microlines, microvalves), and secondly, it is due to pressure pulsation induced by vibration of micro-hydraulic system components and, in particular, excitation of micro-valve controls (e.g., micro-distributor slides or lift microvalve poppets) (Stosiak & Towarnicki, 2022; Stosiak et al., 2023a; Yin et al., 2023). Hydraulic systems cannot operate without pressure control valves, as pressure is the main parameter in hydraulic systems. The lift valve is the primary valve, operating on the principle of supplying pressurized fluid under a ball supported by a prestressed spring. The ball will be moved, causing the liquid to flow, when a force is obtained from the pressure acting on the designated area of the ball. The control element can be a ball, a plug (poppet) in the form of a cone or a plate, as described earlier. The spring is prestressed by a screw that presses down on the spring (Stecki & Garbacik, 2002; Li et al., 2024). Such valves have the drawback of pressure variation based on the flow rate of the fluid through the valve, as discussed in a previous section. Another disadvantage is that the valve operates unstably, which is a consequence of the pressure pulsation generated by the displacement elements of the pump. A momentary increase in pressure causes the valve to open, while a momentary decrease in pressure causes it to close. The absence of damping of the closing element causes this element to fall into vibration, causing additional pressure pulsation (Kudźma & Stosiak, 2012; Szydelski, 1980; Adeoye et al., 2017). This causes problems with the positioning of the actuator, vibrations, noise and faster wear of seals and kinematic pairs. Pressure pulsation causes periodically varying forces acting on each component of the hydraulic system (Puzyrewski & Sawicki, 1987). In particular, variable forces act on the hydraulic lines of the aircraft systems. Agitated vibrations can cause damage to lines, which is particularly dangerous in aircraft (Karpenko, 2022; Karpenko & Nugaras). Damage to the suction pipe of a pump, even of a small size, may cause cavitation in the pump suction port, causing noise and faster wear (Rabsztyn & Klarecki, 2023). The vibration of hydraulic lines also has hydrodynamic effects in the lines, the analysis of which is a complex issue (Karpenko & Bogdevičius, 2020; Karpenko et al., 2023). In many hydraulic systems, filtration of hydraulic fluids is important, which may cause valve operation disturbances (Klarecki et al., 2024).

The purpose of this study is to experimentally determine the frequency range of external mechanical vibrations at which changes are observed in the pressure pulsation spectrum of a hydraulic system with a pressure relief valve. A harmonic component is expected in the pressure pulsation spectrum at a frequency corresponding to the frequency of external mechanical vibrations acting on the micro-hydraulic relief valve.

2. Measuring stand and test object

To determine the influence of external mechanical vibration, a lift microvalve was developed in line with implementation dossier in Kollek (2011), which was designed based on previous work dealing with lift valve operation (Kudźma & Stosiak, 2012). The operating pressure setting range of the valve depends on the stiffness of the spring. If the valve is to operate at a lower pressure, then a spring with a lower stiffness should be inserted. According to the static characteristic of the valve, it's operating pressure changes when the flow rate through the valve is adjusted despite the preset pressure. It is sought that the change in pressure from the change in flow rate be as small as possible possible (Bouzidi et al., 2018). The stiffer the spring, the steeper the static characteristics of the valve.

The measurements to determine the formation of changes in the pressure pulsation spectrum due to external mechanical vibrations were carried out on a previously prepared measuring stand described in Kudźma (2001), which is shown in Figure 1. The external mechanical vibration was applied to a hydraulic relief microvalve.

The power source in the measuring stand is an external gear pump driven by a three-phase electric motor with variable speed implemented using a frequency converter. The pump used has 9 teeth, and its drive shaft rotated at a speed of 750 rpm for a flow of 0.6 dm^3/min; 1000 rpm for a flow of 0.8 dm^3/min; 1250 rpm for a flow of 1 dm^3/min. According to the defining Equation (1), the harmonics of the pump performance pulsations caused by the kinematics of the operating pump's displacement elements:

$$f_K = \frac{n_p \cdot z}{60} \cdot K \left[\mathrm{Hz}\right], \tag{1}$$

where: n_p – speed of the pump's drive shaft [rpm], z – number of pump's teeth, K – number of the component of the pulsation spectrum.

For $K = 1$, the first harmonic component appears at 112.5 Hz for a flow of 0.6 dm^3/min; 150 Hz for a flow of 0.8 dm^3/min; 187.5 Hz for a flow of 1 dm^3/min. The performance pulsation thus induced, along with the impedance of the micro-hydraulic system, produces pressure pulsations in the hydraulic system with a frequency equal to that of the performance pulsation.

