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MODEL AND EXPERIMENTAL ANALYSIS OF INFLUENCE OF SELECTED PARAMETERS OF HYDRAULIC LINE ON DISPLACEMENT PUMP PRESSURE PULSATIONS DAMPING

Modelowa oraz doświadczalna analiza wpływu wybranych parametrów przewodu hydraulicznego na zjawisko tłumienia pulsacji ciśnienia pompy wyporowej

Abstract: *Among practitioners, the prevailing view is that the use of a discharge line made of an elastic hose allows users not only to isolate the pump unit from the rest of the system for transferred vibrations, but also to obtain the passive damping of pressure pulsation generated by the pump flow rate pulsation. The objective of the model and experimental research described in this article was to confirm or disprove the common opinion that a discharge line with an elastic hose is a sufficient element to damp the pressure pulsation caused by the displacement pump. The article presents experimental studies of the influence of various hydraulic lines on pressure pulsation damping. The research was carried out in the time and frequency domains. A mathematical model of a hydraulic long transmission line is also presented and an analysis of the parameters of hydraulic lines affecting the amplification or damping of pump pressure pulsations was carried out.*
Keywords: pressure pulsation, hydraulic line, hydraulic drives, pressure pulsation damping

Streszczenie: *Wśród praktyków dominuje pogląd, że zastosowanie linii tłocznej zbudowanej z odcinka przewodu elastycznego pozwala nie tylko odizolować zespół pompowy od reszty układu pod kątem przenoszonych drgań, ale także uzyskać efekt pasywnego tłumienia pulsacji ciśnienia generowanej przez pulsację wydajności pompy wyporowej. Celem przedstawionych w niniejszym artykule badań modelowych oraz eksperymentalnych było potwierdzenie lub obalenie powszechnego przekonania, że sama linia tłoczna z przewodem elastycznym jest wystarczającym elementem tłumiącym pulsację ciśnienia, spowodowaną*

pulsacją wydajności pomp wyporowych. W artykule przedstawiono wyniki przeprowadzonych analiz wpływu przewodów hydraulicznych o różnych parametrach geometrycznych i sztywnościach na tłumienie pulsacji ciśnienia. Wyniki badań przeanalizowano w domenie czasu oraz domenie częstotliwości.
Słowa kluczowe: pulsacja ciśnienia, linia hydrauliczna, napędy hydrostatyczne, tłumienie pulsacji ciśnienia

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1. Introduction

Monitoring and analysis in the time and frequency domains of pressure pulsation in the discharge line of pumps used in hydrostatic drive systems allows the detection of selected anomalies in the operation of displacement units. Conducting analysis of the pulsation course in the time domain leads to the detection of mechanical damage to the displacement pump elements (e.g., tooth damage in a gear pump, a piston pulling out in a multi-piston pump, problems with a blade in a vane pump, lack of alignment of vortex elements, etc.). Frequency domain pulsation analysis helps to detect the phenomenon of aeration inside a displacement unit, which can protect the pump from accelerated wear or damage. However, due to the unfavorable impact of pressure pulsations on the generated noise [1,2], vibrations [3, 4, 5, 6, 7, 8], fatigue strength of the drive components, as well as the accuracy of the receivers positioning, efforts are being made to minimize the impact of the described phenomenon on the drive system operation [9, 10]. Methods of damping pressure pulsations in hydraulic lines using active and passive damping elements are described in detail in the literature [11, 12, 13, 14, 15, 16].

Among practitioners, the prevailing view is that using a discharge line made of a section of elastic hose allows users not only to isolate the pump unit from the rest of the system (mainly in terms of generated vibrations), but also to obtain the effect of passive damping of pressure pulsations generated by the pump flow pulsation [14, 17, 18, 19, 20, 21].

The objective of the research described in this article was to confirm or disprove the common belief that a discharge line with an elastic hose is sufficient to dampen the pressure pulsation caused by the displacement pump instantaneous capacity pulsation.

2. Mathematical model of a hydraulic line

Due to fluctuations in the positive displacement pump capacity, there is a pulsating flow in hydraulic lines (rigid and flexible). This flow is the sum of two factors: the time-average component of the flow velocity and the variable component (mostly harmonically or polyharmonically variable). The presented flow can take three forms: laminar (stratified), transitional and turbulent. The parameters affecting the type of flow, apart from the average component of the flow velocity, are: the frequency and amplitude of oscillations and the flow duration. The type of flow in the line has a significant impact on the hydraulic line model. Laminar flow occurs in a circular hose if the dimensionless Reynolds number is less than $\mathrm{Re} \leq 2100$. Above this value, transient flow occurs in the hose. Above $\mathrm{Re} \geq 8330$, fully formed turbulent flow appears in the hose.

When analyzing issues related to pulsating flow generated by positive displacement pumps, special attention should be paid to the phenomena accompanying a hydraulic long transmission line. A hydraulic line should be considered as long when the line length is equal to or greater than the length of the pressure wave propagated in it. In [14, 19, 20], it was recommended to treat a hydraulic line as long if it meets the following condition:

$$\lambda_f \leq \frac{c_0}{10 \cdot f_{max}} \tag{1}$$

where:
λ_f – hydraulic line length [m],
c_0 – velocity of pressure wave propagation (front velocity) in the hydraulic line [m/s],
f_{max} – maximum excitation frequency [Hz].

