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ANALYSIS OF IMPACT OF ROTATION SPEED CHANGES ON PRESSURE PULSATION OF POSITIVE DISPLACEMENT PUMP

Analiza wpływu zmian prędkości obrotowej pompy wyporowej na przebieg pulsacji ciśnienia

Abstract: *The paper presents the results of experimental research on the impact of changes in the rotation speed of a hydraulic external gear pump on pressure pulsation in the discharge line for a constant value of pressure. The analyses were performed in the time domain (analysis of peak-to-peak pressure) and frequency domain (analysis of changes in frequency and pulsation amplitude). The results of experimental studies have shown that the change of the pump's rotation speed influences changes in the course of pressure pulsation (peak-to-peak pressure and amplitude), in some cases significantly increasing the parameters mentioned. The obtained test results can be used to develop recommendations for designing hydraulic and hydrotronic systems equipped with a variable-speed pump drive.*
Keywords: hydrostatic drives, pressure pulsation, measurement, FFT analysis, inverter drive

Streszczenie: *W artykule przedstawiono wyniki badań doświadczalnych wpływu zmian prędkości obrotowej pompy zębatej na zmianę przebiegu pulsacji ciśnienia w linii tłocznej dla stałej wartości ciśnienia tłoczenia. Analizy przeprowadzono w dziedzinie czasu (analiza zmian wartości ciśnienia międzyszczytowego) oraz dziedzinie częstotliwości (analiza zmian wartości częstotliwości i amplitudy pulsacji). Otrzymane wyniki badań eksperymentalnych jednoznacznie wykazały, że zmiana prędkości obrotowej pompy wpływa na zmianę przebiegu pulsacji ciśnienia (zarówno na wartość ciśnienia międzyszczytowego, jak i amplitudę), w niektórych przypadkach znacznie zwiększając wymienione parametry. Otrzymane wyniki badań, mogą zostać wykorzystane do opracowania zaleceń dotyczących projektowania układów hydraulicznych i hydrotronicznych, wyposażonych w napęd zmiennoobrotowy pompy.*

Słowa kluczowe: napędy hydrostatyczne, pulsacja ciśnienia, pomiary, analiza FFT, przemiennik częstotliwości

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1. Introduction

Modern power units for hydrostatic drives tend to replace traditional solutions with a variable displacement pump, a mechatronic counterpart with similar functionality. A hydraulic power unit based on this concept often includes a fixed displacement pump driven by an electric motor with infinitely variable speed control. Such a drive of the positive displacement pump allows controlling the pump operation using electric signals to obtain the required parameters of the hydraulic power unit operation, particularly the flow rate generated or the pressure. Thus, a hydrotronic power supply module with volume control is created significantly more efficiently than hydraulic systems with throttle control. This type of control allows not only to change the rotational speed of an electric motor smoothly but also (using vector control) to compensate for the motor rotor slip as the load increases [1-4]. This ensures no excessive drop in flow rate as discharge pressure increases. In addition, such a solution makes it possible to compensate for pump leakage by increasing the preset pump speed depending on the current discharge pressure. However, such a drive system has several disadvantages related to the dynamic phenomena of the flow generator (of the positive displacement pump) and those occurring in the hydraulic line [5–7]. These phenomena are related to the pressure pulsation caused by the pump output pulsation. The pressure pulsation parameters (frequency and amplitude) change as the pump shaft rotational speed changes. Pressure pulsation has an impact on reducing the accuracy of receiver positioning (especially in the case of precise automatic receiver position control systems), accelerated wear of drive hydraulic components (increasing contamination level of hydraulic fluid and leakages of hydraulic components, especially of poppet valves), increased drives vibrations and noise emissions [5, 8–10]. The variable nature of the aforementioned phenomenon may compound the above factors. Additionally, suppressing varying frequency and amplitude pressure pulsations with passive or active dampers is problematic.

This paper focuses on the analysis of pressure pulsation variations in a system with a variable speed pump drive. An analysis of the effect of rotational speed not only on the change of frequency but especially on the amplitude of pressure pulsations in the discharge line of the displacement unit was carried out. Time analysis was performed to determine the peak-to-peak discharge pressure, which affects the fatigue wear of hydraulic system components and generates increased vibration and noise. Frequency analysis was performed to determine the pressure pulsation amplitude caused by the pump output pulsation [5, 7, 11–13].

