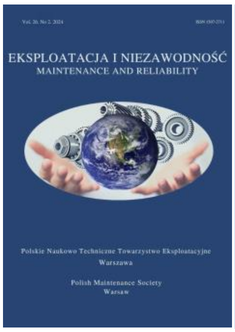

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The influence of the hydropneumatic accumulator on the dynamic and noise of the hydrostatic drive operation

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Highlights

- The possibility of limiting the maximum dynamic load during the start-up process observed.
- A dynamic model of a hydrostatic drive equipped with an accumulator was developed.
- The influence of the accumulator on the sound pressure level was analysed.
- With accumulator capacity increases -value of the dynamic excess coefficient decreases.
- Acoustic tests were carried out for a system with a linear hydrostatic drive simulator.

Abstract

The article presents the possibility of influencing the hydrostatic drive's dynamics and noise using a hydropneumatics accumulator. The possibility of limiting the maximum dynamic load during the start-up period of a drive equipped with an accumulator and controlled by a slide distributor is presented in the article. The influence of the accumulator on the sound pressure level and its relationship with the maximum dynamic values of the working fluid pressure in the hydraulic system is also described. Experimental studies of the dynamics and acoustics of laboratory hydrostatic systems were performed. A dynamic model of a hydrostatic drive equipped with an accumulator was developed, which was used to perform simulation tests illustrating the impact of the accumulator capacity on the system's dynamics. Acoustic tests were carried out for a system with a linear hydrostatic drive simulator.

Keywords
hydraulic drive, hydropneumatic accumulator, dynamic, noise, frequency, experimental measuring, simulation, reliability

1. Introduction

Given the extensive application of hydraulic drives in diverse technical domains, ranging from transportation engineering [11] to aviation engineering [13] and industrial sectors [7], there exists a compelling necessity for the continuous monitoring of the technical condition of their constituent elements. Reliability and maintenance analysis of different machines hydraulic drives is mainly oriented to hydraulic elements like pumps, cylinders and accumulators [8, 24]. The modern hydraulic drives in mobile machines or industrial systems are characterized by complexity, branching, and typically involve a multitude of distinct hydraulic elements (*Fig. 1*), including pumps, valves, throttles, cylinders, and other components. The critical challenge in preventive maintenance lies, for hydraulic drives, in the selection of the most valuable components. A pivotal aspect of this process involves identifying the factors that influence the reliability of the system [12, 27]. This underscores the significance for hydraulic system designers, particularly concerning dynamic influences and noise-related issues.

In recent years, there has been an intensive development of various types of control systems influencing the dynamics of hydrostatic drive systems, in particular "load-sensing" systems [26] or microprocessor-controlled systems using proportional technology [10, 18]. It is worth noting that the designer of the drive system, in addition to such basic parameters as output

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power, efficiency, etc., faces the problem of ensuring specific dynamic properties due to the specific operation of the designed machine - this is an increasingly frequently raised problem formulated as a necessary requirement that must be met must be a machine or device.

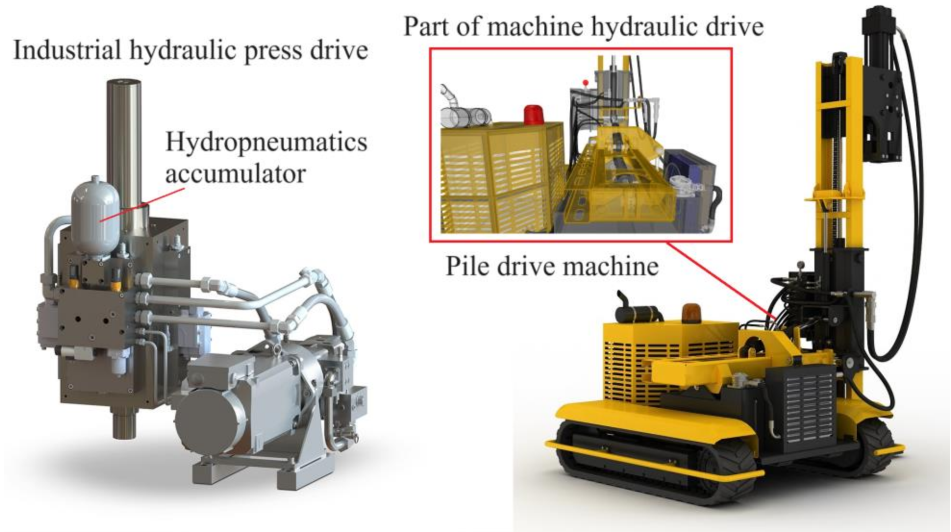


Fig. 1. The example of modern hydraulic drives in mobile machines and industrial systems.