If the condition is met, then the line is treated as an element with distributed parameters. Then, the fact that changes in pressure p and flow rate Q propagate along the duct axis with a certain velocity in the form of traveling and reflected waves should be taken into account [14]. In this case, potential resonance phenomena resulting in more intensive pressure pulsation amplitudes should be considered.
In the literature devoted to modeling in hydraulic long line systems [17, 28], the two most commonly used methods of describing transient or quasi-steady waveforms are described:

- frequency method,
- a method for studying transient processes as a function of time.

The conducted model studies focused on analyzing dynamic phenomena of the hydraulic line using the frequency method [14]. In this method, a hydraulic long line is treated as a hydraulic two-port network with two inputs and outputs. For a quasi-steady pulsating flow, the instantaneous value of the flow rate and pressure can be written in the form [14]:

$$\tilde{q} = Q_s + q_s, \tag{2}$$

$$\tilde{p} = P_s + p_s, \tag{3}$$

where:
$\tilde{q}$ – instantaneous flow rate [dm^3/min],
$\tilde{p}$ – instantaneous pressure value [MPa],
Q_s – average flow rate [dm^3/min],
P_s – average pressure value [MPa],
q_s – deviation from the average flow rate [dm^3/min],
p_s – deviation from the mean pressure value [MPa].

Assuming a laminar flow type (Re < 2100) of the working fluid in the line, the hydraulic two-port network takes the form (Fig. 1).

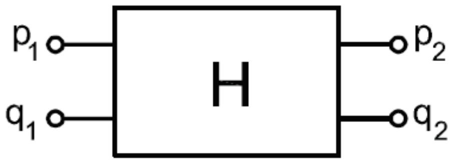

Fig. 1. Hydraulic two-port network, where: p_1 – Fourier transform of the pressure at the beginning of the line, q_1 – Fourier transform flow rate at the beginning of the line, H – operator transition function, p_2 – Fourier transform of the pressure at the end of the line, q_2 - Fourier transform flow rate at the end of the line.

For the described case, the transmittance matrix takes the form:

$$H = \begin{bmatrix} h_{11} & h_{12} \\ h_{21} & h_{22} \end{bmatrix}, \tag{4}$$

where the matrix terms are written as:

$$h_{11} = cosh(T \cdot \psi_z \cdot j\omega) \tag{5}$$

$$h_{12} = Z_c \cdot \psi_z \cdot sinh(T \cdot \psi_z \cdot j\omega) \tag{6}$$

$$h_{21} = \frac{1}{Z_c \cdot \psi_z} \cdot sinh(T \cdot \psi_z \cdot j\omega) \tag{7}$$

$$h_{22} = cosh(T \cdot \psi_z \cdot j\omega) \tag{8}$$

The hose or pipe impedance Z_c is described by the equation:

$$Z_c = \frac{\rho \cdot c_0}{\pi \cdot R^2} \tag{9}$$

where:
Z_c – impedance of hydraulic line [$kg/m^5 \cdot s$],
ρ – fluid density [kg/m^3]
ω – extortion frequency [rad/s],
R – internal radius of the hydraulic line [m].

The time constant T is defined by the equation:

$$T_c = \frac{L}{c_0} \tag{10}$$

where:
T_c – time constant [s],
L – length of the considered hydraulic line [m].

Determining the correct velocity of the pressure wave propagation c_0 in the hydraulic line is important due to its influence on both the amplitude and frequency of the transmission module. This value is determined from the equation [14]:

$$c_0 = \sqrt{\frac{\beta_c}{\rho\left(1+\frac{\beta_c \cdot \varphi d_h}{E \cdot g_p} \cdot c_1\right)}} \tag{11}$$

where:
c_0 – pressure wave propagation velocity [m/s],
β_c – bulk modulus of the fluid [Pa],
φd_h– line flow diameter [m],
E – Young's modulus of the hydraulic line material [Pa],
g_p – line wall thickness [m],
c_1 – constant depending on the method of attaching the hydraulic line.

Analyzing the components of equation (11), it can be seen that the pressure wave velocity depends not only on the hydraulic fluid parameters (β_c, ρ), but also on the hydraulic line parameters (φd_h, E, q_p).

For a line fixed on both sides, the value of the constant c_1 is determined with the following equation:

$$c_1 = 1 - \vartheta_p^2 \tag{12}$$

where:
ϑ_p – Poisson's ratio of the hydraulic line material.

Equation (12) is useful under the assumption of a unidirectional stress state, thus it is limited to thin-walled steel tubes only. In hydraulic drive systems, due to high pressure values, lines that do not qualify as thin-walled elements are used [14]. Therefore, the presented equation is of limited use. The relationship is also not applicable to flow lines equipped with elastic hoses (due to the lack of information regarding the modulus of elasticity of multiple-braid hoses' material).

