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RESEARCH

Pressure Surge Analysis of a Test Bench for Biodegradable Hydraulic Oils

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A B S T R A C T

A test bench for biodegradable hydraulic oils generates pressure surges by cyclic pressure loading. The parameters of these pressure surges (maximum pressure, amplitude, frequency etc.) are defined by the characteristics of the pressure valve and pipe line system. The response of the hydraulic system was recorded and studied in the case of four test bench modifications by comparing the maximum pressure, maximum pressure increase and the opening time of the pressure valve. The test bench with steel pipe between the hydraulic pump and pressure valve shows the highest maximum pressure increase (4.28 MPa). The opening times of pressure valve were measured for the evaluation of dynamic stability of the test bench. In case of all test bench modifications the opening time of pressure valve did not exceed the calculated limit value. The experiments were also aimed at the response of hydraulic system to three different frequencies (1.5 Hz, 2.5 Hz and 5 Hz) of cyclic pressure loading. Increasing the frequency only minimally increases the maximum pressure during the pressure surge. On the other hand, it increases maximum amplitude after the maximum pressure depending up to frequency of cyclic pressure loading. Fast Fourier Transform (FFT) confirms the facts mention above.

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1. INTRODUCTION

Environmental protection is an important and trending topic of the last decade, and it becomes a central problem in the established trend of economic development. The growth of economy also results to the intensification of agricultural production. One of the possibilities of eco-friendly behaviour is the utilization of ecological oils in the field of agricultural machines [1-3].

Currently, the possibility of replacing an increasing number of lubricants by ecological equivalent comes to the fore. The use of environmentally friendly lubricants is one of the ways to eliminate the pollution of soil, water and groundwater with operation fillings of agricultural and construction mechanisms [4]. On the present, the researchers pay attention to the application of biodegradable lubricating oils in hydraulic system of machines, as in [5-7].

Application of biodegradable lubricants requires a development of such a testing method regarding the operation conditions of the actual machine. Biodegradable oils are tested before their application to the machines in a variety of ways [8,9]. The operation conditions namely machine load, operation temperature, ambient environment pollution but also technical condition of the machine affect the selection of optimal lubricant type at most. Actually, before using a lubricant, its properties have to be revealed in detail, to verify if it fulfills the required functions in machine during described time interval. The best way to test hydraulic oils is to use a test bench that consists of the same hydraulic circuits components as the actual machine. This contribution presents the pressure surge analysis in the hydraulic circuit of such a test bench for research of biodegradable hydraulic oils properties.

Due to the cyclic loading in the testing circuit, understanding water hammer is important to prevent the excessive pressure build-up in pipelines. Many researchers have studied this phenomenon, drawing effective solutions through the time- and frequency-domain approaches [10].

1.1 Pressure surge in hydraulic circuit

A sudden change in oil velocity, such as that caused by sudden starting or stopping of hydraulic system, or if the oil makes a sudden change of direction, is commonly known as a "surge" or "water hammer". The common primary cause of this phenomenon is the quick closing of a valve. The wave of pressure surge travels through the pipe line at the speed of sound. The waves travel back and forth and are reflected at each boundary.

The calculation of water hammer described by [11] is based on the solution of the following system of basic differential equations:

$$\rho \frac{\partial v}{\partial t} + \frac{\lambda \rho}{2D}|v|v + \frac{\partial p}{\partial x} = 0 \tag{1}$$

$$a^2 \rho \frac{\partial v}{\partial x} + \frac{\lambda p}{\partial t} = 0 \tag{2}$$

where: ρ – density of liquid [kg/m³], λ – coefficient of friction [-], D – internal pipe diameter[m], a – velocity of propagation of the pressure wave [m/s], x – longitudinal coordinate along the pipe line axis [m], t – time [s], v – mean cross-sectional velocity of the liquid flow in the pipe-line through a cross-section with coordinate x at instant t [m/s].

Using equations (1) and (2), the calculation of the water hammer require to consider assumptions $v \ll a$ and approx. constant density. Closure of the pipe line at time $t = 0$ causes an increase of pressure Δp at the valve and this increment travels at velocity a in a direction opposite to that of the flow. The change of discharge from initial value Q_0 to zero travels at the same velocity [11].

1.2 Influence of pressure valve on the oil flow instabilities

Hydraulic systems are commonly found in various industrial applications. An essential component of such systems is the pressure valve (Fig. 1), which protects other system elements from overpressure peaks by reducing the pressure excess in a safe manner [12].

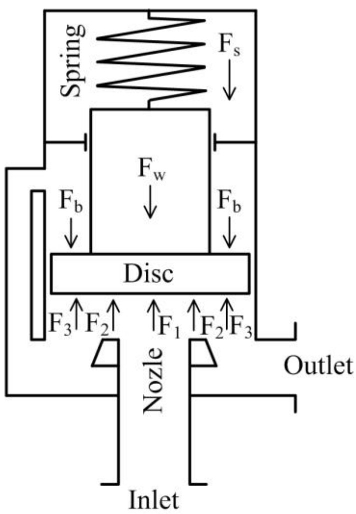


Fig. 1. Schematic diagram for the valve model: F_1 – forces acting under valve disc due to the fluid in the nozzle area, F_2 – forces acting under valve disc due to the fluid in the seat area, F_3 – forces acting under valve disc due to the fluid in the area exposed to the exit plane, F_b – back pressure acting on the disc, F_w – gravitational force on the moving parts, F_s – spring force.