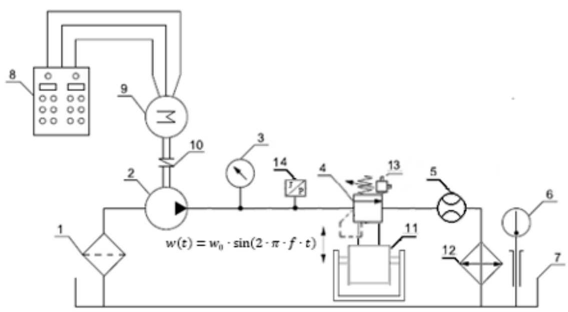

Figure 1. Diagram of the hydraulic system for testing the effect of external mechanical vibrations on hydraulic microvalves: 1 – oil filter, 2 – gear pump WPH PZ3A1G, 3 – pressure gauge Parker ServiceJunior SCJN-400-01, 4 – microvalve under test, 5 – flow meter Parker KSCVF-002-10-07X, 6 – thermometer Elmetron PT-217, 7 – oil tank, 8 – control cabinet, 9 – three-phase electric motor, 10 – flexible coupling, 11 – electrodynamic mechanical vibration exciter TiraVib, 12 – oil cooler with controller, 13 – piezoelectric acceleration sensor PCB Piezotronics 340A16, 14 – piezoelectric pressure sensor PCB Piezotronics 105C23 (source: Stosiak et al., 2020)

The average pressure was measured using a pressure gauge installed between the pump's discharge port outlet and the connection port of the pressure-relief valve. In addition, a pressure pulsation sensor was added to the system. An oval-gear flowmeter was mounted on the drain line of the valve under test, allowing actual measurement of the flow rate of the working medium. An oil-air cooler with an additional fan adjustable by a controller was used to stabilize the working temperature of the fluid. A uniaxial accelerometer attached to the valve body in the direction of external mechanical vibration was used to measure the acceleration and frequency of the valve. The forcing frequency was generated in the range of 100–900 Hz with an increment of 10 Hz. The oil poured into the hydraulic system is Azolla 22AF, ρ = 865 kg/m^3, υ(40 °C) = 22.5 mm^2/s. The system generating external mechanical forces included a vibration exciter and a special-purpose measurement amplifier from TiraVib. A computer with specialized software was used to record measurements and control the system that generated external mechanical forces shown in Figure 2. The measurement system is shown in Figure 2.

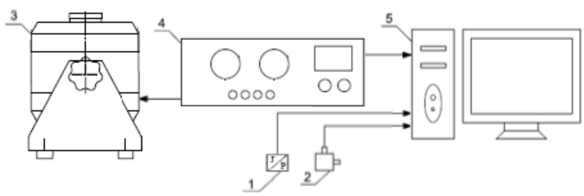

Figure 2. Measurement system: 1 – piezoelectric pressure sensor PCB Piezotronics 105C23, 2 – piezoelectric acceleration sensor PCB Piezotronics 340A16, 3 – electrodynamic mechanical vibration exciter TiraVib, 4 – measurement amplifier BA 1000 TiraVib, 5 – computer (source: Stosiak et al., 2020)

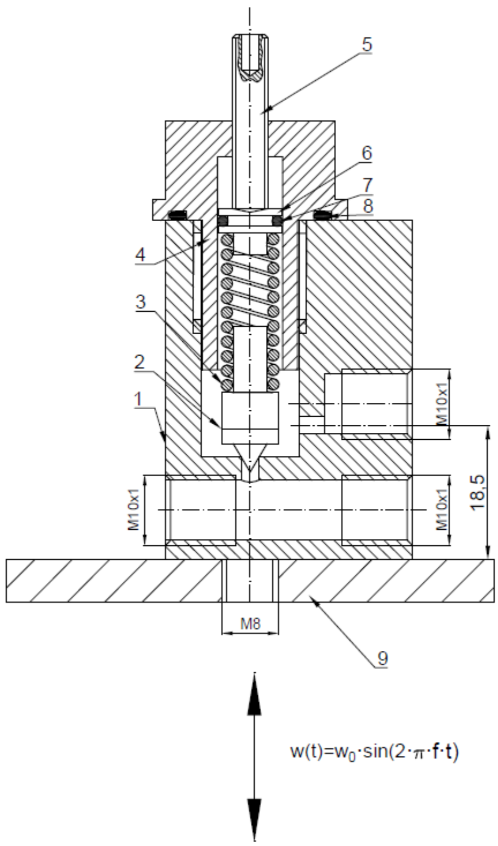

Figure 3. Axial cross-section of the pressure-relief microvalve: 1 – body, 2 – plug, 3 – spring, 4 – main screw, 5 – set screw, 6 – piston, 7 – piston seal, 8 – screw seal, 9 – bracket (source: Stosiak et al., 2020)