In [14], the c_0 pressure wave propagation velocity was experimentally determined depending on the value of pressure p for selected hydraulic lines:

- steel tube with internal diameter ϕd_h = 9mm and wall thickness g_p = 1.5mm:

$$c_{0r} = -0{,}2508p^2 + 10{,}296p + 1190{,}7 \tag{13}$$

- 1 wire braided hose:

$$c_{0r} = -0{,}8041p^2 + 30{,}7466p + 668{,}51 \tag{14}$$

- 2 wire braided hose:

$$c_{0r} = -0{,}8157p^2 + 28{,}394p + 558{,}27 \tag{15}$$

Analyzing equations (11÷15), it can be observed that a change in geometrical dimensions and type of hydraulic line (rigid to flexible with a specific number of braids) affects the change in the pressure pulsation amplitude (damping or strengthening of the pulsation amplitude).

In the presented model, the frictional resistance resulting from the viscosity of the working medium was taken into account by adopting the viscosity function Ψ_z, described by the equation [14]:

$$\psi_z = \varepsilon + j \cdot \delta \tag{16}$$

where:
Ψ_z – viscosity function,
j – imaginary unit,
ε – pressure wave damping coefficient,
δ – coefficient related to the wave phase velocity.

Due to the low excitation frequencies resulting from the pulsation of the positive displacement pump capacity, the quasi-steady friction model was adopted to determine the frequency characteristics of the hydrostatic system [14, 19, 20]. The coefficients of the viscosity function ε and δ for the adopted model are presented by the following equations:

$$\varepsilon = \sqrt{0{,}5} \cdot \Omega \cdot \sqrt{-1 + \sqrt{1 + \left(\frac{R_0}{\Omega}\right)^2}} \tag{17}$$

$$\delta = \sqrt{0{,}5} \cdot \Omega \cdot \sqrt{1 + \sqrt{1 + \left(\frac{R_0}{\Omega}\right)^2}} \tag{18}$$

The constant resistance R_0 was determined by the Darcy - Weisbach equation [17]:

$$R_0 = \frac{\lambda \cdot Re \cdot \mu}{8 \cdot \pi \cdot R^4} \tag{19}$$

where:
R_0 – constant resistance [kg/m^5·s],
λ – dimensionless coefficient of linear friction losses, $\lambda = \frac{64}{Re}$,
μ – dynamic viscosity of the working fluid [Pa·s].

The dimensionless frequency Ω is given by the equation:

$$\Omega = \frac{\omega \cdot R^2}{\nu} \tag{20}$$

where:
Ω – frequency,
ν – kinematic viscosity of the working fluid [m^2/s].

On the basis of the presented mathematical model, the transmittance modulus $G_{q1/p1}$ was determined, which determines the impact of the capacity pulsation on the pressure pulsation in the pump discharge line:

$$G_{q_1/p_1} = \frac{h_{11} \cdot Z_k + h_{12}}{h_{21} \cdot Z_k + h_{22}} \tag{21}$$

Fig. 2 shows exemplary waveforms of $G_{q1/p1}$ transmittance modules for five different values of pressure wave velocity c_0.

Analyzing the obtained waveform, it can be seen that a change in the pressure wave velocity results in a change in the width of one waveform period, but does not affect the peak values (minimum and maximum). The greater the value of the pressure wave velocity c_0, the wider the period. For the frequency of 300 Hz (pulsation frequency of gear pumps with external gearing, equipped with a 12-tooth gear and driven by a motor with a rotational velocity of 1500 rpm), the highest damping occurs for the pressure wave velocity

c_0 = 800 m/s, and the lowest, respectively, for c_0 = 1200 m/s and c_0 = 600 m/s. Model tests clearly show that the damping of displacement pump capacity pulsation is affected not only by the hydraulic drive parameters (flow, pressure in the hydraulic line, fluid viscosity), but also by the hydraulic line parameters, especially the pressure wave velocity c_0. Thus, it can be concluded that different types of hydraulic lines (steel tubes, elastic hoses with different numbers of braids) can be characterized by different damping properties. The following part of the article presents the results of experimental research on the influence of selected hydraulic line parameters on the change in the pressure pulsation damping in the pump discharge line.

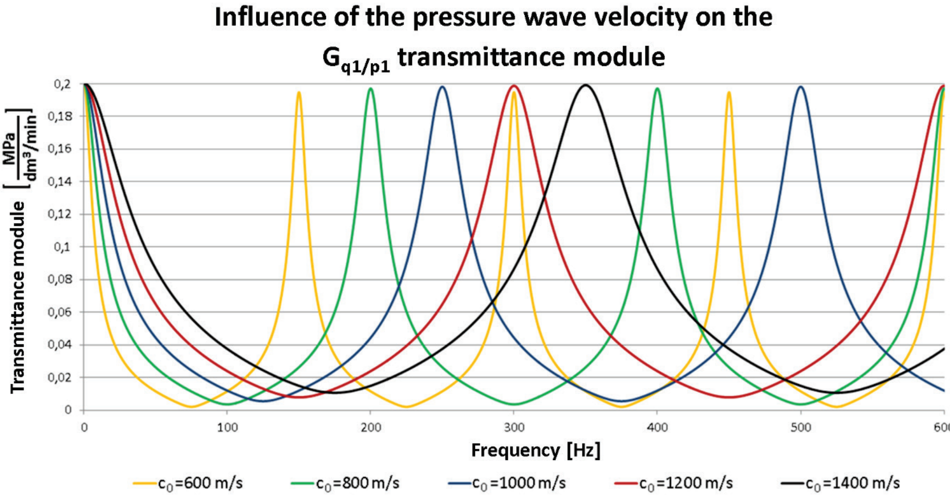

Fig. 2. Influence of the pressure wave velocity c_0 in the hydraulic line on the course of the $G_{q1/p1}$ transmittance module