It is well established that an opening valve connected to upstream piping can be subject to different kinds of instabilities, variously known as chatter, flutter and rapid cycling. These instabilities typically cause the valve body to oscillate and periodically close, often at high frequency, with considerable force. Such oscillations themselves are not necessarily issuable but they can cause significant mechanical vibrations, and large upstream pressure waves (especially in liquids owing to the well-known water-hammer effect) that can cause system damage. The main types of instabilities are characterised by (Fig. 2):

1. Inlet pressure loss. Cycling due to frictional and geometric losses within fluid flow in inlet piping. Causes low-frequency and low-amplitude chatter motion.
2. Cycling or low-amplitude chatter due to insufficient damping and/or the valve venting at small portion of its capacity.
3. Static jump in valve lift due to negative effective valve stiffness.
4. Flutter or chatter due to interaction between valve dynamics and the Helmholtz resonator formed by the tank plus the inlet piping.
5. Flutter or chatter due to negative damping of the fundamental "organ pipe" acoustic mode in an overly long inlet pipe [13].

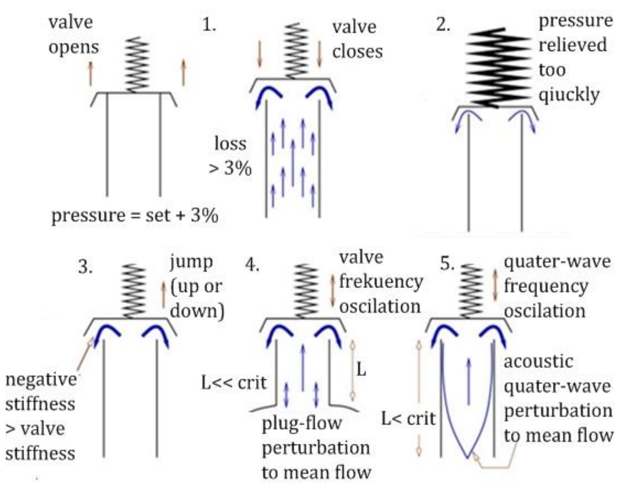


Fig. 2. Schematic description of each type of instability [13].

Erdődi and Hős [14] summarised a number of factors that contribute to the instability of the valve response, although no one study considered all of the following contributing factors:

1. The spring-mass-damping characteristics of the moving parts of the valve;
2. The net forces acting on the disk, which are primarily the pressure and momentum forces exerted by the fluid on the disk. These forces are sensitive to the valve-disk geometry that affects the flow path and direction of the fluid velocity vector as it departs from contact with the disk;
3. The setting of the blow-down ring, which determines the re-closure conditions for the valve. This setting also affects the geometry and dimensions of the flow path of the fluid in contact with the disk [14];
4. The dynamics of the fluid in the piping upstream of the valve, including friction loss, inertial and compressive forces and the capacitance of the inlet line and vessel vapour space, which includes the pressure surge due to the acoustic coupling associated with reflection of the expansion wave generated when the valve opens quickly (i.e. "pop action") [13];
5. The fluid dynamics and the pressure drop in the valve discharge line, which affect the backpressure on the disk and the re-closure conditions;
6. The disc lifts vs. flow characteristic of the relief valve at and below the design capacity.

Technical literature classifies pressure valve dynamic response according to Fig. 3.

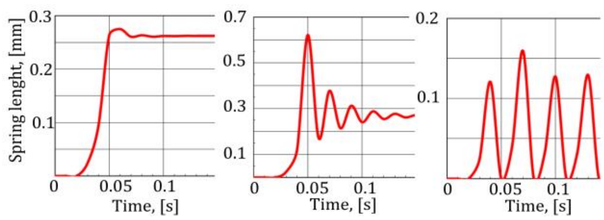


Fig. 3. Example of pressure valve dynamic response: a) stabile, b) flutter, c) chatter [15].

The dynamic response of the disk determines the position of the disk as a function of time, *x(t)*, and depends upon the mass of the moving parts (M), the damping of the moving parts (C), the stiffness of the spring (K), and the net driving force acting on the disk (F_1). The damping factor ζ, from Newton's second law, is:

$$\zeta = \frac{C}{\sqrt{2KM}} \tag{3}$$

where: ζ – damping factor, M – mass of the moving parts [kg], C – damping of the moving parts[N s/m] , K – stiffness of the spring [N/m], The valve is over-damped if $\zeta > 1$, critically damped if $\zeta = 1$, and will oscillate if $\zeta < 1$.

Darby [15] claims that so-called pressure surge criterion is often used to predict whether the opening of the valve will trigger chatter upon valve opening. At least in the case of liquid-service valves such reflected waves can lead to significant instantaneous jumps in pressure at the valve due to the well-known water hammer phenomenon. The valve is expected to operate in a stable manner, resp., not to chatter if:

$$t_{open} > t_w \text{ with } t_w = \frac{2L}{a} \tag{4}$$

where: t_{open} – opening time of the valve [s], t_w – twice the transmission line time[s], a – the effective velocity of sound [m/s], L – the inlet pipe length [m].

The research article of the authors from Budapest University of Technology and Economics [17] also described the instabilities of hydraulic pressure valves which are widely used in industrial hydraulics. They present that these valves tend to self-oscillate even under steady-state conditions, often referred to as valve chatter.