The intensity of external mechanical forces and the effect on the hydraulic system was measured by an acceleration sensor and a pressure pulsation sensor, respectively. Communication between the measurement system and the recording of measurements was possible thanks to PUMA software. The final measurement results and characterizations were obtained in Microsoft Excel and converted using OriginPro 9.0. The valve tested was the relief microvalve shown in Figure 3. Mechanical forces acting on the control element were applied along the axis of the controls.

The valve shown has a classic sharp-edged seat (Kudźma & Stosiak, 2012) with a closing element in the form of a cone, also called a plug. The pressure and intensity valves have typical designs. They have high tightness, simple and low-cost geometry, and are resistant to contamination. The valve in Figure 3 weighs 700 g. The closing element has a conical design with an opening angle of 30°. The opening pressure of the valve results from the spring prestressed with the adjustment screw. To maintain the spring's stability, the piston is guided by a suitable mechanism, and an additional channel with a circular sealing ring is created to ensure tightness. The design allows for the springs supporting the closing element to be changed with different stiffness levels. This is mainly due to a change in the diameter of the wire from which the spring was made. The valve was attached to the vibration exciter via a rigid connection. The valve attached onto the vibration exciter is shown in Figure 4 and 5.

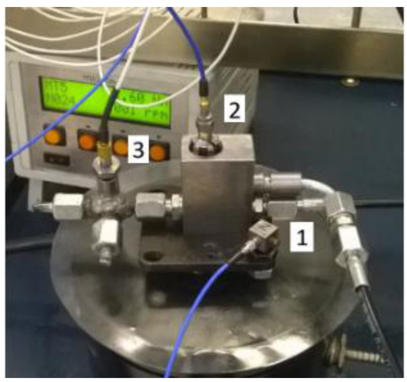

Figure 4. View of the tested relief microvalve: 1 – triaxial piezoelectric acceleration sensor PCB Piezotronics 356B21, 2 – single axis piezoelectric acceleration sensor PCB Piezotronics 340A16, 3 – piezoelectric pressure sensor PCB Piezotronics 105C23

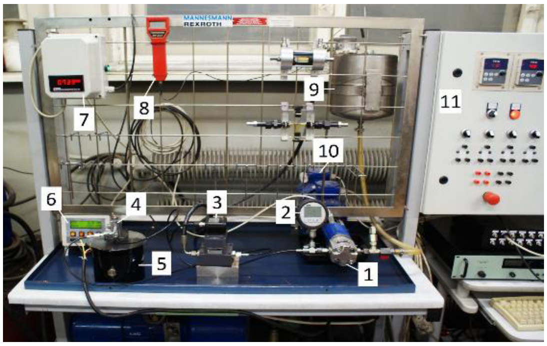

Figure 5. Test bench for the effect of vibrations on micro-hydraulic valves: 1 – gear pump, 2 – pressure sensor, 3 – flow meter, 4 – valve under test with pressure and acceleration sensors, 5 – vibration exciter, 6 – tachometer and torque meter display, 7 – flow meter display, 8 – thermometer, 9 – oil tank, 10 – electric motor, 11 – control cabinet

3. Measurement results

The effect of external mechanical vibrations on the amplitude-frequency spectrum of pressure pulsations was measured for three flow rates of 1, 0.8 and 0.6 dm^3/min, and with three average pressures of 5, 8 and 10 MPa for each flow rate. Selected measurement results are presented in Figures 6, 7, 8, 9, 10, 11.