3. The test stand

In order to conduct experimental research, a dedicated test stand was created. The test stand (Fig. 3) was built with an external gear pump (equipped with 12-tooth gears) of the Parker Hannifin PGP511 series, with a specific capacity of q_p = 8 cm^3/rev, driven by an asynchronous AC motor with a power of P = 5.5 kW and a rotational velocity of n = 1450 rpm (at nominal torque), a 9N600S throttle valve (four nominal pressure settings p), a pressure limiting valve set to the opening pressure p_r = 25 MPa, tested hydraulic lines with variable parameters (hose type, hydraulic diameter, number of braids), and a fixed length of L = 2.2 m.

Between the tested hydraulic line, a measuring system was placed in the form of two HYDAC HDA 4748-H-0250 pressure transducers. Archiving of the measurement results was carried out using a HYDAC HMG 3010 diagnostic recorder . The sampling rate of the signals recorded during the experiment was f_s = 10 kHz. The pressure transducer used in the system, according to the manufacturer's declaration, has a response step of t_t = 0.5 ms. Assuming that the pressure transducer is similar to the first-order inertial element, the authors of the research assumed that its time constant is three times shorter, which results in a lower pass frequency than the pressure transducer f_{pt} = 950Hz. For this reason, a ten times higher sampling frequency of pressure signals is sufficient to obtain oversampling. The HDA 4748-H-0250 transducer was also used as a low-pass filter.

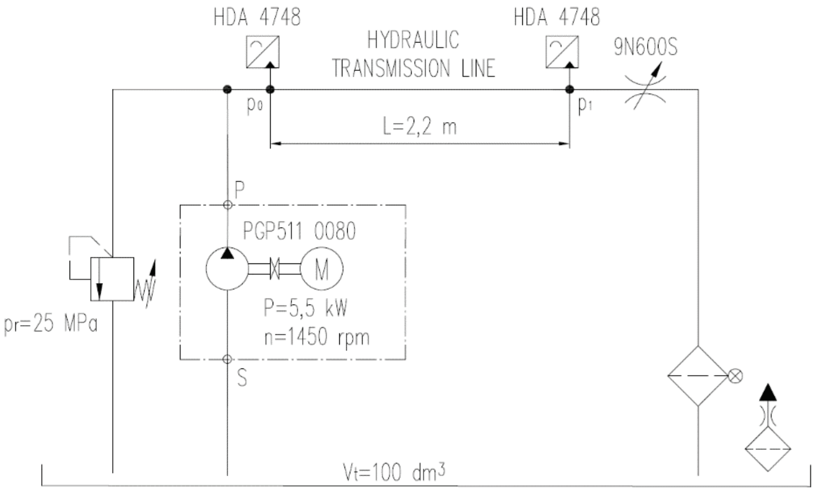

Fig. 3. Simplified hydraulic diagram of the test stand

4. Determining the research plan and variable parameters of the experiment

The course of the pressure pulsation in the delivery line is affected not only by the displacement unit capacity pulsation, but also by the hydraulic line parameters (hydraulic and geometric) and the working fluid parameters. Some research [14, 19, 20] has shown that the pressure wave velocity c_0, associated primarily with the line stiffness, and the flow diameter ϕd_h of the hydraulic line have a significant impact on the change in the line damping parameters.

The conducted research focused on the experimental analysis of the impact of hydraulic line stiffness (related to the compressibility of the working fluid and deformation of the hydraulic line walls) and the flow diameter ϕd_h on the change in the damping properties of the pressure pulsation line caused by displacement unit capacity pulsation. The hydraulic stiffness was adjusted by changing the type of hydraulic line (steel tubes, elastic hoses with different numbers of braids). Based on the equations described in [14], related to the determination of the pressure wave velocity c_0, it was found that the highest hydraulic stiffness is characteristic of rigid lines (steel tubes). The stiffness of elastic hoses depends primarily on the number and type of braids. According to [14, 19, 20], with increases in the number of braids, the pipe stiffness increases, and thus the pressure wave velocity c_0 increases.

The tests were carried out for three flow diameters, and thus three different Reynolds numbers and three C_0 capacitance values of the line. The analyses were carried out for four settings of the nominal pressure p in the discharge line, at a constant working fluid temperature $t_c = 40 \pm 2°C$.

According to equation (1), for the assumed test stand parameters, the discharge line can be treated as hydraulic long transmission line from the line length $L = 1.2$ m (for the assumed $c_0 = 1400$ m/s). During the experiments, lines with a fixed length of $L = 2.2$ m were used.