Similarly, Hayashi et al. [18] studied the instabilities and chaos occurring in a hydraulic control valve using an experimental hydraulic system. In this case, the circuit becomes unstable for an initial disturbance beyond a critical value and develops a self-excited vibration although the poppet valve rests on the seat in a stable manner for a supply pressure lower than the cracking pressure.

2. MATERIALS AND METHODS

2.1 Test bench generating the pressure surges

Design of the hydraulic circuit generating pressure shocks is based on the concept of a test bench intended for durability tests of tractor hydraulic pumps and biodegradable hydraulic oils [20-22]. The test bench uses the cyclic pressure loading (Fig. 4) to load the biodegradable oils. Standard describes the parameters of cyclic pressure loading, frequency interval from 0.5 Hz to 1.25 Hz.

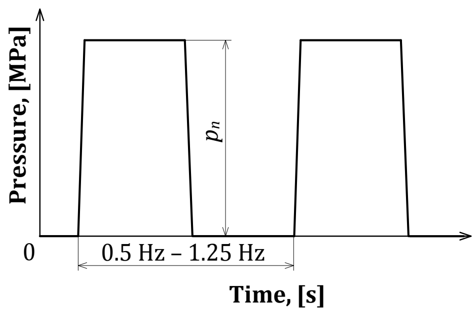


Fig. 4. Cyclic pressure loading according to standard [18].

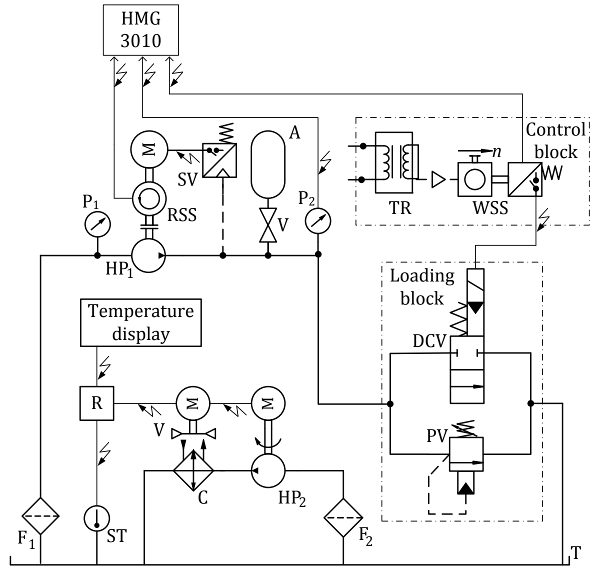


Fig. 5. The test bench: M – electric motor, RSS – rotation speed sensor, HP_1 – hydrostatic pump, PV – pressure valve, P_1 – pressure gauge of pressure in the inlet, P_2 – pressure gauge of pressure in the outlet, T – tank, DCV – two-positions, two-port directional control valve with closed centre which is operated electro-hydraulically, C – cooler, V – ventilator, HG_2 – hydrostatic pump for fan, ST – oil temperature sensor, R – thermostatic regulator which control switching on of cooling fans, F_1, F_2 – filters, TR – transformer, WSS – wiper-speed switch, SV – safety oil pressure switch, A – hydraulic accumulator, V – shut-off valve.

An external gear hydrostatic pump is used in the test bench to pump the oil through hydraulic circuit (Fig. 5). The hydrostatic pump is marked as HP_1. At process time, the loading of hydrostatic pump and hydraulic oil is realized by two basic parts, namely pressure valve PV and electro-hydraulically operated directional control valve DCV. Electromagnets of the valve are supplied by control block. The main parts of the control block is wiper-speed switch WSS. Therefore, the electromagnets of the directional control valve changes its-position depending on electric signal which is generated by control block. The directional control valve without voltage is in base-position and it blocks the oil flow to the tank T. In this position the oil flows through the pressure valve which is adjusted on nominal pressure (p_n = 20 MPa) of hydrostatic pump HP_1. An impulse of the control voltage changes the directional control valve position to the left position and allows the flow of oil straight to the tank T. In such a way, the test bench generates pressure surges during the loading of hydrostatic pump and testing of biodegradable hydraulic oils.

The data acquisition system consists of a portable measuring device HMG 3010 type (Hydac GmbH, Germay) with adapter and personal computer for evaluation of measured parameters [23-25].

The pressure valve VP 4-10 type (Pelikán, Vrchlabí, Czech Republic [26]) was used in the test bench to limit the maximum pressure in the hydraulic circuit. The base parameters of the pressure valve are as follows:

- maximum oil flow through pressure valve (100 dm^3/min),
- pressure range (0.5 MPa – 32 MPa),
- hydraulic oil temperature range (20 °C – 60 °C), [26].

2.2 Hydraulic oil in test bench

The hydraulic oil MOL Traktol ERTTO type [27] was used in the test bench (Slovnaft a.s., Slovak Republic). The hydraulic oil is an ERTTO-type (Environmentally Responsible Tractor Transmission Oil) and it is biodegradable tractor oil. The oil is made from vegetable nature oil and specific additives. The oil is dedicated to use in gearbox and hydraulic circuit of agricultural and construction machines. Primary biodegradation per CEC L-33-A-93 [28] is 90 % within 28 days. Parameters of the oil are as follows:

- kinematic viscosity at 100 °C, 10.38 mm^2/s,
- kinematic viscosity at 40 °C, 47.89 mm^2/s,
- viscosity index 213,
- pour point –39 °C.