2. Relationship between flow and pressure pulsation

Pressure pulsation in pumps used in hydrostatic drives results from the periodically varying flow rate of the working fluid due to the cyclic nature of the working elements of the displacement units [5, 11–16]. The frequency of output and pressure pulsations is directly related to the number of displacement elements and the rotation frequency of the drive shaft:

$$f_{pw} = \frac{i \cdot n}{60} \ [Hz] \tag{1}$$

where:
f_{pw} - fundamental frequency of pump output pulsation [Hz],
i - the number of teeth in the gear wheel,
n - pump shaft rotational speed [rpm].

Pressure pulsation is primarily caused by the positive displacement of the pump output pulsation. It should be remembered, however, that the processes associated with the flow of the working fluid through the tubing (flexible and rigid) in high-pressure hydraulic systems, resulting from forced or accidental activation of the drive system elements, have an impact on their dynamic characteristics and operating properties. The nature of the non-stationary processes depends on the selected physical properties of the working fluid, the elasticity of the tubing and its geometrical dimensions [5, 7].
The above make up the following hydraulic line parameters:

- resistance R_0 taking into account the friction between the fluid particles,
- inertance M_0 taking into account the inertia of the fluid,
- capacitance C_0 taking into account the fluid compressibility β_0 and the elasticity of the line tubing material.

The combined effects of resistance, inertance and capacitance under dynamic operating conditions cause distortions and delays in the flow tubes of hydraulic power systems, thereby altering their dynamic properties.

In the literature related to the modelling of the hydraulic long line [5–7, 17, 18], the two most commonly used methods of describing transient or quasi-steady waveforms are presented:

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

The frequency method [5, 7] is popular for correlating output pulsations with pressure pulsations, which consists in determining the characteristics of the transmittance modulus $G_{p1/q1}$ combining the influence of flow pulsations of pump q_1 and pressure pulsations p_1 in the pump discharge line.

In the frequency method, the relationship between input and output ports is described by a matrix equation:

$$\begin{bmatrix} p_{in} \\ q_{in} \end{bmatrix} = \mathbf{H}(j\omega) \begin{bmatrix} p_{out} \\ q_{out} \end{bmatrix} \tag{2}$$

In an extended version, the matrix equation of the two-port hydraulic line is described as follows:

$$\begin{bmatrix} p_{in} \\ q_{in} \end{bmatrix} = \begin{bmatrix} h_{11} & h_{12} \\ h_{21} & h_{22} \end{bmatrix} \begin{bmatrix} p_{out} \\ q_{out} \end{bmatrix} \tag{3}$$

In accordance with [5], elements of the transfer function H(jω) can be described as:

$$h_{11} = \cosh(T\psi_Z j\omega)$$

$$h_{12} = Z_c \psi_Z \sinh(T\psi_Z j\omega)$$

$$h_{21} = \frac{1}{Z_c \psi_Z} \sinh(T\psi_Z j\omega)$$

$$h_{22} = \cosh(T\psi_Z j\omega)$$

hereby:

$$T = \frac{L}{c_0} \tag{4}$$

$$Z_c = \frac{\rho_0 c_0}{\pi R^2} \tag{5}$$

where:

ψ_Z - viscosity function [5],

T - time constant,

Z_c - the characteristic impedance of the pipe (hose) with a radius R and length L,

c_0 - the speed of sound in the liquid inside the pipe (hose),
ρ_0 - fluid density.

Based on equations (2÷5), the transmittance modulus $G_{q1/p1}$ was defined as:

$$G_{p_1/q_1} = \frac{h_{11} \cdot Z_k + h_{12}}{h_{21} \cdot Z_k + h_{22}} \tag{6}$$

Figure 1 shows an example of the transmittance modulus $G_{p1/q1}$ for selected hydraulic line parameters.