Table 1 presents the experiment plan for testing the influence of selected hydraulic line parameters on pressure pulsation damping.

Table 1

Plan of the experiment to study the influence of selected hydraulic line parameters on pressure pulsation damping

No.	Line type	Hydraulic line designation	Hydraulic diameter φd_h [mm]	Number of braids / thickness of the wall	Line volume V [cm^3]	Fluid flow velocity v [m/s]	Reynolds number for ν=46mm^2/s
1	Steel tube d = 10 / 1.5mm (steel E235N)	TL6_R1.5	6	1.5 mm	0.195	7.08	923
2	Parker Elite 492 WP 22.5MPa DN6 (1/4”) 1SC	TL6_1SC	6	1			
3	Fluidconnecto WP 33MPa DN6 (1/4”) 2SN	TL6_2SN	6	2			
4	Steel tube d = 14 / 2mm (steel E235N)	TL10_R2	10	2 mm	0.542	2.55	554
5	Parker Elite 492 WP 22.5MPa DN10 (3/8”) 1SC	TL10_1SC	10	1			
6	Fluidconnecto WP 28MPa DN10 (3/8”) 2SN	TL10_2SN	10	2			

Table 1 cont.

No.	Line type	Hydraulic line designation	Hydraulic diameter φd_h [mm]	Number of braids / thickness of the wall	Line volume V [cm^3]	Fluid flow velocity v [m/s]	Reynolds number for ν=46mm^2/s
7	Manuli Rockmaster WP 45MPa DN10 (3/8”) 4SP	TL10_4SP	10	4	0.542	2.55	554
8	Manuli Tractor WP 16MPa DN12 (1/2”) 1SN	TL12_1SN	12	1	0.781	1.77	462
9	Manuli Tractor WP 28MPa DN12 (1/2”) 2SN	TL12_2SN	12	2			

In order to assess the impact of selected hydraulic line parameters, as listed in Table 1, on the change in the pressure pulsation course, an analysis of the obtained measurement results in the time and frequency domains was carried out. The time-domain analysis aimed to determine changes in the value of the peak-to-peak pressure Δp between the beginning and the end of the hydraulic line. The frequency analysis made it possible to determine the damping (or amplification) value of the pressure pulsation amplitude caused by the displacement pump capacity pulsation at the beginning and end of the hydraulic line.

To compare the test results, the T line damping coefficients were introduced, determined in accordance with the following equation:

$$T = 20 \cdot log\left(\frac{A_1}{A_0}\right) \tag{22}$$

where:
T – hydraulic line damping [dB],
A_0 – parameter at the beginning of the discharge line [MPa],
A_1 – parameter at the end of the discharge line [MPa].

In the case of the time-domain analysis, the parameters A_0 and A_1 determine the value of the peak-to-peak pressure at the beginning Δp_0 and end Δp_1 of the hydraulic line, respectively. When analysing the frequency domain, the parameters A_0 and A_1 correspond to the values of the pressure pulsation amplitudes at the beginning p_{p0} and end p_{p1} of the line. A negative value of coefficient T indicates damping, while positive values indicate amplification of hydraulic line pulsation.

5. Determining the range of resonance frequencies of the tested hydraulic lines

In hydraulic lines, resonance may occur due to excitation. Resonance in the flow lines leads to changes in the hydraulic line parameters (including damping), which is why this subsection focuses on determining the range of possible resonance frequencies of the tested flow lines.

Hydraulic lines can be approximately treated as cylinders closed on one side with a throttle valve. For such a line, the resonance frequencies f_{rez} can be determined from the equation [14]:

$$f_{rez} = \frac{(2 \cdot n - 1) \cdot v}{4 \cdot L} [Hz] \tag{23}$$

where:

f_{rez} – resonance frequency of the hydraulic line [Hz],
n – number of resonant mode.

Assuming pressure wave velocity values in the range of c_0 = 600 ÷ 1400 m/s, the first resonance frequency of the line occurs in the range of f_{rez} = 330 ÷ 770 Hz. As the pump unit applied in the test stand is characterized by efficiency pulsation with the frequency f_p = 292 ÷ 300 Hz, the risk of resonance extinction in the tested hydraulic lines was excluded. In the case of the described phenomenon occurring in the hydraulic line, a much greater pulsation amplitude should be observed at the beginning of the line p_0 and strong damping at the end p_1.

6. Results of hydraulic line damping tests in the time domain

Fig. 4 shows a graphical comparison of the obtained values of the damping coefficient T during analysis in the time domain.

Analyzing the obtained values of the coefficient T, it was found that hydraulic lines with a diameter of φd_h = 6 mm are characterized by the greatest damping. This phenomenon may be caused by energy dissipation (hydraulic energy losses), due to a relatively high pressure drop caused by the high fluid flow velocity (v = 7.08 m/s), exceeding the recommended values. A significant increase in the amplification of the peak-to-peak pressure Δp was observed in the range of higher pressures only in the case of a single-braid hose (TL6_1SC test). This effect is probably caused by the influence of the hydraulic line parameters on the course of the analysed pressure pulsation, for the pressure wave velocity c_0 occurring in the line.