2.3 The measurement conditions and procedure

Experiments were aimed at pressure surge analysis on the basis of measurements as follows:

- pressure surge by four modifications of the test bench,
- dynamic stability of the test bench operation,
- pressure surge at different frequencies of cyclic pressure loading.

The first experiment was realised on the basis of a hypothesis, that the damping capacity of the hydraulic system affects the pressure increase during the pressure surge according to Eq. (1) a (2).

To evaluate the pressure surge and dynamic stability of the test bench, four modifications of the test bench were used as follows:

- steel pipe between hydraulic pump and pressure valve and blocked hydraulic accumulator with closed shut-off valve; the steel pipe with internal diameter D = 15 mm and wall thickness 1.5 mm (test bench without any damping capacity),
- pressure rubber hose instead of steel pipe and blocked hydraulic accumulator with closed shut-off valve; the pressure rubber hose DIN EN 853 28N DN ½" W.P.275BAR/3990PSI - Masterline (test bench with the base damping capacity),
- correctly designed hydraulic accumulator (0.32 dm^3) with opened shut-off valve; SBO 210-0.32 E1 type with fluid volume 0.32 dm^3 (test bench with correctly damping capacity),
- overdesigned hydraulic accumulator (1 dm^3) with opened shut-off valve; SBO 200-1E1 type with fluid volume 1 dm^3 (test bench with oversize damping capacity).

The pressure surge measurements were realized with two hydraulic accumulators (Hydac, Sulzbach, Germany) with different volume. Accumulator simulation programme (ASP 5.0 – Hydac, Sulzbach, Germany) was used to select the hydraulic accumulator type and fluid volume. The first one was selected according to the standard selection procedure to reduce the pressure surge on the test bench. In the case of the second one, the higher fluid volume was selected to increase the damping properties of hydraulic system.

The test conditions during the pressure surge measurement by four modifications of the test bench were as follows:

- oil temperature (t = 50 °C ±2 °C),
- oil flow rate through hydraulic system (Q = 37.5 dm^3/min),
- pressure in hydraulic system - nominal pressure of hydraulic pump (p_n = 20 MPa),
- rotation speed of hydraulic pump (n = 1500 rpm ±50 rpm),
- inlet steel pipe line length between hydraulic pump and pressure valve (L = 2.2 m),
- frequency of cyclic pressure loading (1.5 Hz).

In this case, the hydraulic pump and pressure valve were connected to steel pipe with parameters mention above. These measurements were realized at sampling frequency 200 Hz.

The second experiment was aimed at the influence of various pressure surge frequencies on pressure increase. We tested factors affecting the pressure surge according to Eq. (4) at different time periods. These factors characterise dynamic response of moving parts in the valve, therefore we hypothesised that the shorter time period affects the pressure surge. During the pressure surge measurement at different frequencies of cyclic pressure loading, the oil temperature, the oil flow rate and the rotation speed of hydraulic pump were the same as in case mentioned above.

In this case, we modified the measurement conditions as follows:

- pressure of hydraulic system (p_n = 5 MPa),
- frequencies 1.5 Hz, 2.5 Hz, and 5 Hz of cyclic pressure loading were set on the test bench to observe the maximum pressure increase during the pressure surge.

Three frequencies were used to observe the pressure surge. Due to the relatively long time of the period by 1.5 Hz, the pressure set on pressure valve (5 MPa) is stabilized after the maximum pressure increase. The highest frequency (5 Hz) does not allow stabilising the pressure after the maximum pressure increase due to the short time period. The third one (2.5 Hz) was chosen in the middle of the interval from minimal to maximal value.

The pressure of hydraulic system was set by lower value than nominal pressure and the pressure hose connected the hydraulic pump with pressure valve by the reason of measurement safety.

We used the Fast Fourier Transform (FFT) to evaluate frequency spectres of pressure courses at different frequencies of the cyclic pressure loading.

2.4 Evaluation parameters

Parameters for evaluation of pressure surges and dynamic stability of the hydraulic system at the pressure set by the pressure valve (p_{pv}) are as follows, (Fig. 6):

- maximum pressure (p_{max}),
- maximum pressure increase ($p_{increase}$),
- maximum amplitude after the maximum pressure increase (p_a),
- opening time of the pressure valve for the statement of dynamic stability of the test bench (t_{open}).

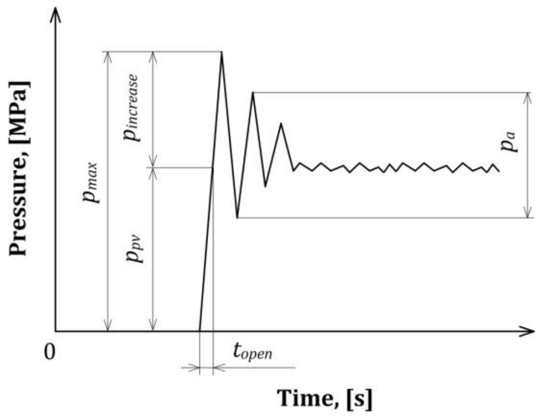


Fig. 6. Evaluative parameters.

These parameters were measured at steady state regime during the 30 seconds time period. 5 kHz sampling frequency was used during the measurements.