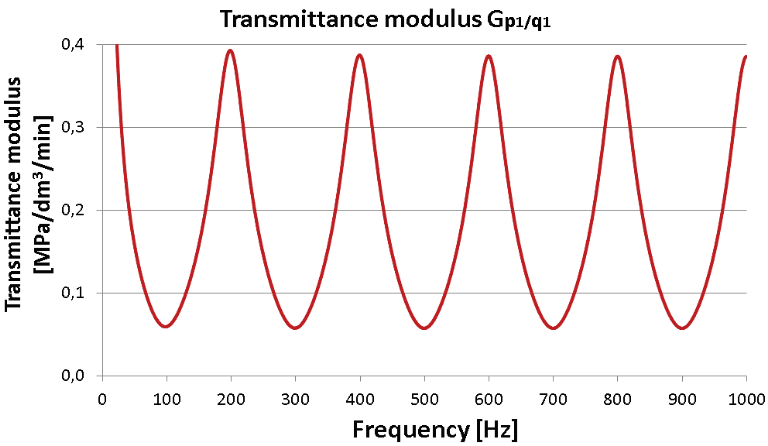

Fig. 1. Example of the characteristic of the transmittance modulus $G_{p1/q1}$ of a hydraulic line

Change in the pulsation frequency due to the speed change of the drive shaft affects the value of the transmittance modulus, and, thus, the amplitude of the pressure pulsation. The larger the amplitude of the pressure pulsations, especially at lower frequencies, the more unfavourable the dynamic operating conditions of the system. The optimal solution is to get the lowest transmittance value (highest damping).

3. Measuring stand and experiment plan

The test stand consisted of a Parker PGP511 0080 gear pump with unit capacity $q = 8$ cm^3/rev, driven by a $P = 5.5$ kW asynchronous motor powered by a Parker AC10 frequency converter. Pumps PGP511 series characterized 12 displacement elements (teeth on the gear wheels) per one shaft revolution. The maximum pressure in the discharge line was limited to $p_z = 21.5$ MPa with a Parker VS 210 valve. The pressure was adjusted by selecting the hydraulic resistance of the discharge line with a Parker 9N throttle valve.

Analyses were performed with HLP46 oil, according to DIN 51524, at T = 40 ± 2 °C. The setting of the filter bypass valve was p_{bypass} = 0,25 MPa, however, the pressure drop across the filter did not exceed a maximum of p_{filter} = 0,05 MPa. Hydraulic parameters were recorded using pressure transducer HYDAC HDA 4748 0250 [19], flow transducer HYDAC EVS 3108 0020 [20], rotational speed transducer SICK WL18-3P130 and temperature transducer HYDAC ETS 4148, coupled to an HYDAC HMG3010 recorder [21]. Figure 2 shows a simplified diagram of the test stand.

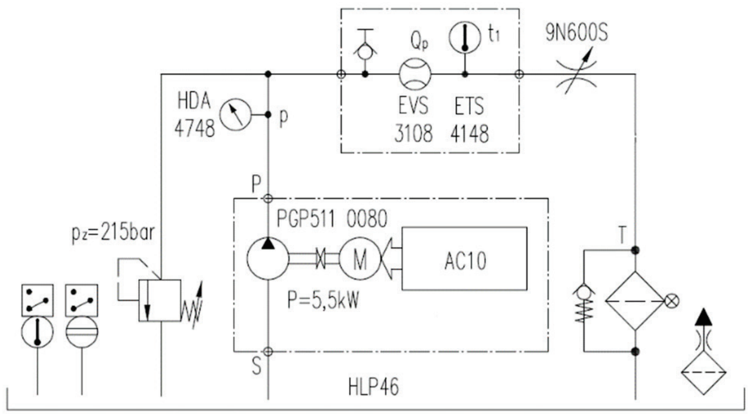

Fig. 2. Simplified diagram of the measuring stand

In the experiment, authors analysed pressure pulsation waveforms for selected rotational speeds of the drive shaft for a constant pressure of 10 MPa. Table 1 shows the developed experimental plan.

Table 1

Developed experimental plan

No	Test designation	Shaft speed n [rpm]	The theoretical frequency of shaft rotation f_{nt} [Hz]	Theoretical frequency of pressure and flow pulsation f_{pt} [Hz]
1	PF_50	1500	25	300
2	PF_45	1350	22.5	270
3	PF_40	1200	20	240
4	PF_35	1050	17.5	210
5	PF_30	900	15	180
6	PF_25	750	12.5	150
7	PF_20	600	10	120

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 low pass frequency than the pressure transducer f_p = 950 Hz. For this reason, a ten times higher sampling frequency of pressure signals is sufficient to avoid frequency masking. The HDA 4748 transducer was also used as a low-pass filter.

The peak-to-peak pressure is based on a 100 ms time interval. For FFT data processing, 16384 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.