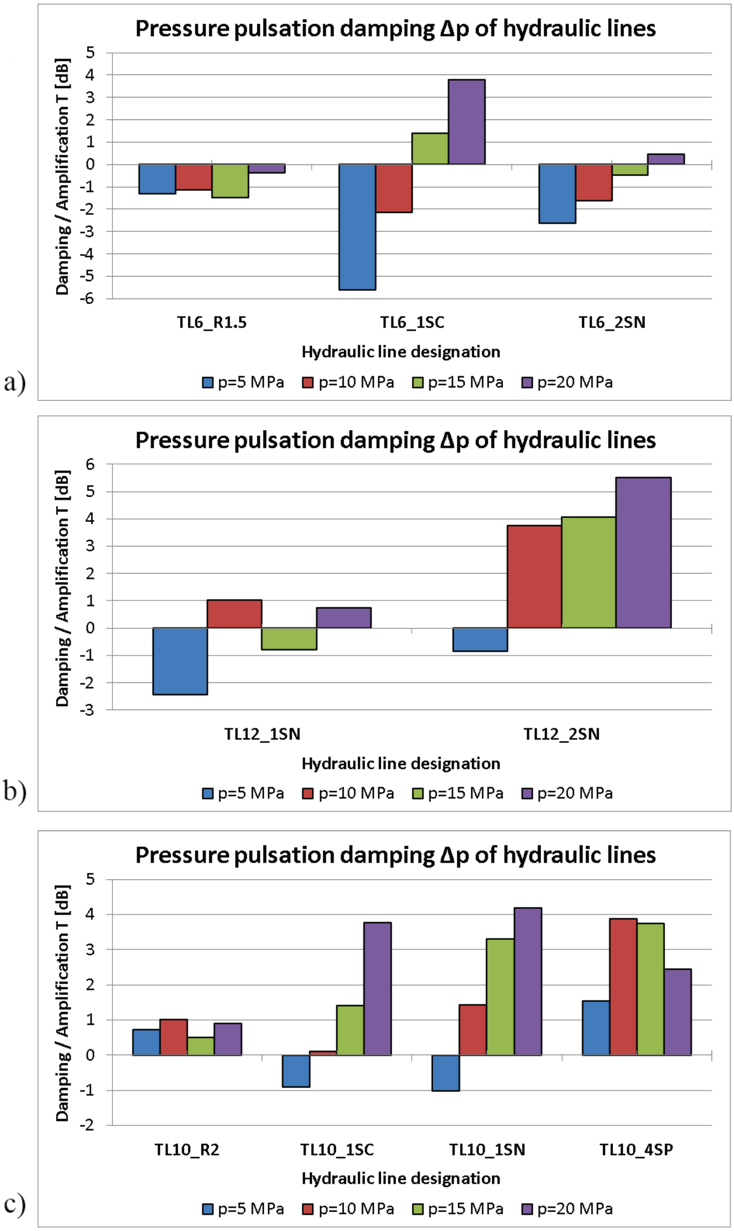

Fig. 4. Comparison of the obtained damping coefficients T for analysis in the time domain

In the case of elastic hoses with flow diameters φd_h = 10 mm and φd_h = 12 mm, an increase in the pulsation waveform was recorded in most experiments. A clear influence of the nominal pressure p value on the coefficient T was also observed. With the increase in the pressure p, the value of the peak-to-peak pressure Δp_1 at the end of the hydraulic line increased in relation to the value of the pressure Δp_0 at the beginning of the line. The influence of stiffening the elastic hoses on the change in the T line damping module was also recorded. At low pressure values (p = 5 MPa) in elastic hoses, there is always damping due to the greater susceptibility of elastomer ducts to elastic deformation compared to steel tubes.

In the case of steel tubes, the value of the coefficient T, regardless of the value of the nominal pressure p, is at a similar level. This phenomenon is caused by the high resistance to deformation of the inner surface of steel tubes, which results in a lack of tube stiffening.

Fig. 5 presents the course of pressure pulsation at the beginning and end of the hydraulic line for the elastic hose characterized by the highest reinforcement (TL12_2SN test).

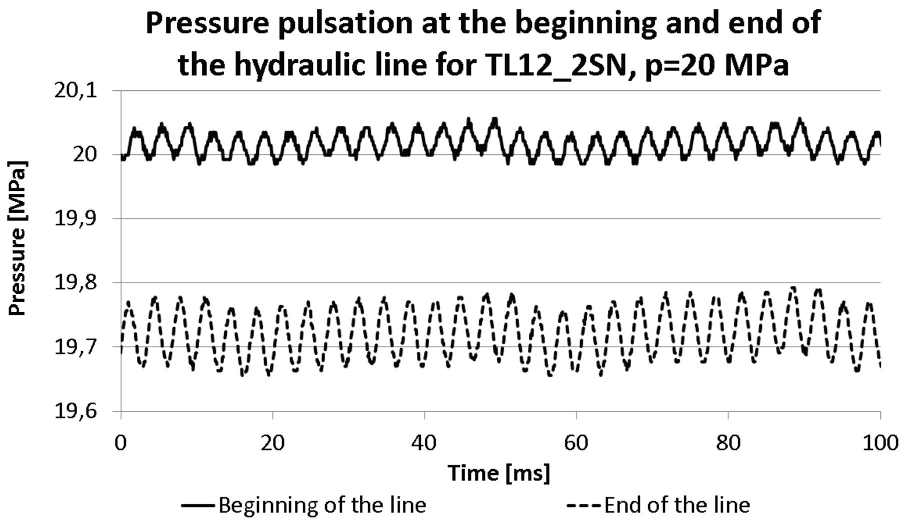

Fig. 5. Course of pressure pulsation at the beginning and end of the hydraulic line for the elastic hose characterized by the highest amplification (TL12_2SN test)