3. RESULTS AND DISCUSSION

3.1 Pressure surge in the case of the four modifications of the test bench

Figures 7, 8, 9 and 10 show the pressure surge on the test bench at four modifications, namely (1) steel pipe between hydraulic pump and pressure valve, (2) rubber pressure house between hydraulic pump and pressure valve, (3) correctly selected hydraulic accumulator (0.32 dm^3) and (4) oversized hydraulic accumulator (1 dm^3). The maximum pressure increase and amplitude of the pressure are shown in detail. The steel pipe has the smallest damping potential and so the pressure increases to the highest value (Fig. 7). The pressure rubber house represents the connection to the better damping properties comparing to the steel pipe. The damping of the pressure surge can be observed (Fig. 8).

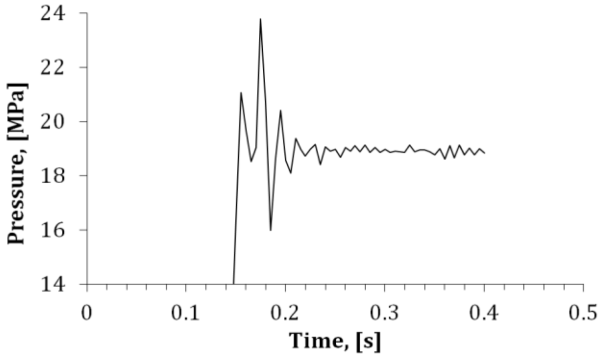


Fig. 7. Hhydraulic circuit with steel pipe.

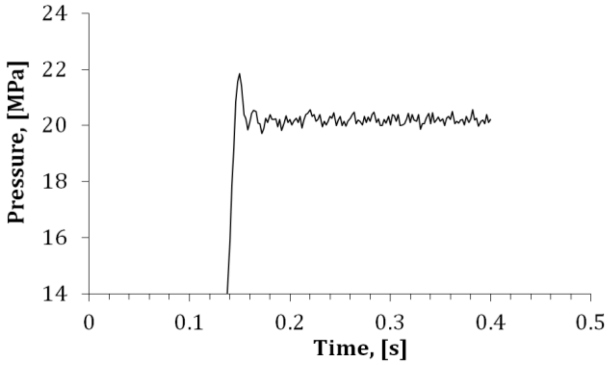


Fig. 8. Hydraulic circuit with rubber pressure hose.

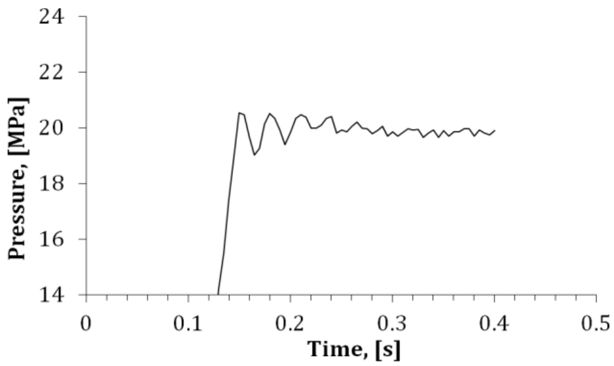


Fig. 9. Hydraulic circuit with correctly selected hydraulic accumulator (0.32 dm^3).

The hydraulic accumulator is an original damping device to eliminate the pressure surges in hydraulic circuits. The correctly selected type and volume of hydraulic accumulator can eliminate the pressure surge, close to the value of the pressure pulsation of hydraulic pump (Fig. 9). The oversized hydraulic accumulator with higher volume than correctly selected accumulator also eliminates the pressure surge but affects the origin pressure course (cyclic pressure load in this case), (Fig. 10).

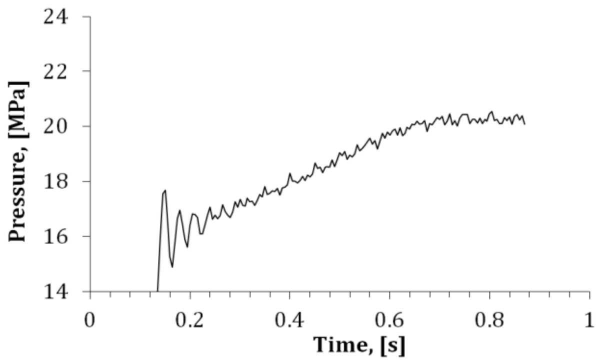


Fig. 10. Hydraulic circuit with oversized hydraulic accumulator (1 dm^3).

The maximum pressure in hydraulic circuit due to the pressure surge reached the values listed in Table 1.

Table 1. Pressure increase during the pressure surge at nominal pressure 20 MPa.

Type of inlet hydraulic line	Maximum pressure, [MPa]	Maximum pressure increase, [MPa]	Amplitude of pressure pulsation, [MPa]
Steel pipe	24.28	4.28	0.71
Pressure rubber hose	21.86	1.86	
Hydraulic accumulator (0.32 dm^3)	20.54	0.54	
Hydraulic accumulator (1 dm^3)	20.36	0.36	

The highest pressure increase in hydraulic circuit with steel pipe between hydraulic pump and pressure valve reached value 4.28 MPa. The pressure rubber hose was placed instead of the steel pipe. It reduced the pressure increase to the value 1.86 MPa. The correctly selected hydraulic accumulator with volume 0.32 dm^3 reduced the maximum pressure increase to value 0.54 MPa. The highest reduction of the pressure increase occurs in the case of oversized hydraulic accumulator with volume 1 dm^3. The pressure increase reached only 0.36 MPa. This value corresponds to amplitude of the pressure pulsation. In hydraulic circuit, the pressure pulses within amplitude 0.71 MPa.