4. Results and discussion

Table 2 presents the selected results of the experimental studies.

Table 2

Selected experimental results

No	Test designation	Shaft speed n [rpm]	Time domain analysis			Frequency domain analysis			
			p_{min} [MPa]	p_{max} [MPa]	Δp [MPa]	f_n [Hz]	p_n [MPa]	f_p [Hz]	p_p [MPa]
1	PF_50	1500	9.92	10.08	0.16	25	0.006	304	0.054
2	PF_45	1350	9.97	10.17	0.18	22.5	0.013	270	0.044
3	PF_40	1200	9.91	10.14	0.35	20	0.012	240	0.094
4	PF_35	1050	9.88	10.24	0.36	16.5	0.017	210	1.019
5	PF_30	900	9.84	10.19	0.23	14.5	0.033	189	1.019
6	PF_25	750	9.96	10.14	0.2	12.8	0.026	149	0.054
7	PF_20	600	10.03	10.12	0.09	9.8	0.02	118	0.016

Figure 3 shows the pressure pulsations for the lowest and highest peak-to-peak pressure experiment (tests PF_20 and PF_35), analysed in time domain.

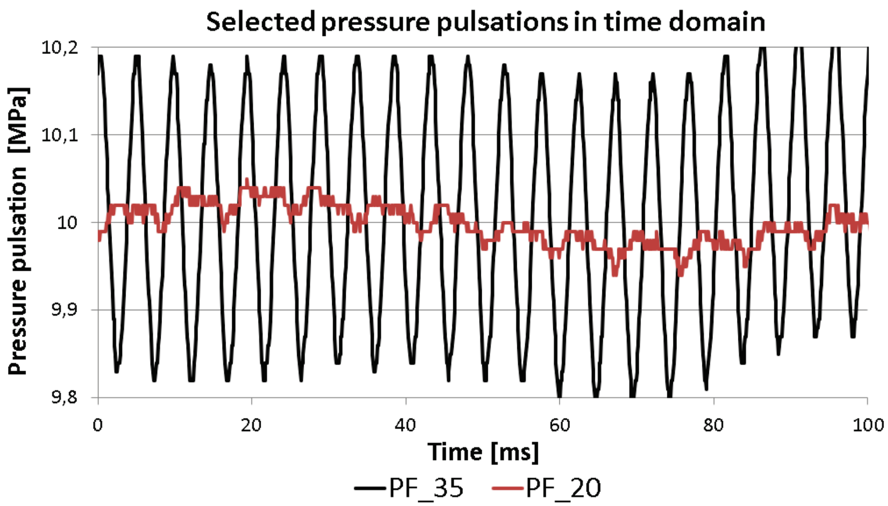

Fig. 3. Pressure pulsation in time domain for experiments PF_20 and PF_35

Analysing the recorded waveform, noted that the peak-to-peak pressure of the PF_20 experiment was almost four times lower than that of the PF_35. The peak-to-peak pressures for the indicated tests are $\Delta p = 0.09$ MPa and $\Delta p = 0.36$ MPa respectively

Figures 4 and 5 shows comparison of pressure pulsation results for experiments PF_20 and PF_30, analysed in frequency domain. The presented waveforms characterized by the lowest and highest p_p pressure amplitudes.

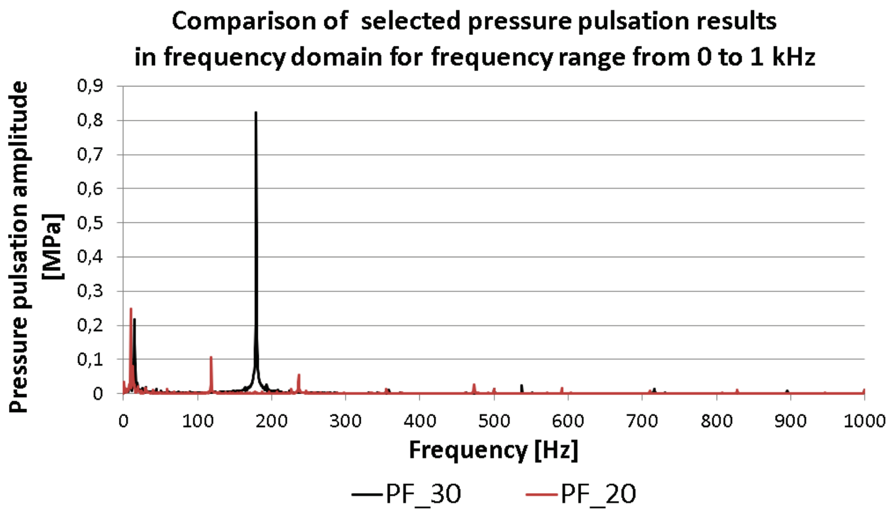

Fig. 4. Comparison of pressure pulsation results in frequency domain for PF_20 and PF_30 experiments for frequency range from 0 to 1 kHz