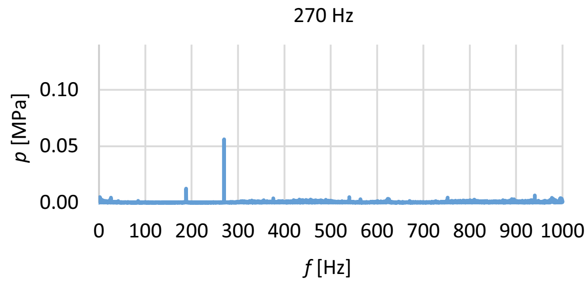

Figure 6. Amplitude-frequency spectrum of pressure pulsations at a flow rate of 1 dm^3/min, at a pressure of 8 MPa and external forcing for a frequency of 270 Hz, spring stiffness of 7.49 N/mm

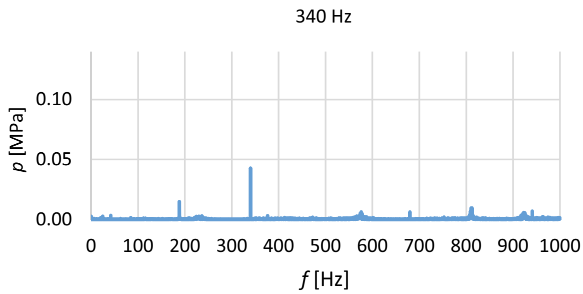

Figure 7. Amplitude-frequency spectrum of pressure pulsations at a flow rate of 1 dm^3/min, at a pressure of 5 MPa and external forcing for a frequency of 340 Hz, spring stiffness of 7.49 N/mm

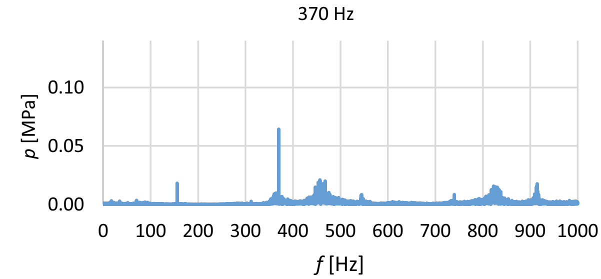

Figure 8. Amplitude-frequency spectrum of pressure pulsations at a flow rate of 0.8 dm^3/min, at a pressure of 8 MPa and external forcing for a frequency of 370 Hz, spring stiffness of 7.4 N/mm

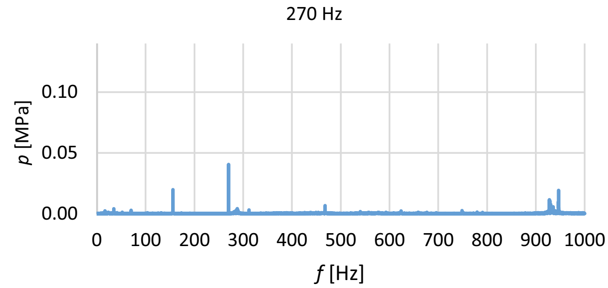

Figure 9. Amplitude-frequency spectrum of pressure pulsations at a flow rate of 0.8 dm^3/min, at a pressure of 5 MPa and external forcing for a frequency of 270 Hz, spring stiffness of 7.49 N/mm

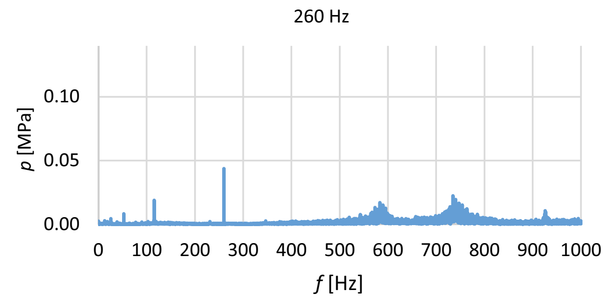

Figure 10. Amplitude-frequency spectrum of pressure pulsations at a flow rate of 0.6 dm^3, at a pressure of 8 MPa and external forcing for a frequency of 260 Hz, spring stiffness of 7.49 N/mm

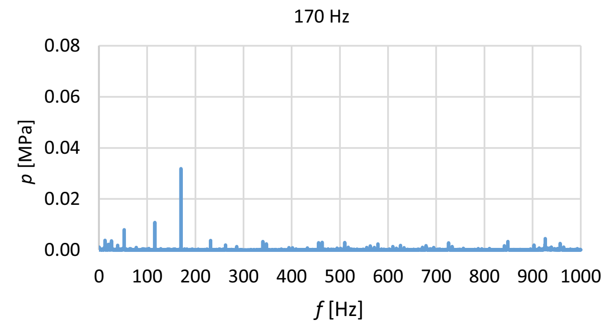

Figure 11. Amplitude-frequency spectrum of pressure pulsations at a flow rate of 0.6 dm^3/min, at a pressure of 5 MPa and external forcing for a frequency of 170 Hz, spring stiffness of 7.49 N/mm

Significant pulsation occurs, especially when the frequency of the force acting on the element is near the resonant frequency of the valve control element, which has a mass of 4.4 g. In the simplest case, the resonant frequency of the valve plug can be determined using the well-known Equation (2):

$$f_0 = \frac{1}{2\pi} \cdot \sqrt{\frac{c}{m}} \text{ [Hz]}, \tag{2}$$

where: c – stiffness of the valve spring [N/m], m – mass of the valve plug [kg].