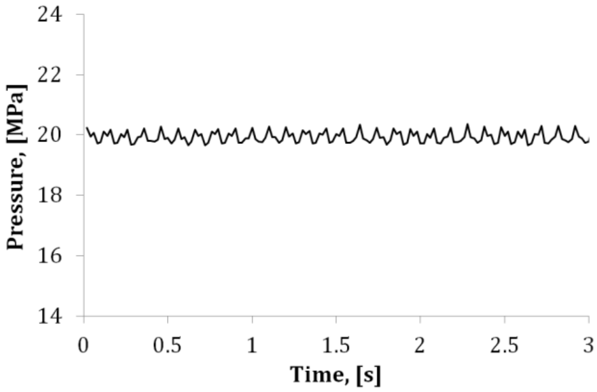


Fig. 11. Pressure pulsation of the hydraulic circuit at nominal pressure (20 MPa).

The external gear hydraulic pump generates the pressure pulsation due to the gears rotation, Fig. 11.

3.2 Dynamic stability of the test bench operation

Darby [15] stated that 'valve is expected to operate in a stable manner, resp., not to chatter, if twice the transmission line time of the expansion wave in the inlet pipe, generated by the abrupt valve opening, is shorter than the opening time of the valve'. In case of the test bench, it was stated that the opening time t_{open} (Table 2) is higher than twice transmission line time t_w = 0.003 s according to the equation (4), considering the inlet hydraulic line length L = 2.2 m and effective velocity of the sound in oil a = 1460 m/s. The opening time was stated from measured courses of pressure loading shown in Figs. 12, 13, 14 and 15. Therefore, the pressure valve on the test bench is expected to operate not to chatter.

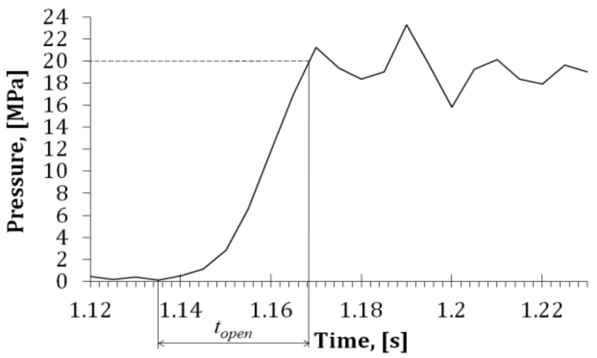


Fig. 12. The opening time of the pressure valve in hydraulic circuit with steel pipe.

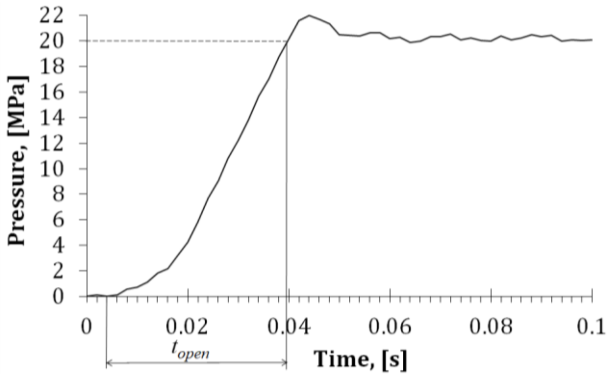


Fig. 13. The opening time of the pressure valve in hydraulic circuit with rubber pressure hose.

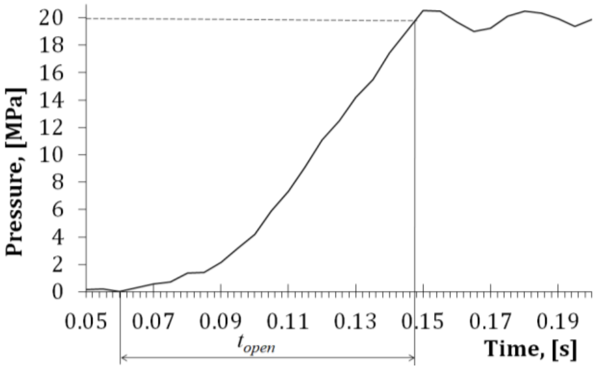


Fig. 14. The opening time of the pressure valve in hydraulic circuit with correctly designed hydraulic accumulator (0.32 dm^3).

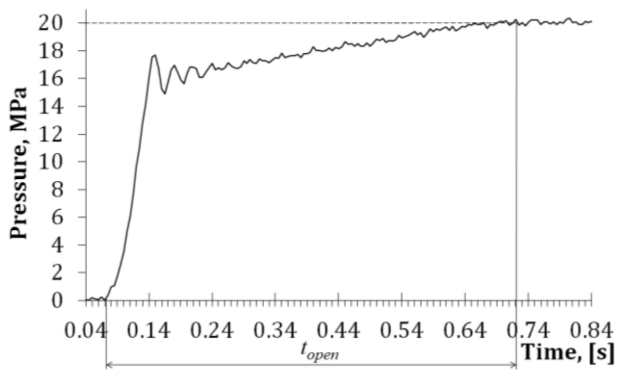


Fig. 15. The opening time of the pressure valve in hydraulic circuit with overdesigned hydraulic accumulator (1 dm^3).