Fig. 5 clearly shows an almost two-fold increase in the pressure pulsation value, which can lead to increased vibrations and noise, accelerated wear of the drive components installed at the end of the hydraulic line, and also reduce the receiver's positioning accuracy.

7. Results of hydraulic line damping tests in the frequency domain

The research carried out in the above subsection focused on analysing the influence of the stiffness and diameter φd_h of the hydraulic line on the value of the peak-to-peak pressure Δp. Δp is not only the work of the displacement pump's elements, but also the component resulting from the drive shaft rotational velocity. Therefore, this subsection focuses on studying the impact of the pressure wave velocity c_0 and the internal diameter of the line φd_h on the damping of the pressure pulsation amplitude p_p for the frequency f_p related to the positive displacement operation of the pump.

During the experiments, the pressure pulsation frequency caused by the displacement pump elements was:

- f_p = 298 Hz for p = 5 MPa,
- f_p = 296 Hz for p = 10 MPa,
- f_p = 294 Hz for p= 15 MPa,
- f_p = 292 Hz for p = 20MPa.

For FFT data processing, i_s = 8192 samples were analysed using the rectangular window function. This window type was used due to its best spectral resolution and, thus, the ability to distinguish between two close frequencies of pressure pulsations. The frequency analysis results were averaged based on three FFT windows.

Fig. 6 shows the obtained values of the damping coefficient T for the pressure pulsation frequency f_p, caused by the displacement unit capacity pulsation.

When interpreting the obtained measurement results, it was noticed that when the flow diameter ϕd_h = 6 mm, the damping of the pressure pulsation amplitude pp occurs almost in the entire test range. This phenomenon (as in the case of the time-domain analysis of damping) is most likely caused by the dissipation of pressure energy due to a large pressure drop caused by fluid flow resistance. In the case of the tested elastic hoses, a clear increase in reinforcement was observed with the increase in the nominal pressure p. This phenomenon is most likely caused by the stiffening effect of the elastic hose.

For the flow diameters ϕd_h = 10 mm and ϕd_h = 12 mm, the amplification of the pressure pulsation amplitude pp for the frequency f_p was recorded for all experiments. During the analysis of the recorded measurement results for elastic hoses, a significant influence of the pressure value in the hydraulic line p on the change in the damping coefficient T was noticed. With the increase in the pressure p, the amplification value increases, and thus the amplitude p_p increases.

In the case of rigid lines, the values of the coefficient T for variable values of the nominal pressure p are at a similar level. Thus, a change in the value of pressure p in a line made of rigid tubes will not significantly affect the amplitude of pulsations p_p.