More accurate values of the valve's resonant frequency are obtained when the static characteristics of the valve, its dimensions, and operating parameters are taken into account (Kudźma, 2012; Kollek et al., 2010) (Equation (3)):

$$f_{0z} = \frac{1}{2\pi} \cdot \sqrt{\frac{W_z \cdot \mu \cdot \pi^2 \cdot d_z^2 \cdot sin\alpha_g \cdot \sqrt{\frac{2p_1}{\rho_0}}}{4m}} \text{ [Hz]}, \tag{3}$$

where: W_z – valve amplification factor, the measure of which is the tangent of the slope angle of the linearized static characteristic of the valve, μ – valve flow rate, d_z – valve seat diameter, α_g – half of the opening angle of the valve plug cone, ρ_0 – density of the working medium, p_1 – value of the valve operating pressure, m – mass of the valve plug.

Considering the numerical characteristics of the valve, its operating conditions, and the static characteristics of the valve, the eigenfrequency of the microvalve was determined to be 207 Hz, as per Equation (3).

Therefore, for the proper operation of hydraulic valves, it is important that the resonant frequency of a given valve control element, such as a spool, plug, or ball, is outside the range of external mechanical vibration, especially in the quasi-steady state. The pressure pulsation caused by external forcing sometimes exceeds the pressure pulsation caused by the kinematics of the pump's displacement components. The maximum measurement uncertainty is the sum of individual measurement uncertainties in the entire measurement path, starting from the sensor and ending with the data recording. Piezotronics piezoelectric sensors with built-in electrometric amplifiers were used for the research. They are produced for a specific range of measured values. They can be used with typical coaxial cables, the length of which does not have to be precisely defined. Their measurement accuracy is 2%. The measurement accuracy of the amplifier is approximately 1% and the same applies to the computer. Taking into account the above data, the measurement uncertainty of the obtained data is within the range of 4%.

4. Conclusions

Hydraulic systems cannot operate without pressure control valves, as pressure is the main parameter in hydraulic systems. The lift valve is the primary valve, operating on the principle of supplying pressurized fluid under a ball supported by a prestressed spring. Such valves have the disadvantage of pressure changing with the rate of fluid flowing through the valve. Another disadvantage is that the valve operates unstably, which is a consequence of the pressure pulsation generated by the displacement elements of the pump. A momentary increase in pressure causes the valve to open, while a momentary decrease in pressure causes it to close. The absence of damping of the closing element causes this element to fall into vibration, causing additional pressure pulsation. For this reason, valves with a damping element were introduced to reduce the adverse effects of pressure pulsations. Pressure pulsation also occurs when the valve control element is excited by external mechanical vibrations acting on the valve body. For this reason, the valve body must be isolated from the ground with rubber vibration isolators and other elements. Rubber isolators are produced in large batches, which results in a low price and high availability. Different materials should be used for micro-hydraulic valves, as they have low mass. This would involve developing vibration isolators made of porous rubber or foamed polyurethane. Another approach involves reducing the vibrations of the valve control element by making structural modifications. A damping disc is utilized in patent study No. 221214 to minimize the vibrations of the closing element in the maximum pressure relief microvalve. An alternative method for dampening the vibration of the control element of the lift valve is described in patent application P443296, "Hydraulic lift valve with compensation of external mechanical vibration," and patent application P443297, "Hydraulic lift valve with compensation of external mechanical vibration." In most cases, the main component dominating the pressure pulsation spectrum is the first harmonic caused by the kinematics of the pump operation. In a special case, this value may be lower than the pulsation resulting from the activation of the valve control element, which is presented in the attached pressure pulsation spectra. Pressure pulsation may be amplified by wave phenomena occurring in the hydraulic conduit, especially when the length of the conduit corresponds to half or a quarter of the length of the propagated wave. In the future, further research directions are planned to reduce the impact of external mechanical vibrations transmitted to the valve through passive vibration isolation. Due to the low weight of the valves, the recommended materials for vibration isolation are porous rubber and foamed polyurethane.

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