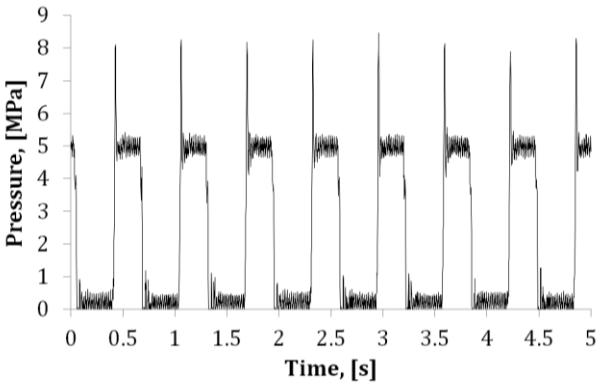
Table 2. Opening time of the pressure valve.

Type of inlet hydraulic line	Opening time, [s]
Steel pipe	0.0333
Pressure rubber hose	0.0360
Hydraulic accumulator (0.32 dm^3)	0.0879
Hydraulic accumulator (1 dm^3)	0.6043

Erdődi and Hős [14] researched an influence of changes of inlet pipe length (L) on dynamic response of the pressure valve API 526 Standard 1.5x3.0. The inlet pipe internal diameter was 3.4 cm with various lengths of inlet piping 0 m, 0.919 m and 1.83 m. The valve opening time was 30 ms.–Different inlet pipe length affects the stabile, flutter and chatter response of pressure valve. The dynamic response of the pressure valve VP 4-10 type is characterised by flutter considering the inlet pipe diameter and length.

3.3 Pressure surge at different frequencies of the cyclic pressure loading

Figures 16a, 17a and 18a show the pressure surge at different frequencies of the cyclic pressure loading (1.5 Hz, 2.5 Hz and 5 Hz). Figures 16b, 17b and 18b show FFT of pressure courses.



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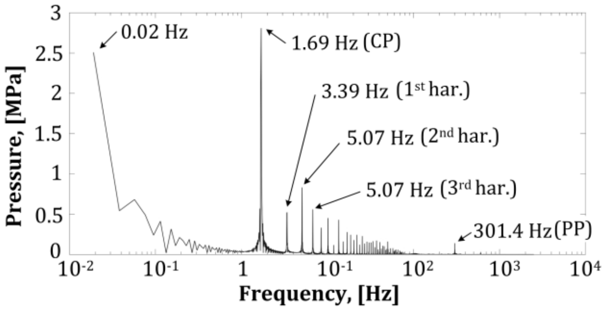
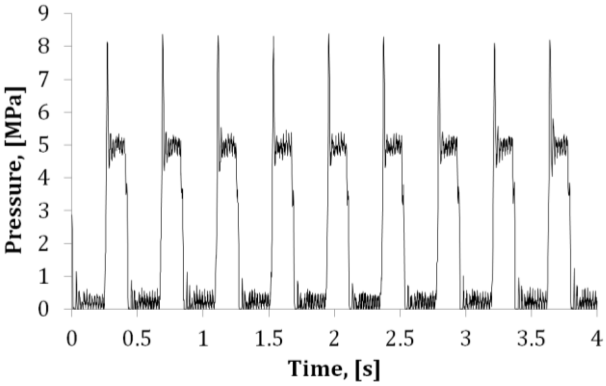


b)

Fig. 16. 1.5 Hz of the cyclic pressure load.



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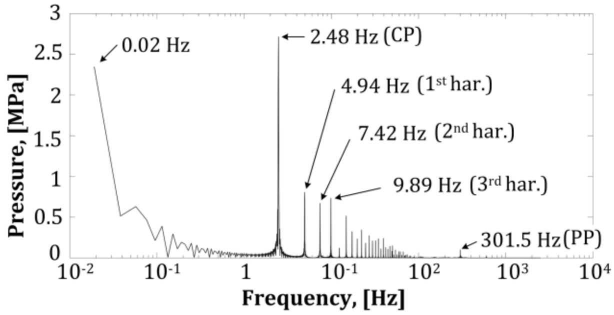
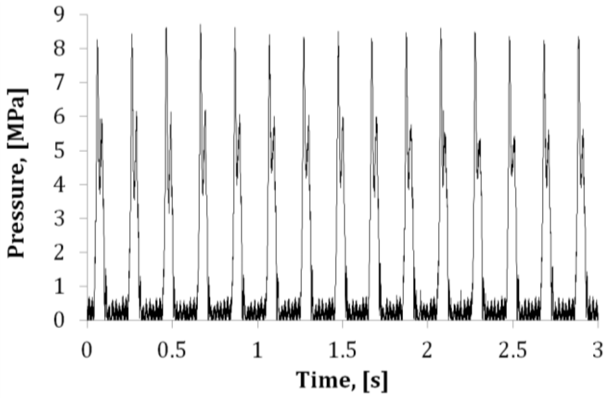


b)

Fig. 17. 2.5 Hz of the cyclic pressure load.



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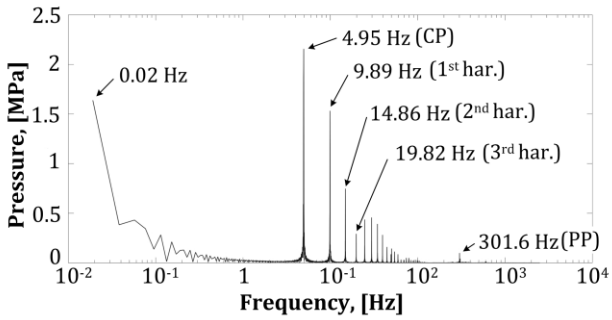

b)

Fig. 18. 5 Hz of the cyclic pressure load.