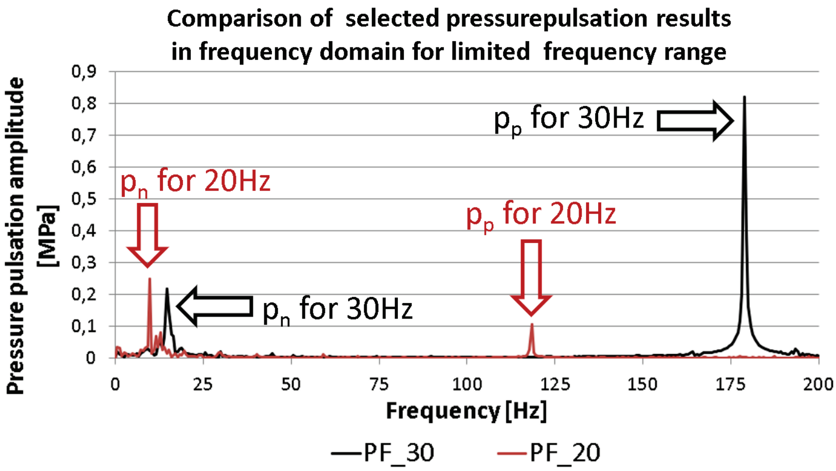

Fig. 5. Comparison of pressure pulsation results in frequency domain for PF_20 and PF_30 experiments for limited frequency range; p_n - pressure pulsation related with pump rotation speed; p_p - pressure pulsation related with pump flow pulsation.

Comparing the recorded waveforms in the frequency domain of experiments PF20 and PF_30, a difference of more than eight times in the amplitude of the pressure pulsation p_p, resulting from the pulsating operation of the displacement elements, was observed. The values of the pressure pulsations at low frequencies resulting from the shaft speed p_n were similar for both experiments.

Figures 6 and 7 present the summary results of the analysis of the effect of pump shaft speed on the peak-to-peak pressure and the amplitude of pressure pulsations.