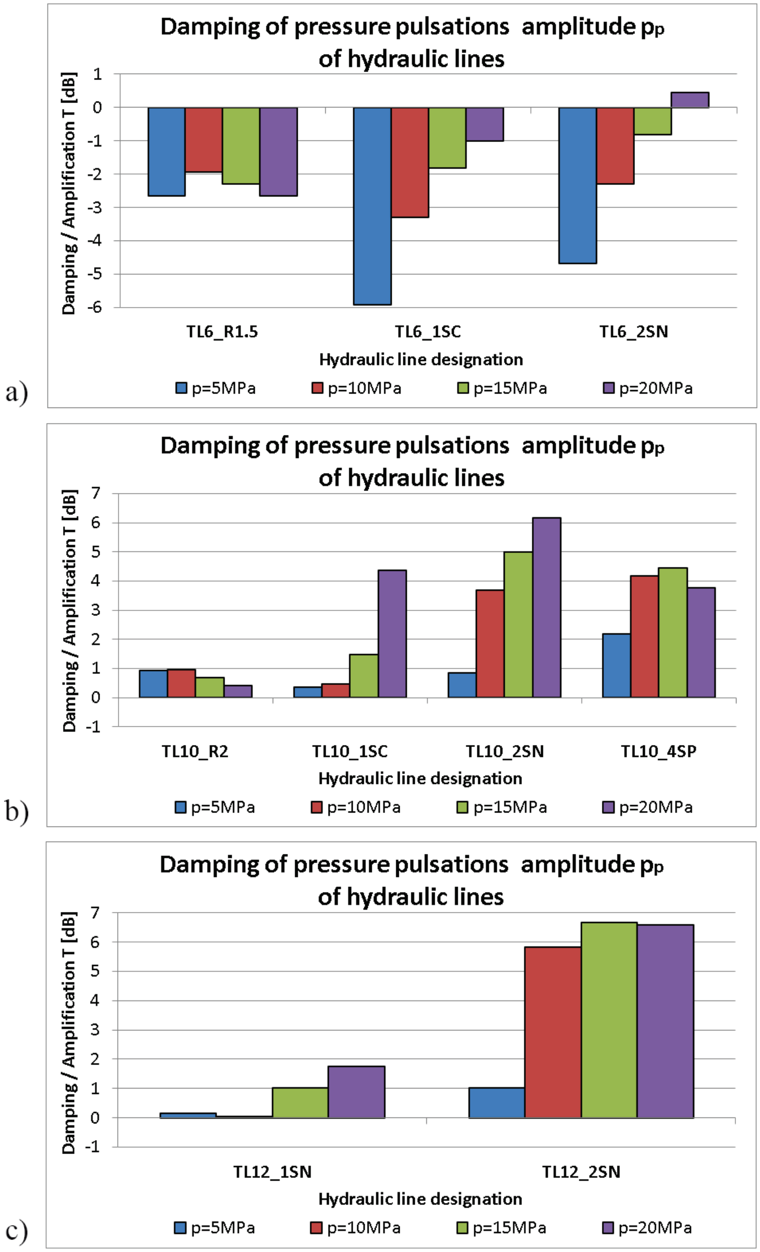

Fig. 6. Comparison of the value of the damping coefficient T for the frequency of pressure pulsations f_p, caused by the displacement unit capacity pulsation

Fig. 7 shows a comparison of the results of FFT analysis of hydraulic lines with the highest and lowest damping coefficient for the nominal pressure p = 20 MPa (TL6_2SN and TL12_2SN tests).

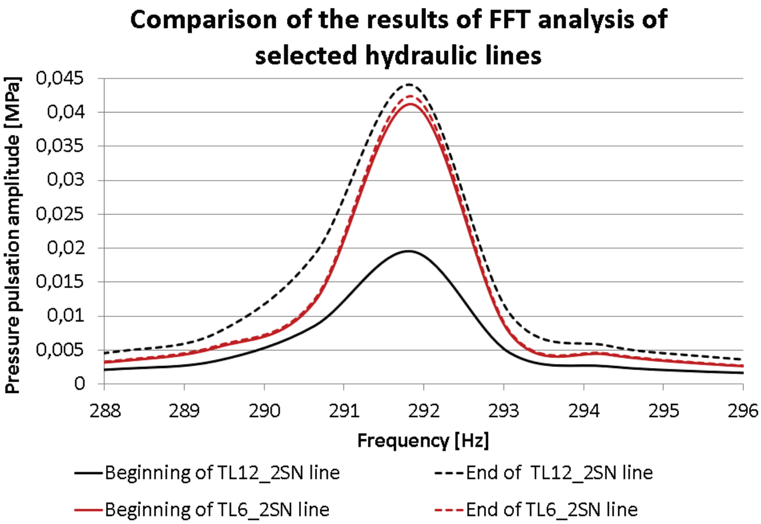

Fig. 7. Comparison of the results of FFT analysis of selected hydraulic lines (Tests TL6_2SN and TL12_2SN).

The analyses carried out in the frequency domain confirmed the influence of the hydraulic line on the hydraulic line damping properties, especially the parameters related to the internal stiffness of the elastic hoses. However, the conducted research failed to obtain satisfactory results. In the case of hydraulic diameters above ϕd_h = 6 mm, the phenomenon of pulsation damping was not achieved. In each case, the phenomenon of amplification of the p_p amplitude was recorded. However, for a rigid tube, the amplification has the lowest value.

8. Conclusions

Based on the obtained test results, it can be concluded that the type and flow diameter φd_h of the hydraulic line have a significant impact on pressure pulsation damping. In addition, in the case of elastic hoses, a stiffening effect was observed at higher pressure p values, which resulted in an increase in line amplification. The stiffening phenomenon is associated with a decrease in the susceptibility of the elastic hose due to the pressure effect, which results in a change in volume resulting from the deformation of the inner layer of the elastic hose made of elastomer. Therefore, the variable damping properties of lines made of elastic hoses should be taken into account, along with the change in the pressure parameters of the hydraulic drive.

It should be remembered that the tests carried out did not take into account the influence of the hydraulic line length L and changes in the pulsation frequency of the pump efficiency f_p.

The frequency-domain analyses also confirmed the relationship that the velocity of the pressure wave c_0 in an elastic hose depends on the pressure value and the number of braids [17]. An increase in the line pressure results in increased velocity of the pressure wave c_0, which affects the change in the line damping properties.

The results obtained to some extent confirmed commonly held beliefs. Indeed, the discharge pipe dampens pressure pulsations to some extent (assuming appropriate parameter selection), generated by the pulsating nature of the displacement pump operation. However, replacing a rigid line with an elastic hose of the same length L and flow diameter φd_h without carrying out numerical analyses or experimental tests may be counterproductive. Instead of having a damping effect, the pressure pulsation phenomenon will be amplified. Therefore, in the case of problems with the selection of optimal hydraulic line parameters, it is recommended to use active dampers, as widely described in the literature [6, 10, 11, 12, 13, 14, 15, 16, 20].

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