The time between the neighbouring pressure surges is long enough to stabilize the pressure set on the pressure valve (5 MPa) at 1.5 Hz of the cyclic pressure loading and oscillates only due to the hydraulic pump operation (Fig. 16). At higher frequency of the cyclic pressure loading, (2.5 Hz) the time for the pressure stabilization after pressure surge is shorter. The oscillation after the maximum pressure increase is reaching higher amplitude comparing to the 1.5 Hz of frequency. At 5 Hz of frequency, no stabilized pressure after the maximum pressure increase is observed, due to the very short time period. The maximum amplitude of the pressure oscillation after the maximum pressure increase reached the highest value (Tab. 3) at the highest frequency (5 Hz).

Table 3. Pressure increase at different frequencies of the cyclic pressure loading.

Parameter	Frequency of the cyclic pressure loading, [Hz]		
	1.5	**2.5**	**5**
Maximum pressure, [MPa]	8.66	8.76	8.84
Pressure set on pressure valve, [MPa]	5		
Maximum pressure increase, [MPa]	3.66	3.76	3.84
Maximum amplitude after maximum pressure increase, [MPa]	1.02	1.78	2.88

Figures 16, 17 and 18 show the course of the cyclic pressure loading. An experiment was realized under constant oil flow during the constant rotation speed of the hydraulic pump. The oil pressure on the test bench was set on the value of 5 MPa using the pressure valve.

A review of the numerical obtained results indicates that the self-excited vibration occurs due to the coincidence of water hammer, acoustic feedback in the downstream piping, high acoustic resistance at the control valve, and negative hydraulic stiffness at the control valve [29]. The spring of pressure valve is sensitive to reflected waves due to the pressure surge. It causes the instability of the pressure valve on the test bench during the opening and it leads to the pressure increase due to the pressure surge. The four frequencies of the control voltage were set to observe the pressure surge characteristic depending on the change of the net forces (F_1) acting on the pressure valve disc. Darby [15] presents the net forces as one of the parameters affecting the pressure valve instability. We suppose that the change of the pressure loading frequency affects the net forces and influences the pressure valve dynamic response.

The valve behaviour depends on how fast the valve can respond to the change in pressure and can be quantified by force distribution and momentum effects around the valve disc and the spring constant [12] according to equation (3). Equations (1) and (2) describe the parameters affecting the pressure increase during the pressure surge due to the sudden change of the oil velocity. In our case, all these parameters are constants. Therefore, the forces distribution around the valve disc and the spring constant (Fig. 1) affect the pressure surge response of the pressure valve.

Using the Fast Fourier Transform (FFT), the pressure courses at different frequencies of the cyclic pressure loading (Figs. 16, 17 and 18) were sampled over a period of time. The figures show the real frequencies (1.69 Hz, 2.48 Hz and 4.95 Hz) of the cyclic pressure loading (CP) which differ from frequencies set on wiper-speed switch (1.5 Hz, 2.5 Hz and 5 Hz). The wiper-speed switch setting error caused these not significant differences. The hydraulic pump generates the same pressure pulsations (PP) in the case of the all cyclic pressure loading frequencies. They reach values 301. 4 Hz (Fig. 16), 301.5 Hz (Fig. 17) and 301.6 Hz (Fig. 18) due to the rotation speed of the hydraulic pump (1,500 rpm) and teeth number (12) of the hydraulic pump gears. The next interesting parameter is the maximum amplitude after maximum pressure increase which varies depending on the frequency of the cyclic pressure loading (Table 3). The first harmonic frequency (1st har.) confirms this fact because the pressure at this frequency (Figs. 16, 17 and 18) also increases depending on the frequency of the cyclic pressure loading.

4. CONCLUSION

The characteristic of the all parts of the hydraulic circuit affects the pressure increase during the pressure surge. In the simple pipe-line system, the pressure surge occurs when the fluid flow is suddenly stopped. The pressure valve instabilities can be eliminates by the pressure valve construction in the best way or by regime of the valve operation in part. We analysed the influence of the four modifications of the test bench hydraulic circuit using steel pipe, pressure rubber hose, correctly selected accumulator and overdesigned accumulator to describe the system response. The rubber hose significantly eliminates the pressure surge if compared to the steel pipe. The hydraulic accumulators totally eliminate the pressure surge.

The opening time of the pressure valve is the parameter stating the stability of the hydraulic system. All the test bench modifications show higher opening time t_{open} than twice transmission line time t_w = 0.003 s considering the inlet hydraulic line length and effective velocity of the sound in oil.

The response of the hydraulic system about change in frequency of the cyclic pressure loading was also tested. The spring is a part of the pressure valve which oscillates during the valve opening and closing and generates the pressure waves. The increasing of the frequency affects the spring behaviour only minimally in the terms of maximum pressure increase; on the other hand, the maximum amplitude after the maximum pressure increase reached the highest value at the highest frequency due to the change in time for opening and closing of the pressure valve. The response of the hydraulic system about change in frequency of the cyclic pressure loading was also evaluated by Fast Fourier Transform (FFT). This method confirms the parameters of the pressure courses namely the frequency of the cyclic pressure loading, the pressure pulsation of the hydraulic pump and the pressure increase during the pressure surge.

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