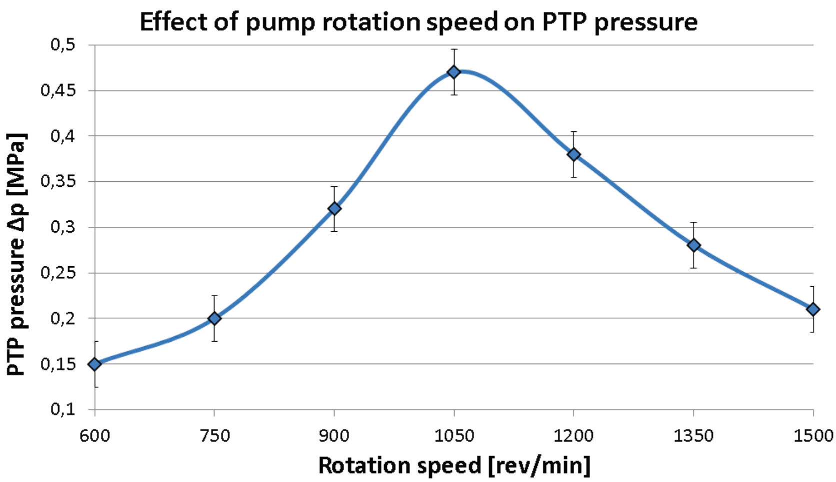

Fig. 6. Effect of pump shaft speed on peak-to-peak pressure

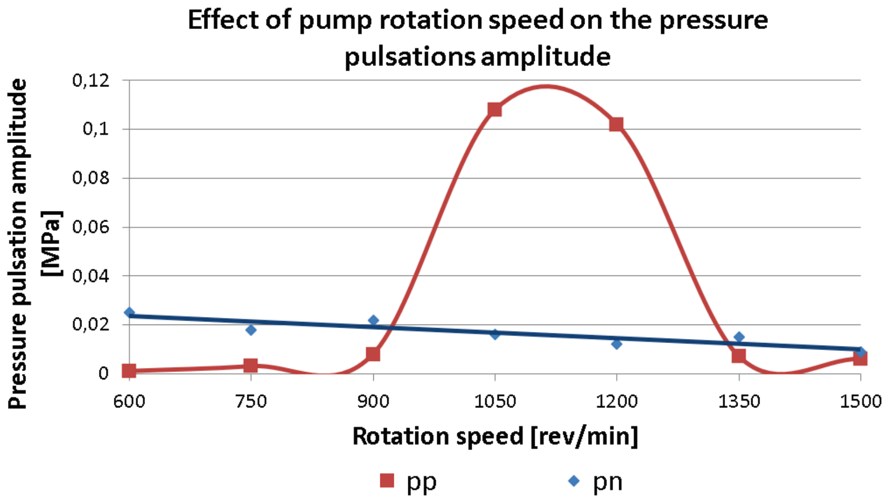

Fig. 7. Effect of pump shaft speed on the amplitude of discharge line pressure pulsations

By analysing the summary results of the measurements, it was observed that with a change in the pump drive shaft speed, there is a change in the peak-to-peak pressure. The highest pressure occurs at speeds between n = 900 and n = 1050 rpm. The lowest pressure was recorded for a shaft speed frequency of n = 600 rpm. A change in speed does not linearly affect the change in peak-to-peak pressure.

With the shaft speed increase to n = 900 rpm, the peak-to-peak pressure and the amplitude of the pressure pulsations p_n and p_p increase. The peak-to-peak pressure exceeds 3.5 bar, and the pressure amplitude resulting from the pump tooth engagement reaches 1 bar. Above n = 1050 rpm, the p_n and p_p amplitudes decrease. The reason for the recorded phenomenon is the effect of hydraulic line parameters. As the pump output pulsation frequency changes, the line damping parameters change, thus changing the pressure pulsation amplitude. The form of the resulting waveform is similar to the hydraulic line transmittance shown in Fig. 1. For the conducted tests, the most favourable conditions of the drive operation (in terms of pressure pulsation minimisation) are observed at the pump shaft rotational speed n = 600 rpm. However, it should be remembered that the pump's volumetric efficiency is low at such a low speed.

5. Conclusions and summary

Hydrostatic drives with a hydraulic power unit and a variable speed pump are often found in various industrial machines. Ease of electrical control of pump output and energy savings compared to hydrostatic drives with throttle control and relatively simple diagnostics are the primary advantages of this drive type. However, it should be remembered that the described drive has some negative features that may affect the correct operation of the hydraulic system and its durability and reliability. One of these features is the variable nature of pressure pulsations. Experimental studies showed that as the pump drive shaft speed changes, there is a significant change in the peak-to-peak pressure (time domain analyses) and the pulsation amplitude for frequency related to the pump output pulsation (frequency domain analyses). High values of the above parameters adversely affect the drive dynamics, accuracy and stability of the receiver positioning and lead to accelerated wear of the drive components.

The experiments conducted by the authors are preliminary. It is recommended that research be expanded to include analyses (model and experimental) of the influence of additional factors (e.g. internal leakage of positive displacement pumps) on changing the pressure pulsation pattern.

Below there are some tips for developing design recommendations that take into account pressure pulsation variation in the discharge line of variable speed pumps:

- eigenfrequencies of pressure valves mounted on the discharge line should be outside of the range of pressure pulsation generated by the pump,
- based on model analysis should be determined the amplitude – frequency characteristics of hydraulic discharge lines to select a pressure pulsation damping system properly,
- pressure pulsation value (caused by flow pulsation) of assumed pumps should be minimalized by their proper construction (e.g. gear pumps with internal gearing or constant displacement vane pumps),
- the pressure transducers used in the prototype of the designed hydraulic system must have a high cutoff frequency value and the possibility of deaeration of the measuring chambers.

During acceptance tests of the new hydraulic system, measuring pressure pulsation, vibrations level, and noise of hydraulic power pack with variable speed pump in the whole range of its rotation speed.

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