Nowadays, the criteria for assessing machines and devices, especially machines with hydrostatic drives, have been recently expanded to include the criterion of their operation noise. For this reason, a properly constructed system, in addition to the assumed static and dynamic properties, should ensure the lowest possible level of emitted noise [6] - not exceeding the normative requirements. This may be confirmed by the requirement set out in Directive 2000/14/EC [4] regarding the permissible sound power level of machines emitting noise into the environment, for example, for a wheel loader with a power of up to 55 kW, the level of this power until January 3, 2002, was 104 dB, while as of January 3, 2006, the permissible acoustic power level for this type of machine was reduced to 101 dB. It should be mentioned that since February 15, 2006, the new Directive 2003/10/EC [5] on noise protection has been in force in Poland, as in all EU countries. This directive lowers the permissible maximum noise value at the operator's station, determined by averaging the value from 8-hour exposure to 80 dB(A) (previous value of 85 dB(A)). The subject of excessive noise raised is particularly important with regard to high-power hydraulic systems (greater than 1000 kW) mounted on decks aboard seagoing ships, where the noise level in the room of the hydraulic pump unit sometimes actually reaches as high as 110 dB posing a serious exploitation problem [1, 2]. It is also necessary to emphasize that noise and vibrations, especially in the low-frequency range (infrasound), have a very adverse effect on the human body [15, 21].

It should be noted that the human-machine dynamic system has a complex dynamic structure and is generally considered a non-linear and non-stationary system with feedback. The study of the impact of vibrations on the human body can be considered depending on [14]:

– The values of parameters describing vibrations, namely frequency, displacement amplitudes, velocities, and accelerations, their course and duration (these values are generally determined at the place where humans receive vibrations), and the direction of vibration action;

– The places of transmission of vibrations to a person and the position of reception, namely in a standing position, transmission through the feet and upper limbs, in a sitting position, transmission through the hips, and finally, in a lying position, transmission mainly through the back. The effect of vibrations is divided into general vibrations (e.g., transmitted by the legs or seat) and local vibrations (transmitted to the upper limbs);

– Individual human characteristics, such as age, height, build, weight, gender, health condition, nervous excitability, mental state, etc.

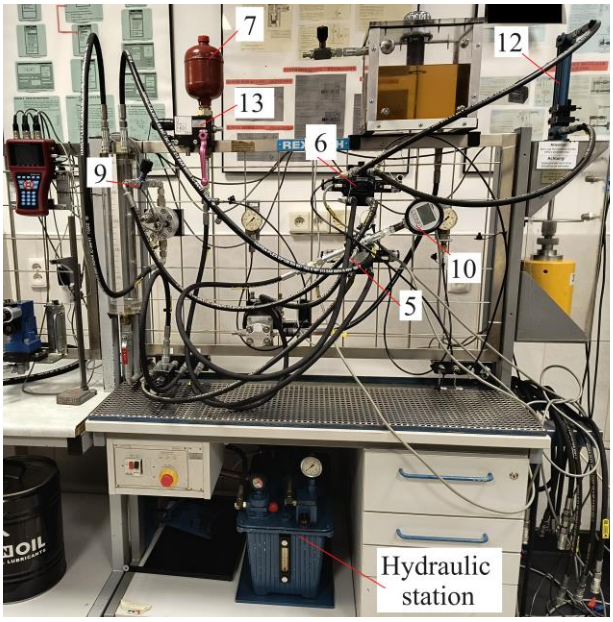
Measurement practice shows that the displacement pump is one of the primary sources of noise in hydrostatic drive systems.

Both literature data and our own experience indicate that the noise level emitted by a displacement pump is proportional to its load. Noise is emitted during transient states of hydrostatic drive systems, where, in most cases, the pressure course during start-up is oscillatory with damping, and therefore, the maximum pressure value occurs. It can be expected that the noise intensity during the period of drive transients will be variable, with extreme values also occurring. The coincidence between the maximum pressure value and the maximum noise value may be interesting.

Typically, the maximum pressure values in the dynamic states of hydrostatic drives are limited by appropriate control and setting the appropriate opening pressure of the pressure valves. The noise level of the pump affecting the operator can be easily reduced by using a suitable sound-absorbing housing. Sometimes, a hydropneumatic accumulator is used to alleviate transients. The hydropneumatic accumulator is also used to increase the energy efficiency of hydraulic systems. In the research [28] proposed an energy-saving system for a hydraulic excavator based on a three-chamber accumulator, which can store and reuse energy losses from throttling and overfilling the hydraulic system without changing the hydraulic system of the excavator. The potential energy of the boom during descent is converted to hydraulic energy and stored in a three-cell accumulator, which is then released to power the boom lift. The research [17] conducted research on a hydropneumatic accumulator with adaptive control in order to improve its efficiency and energy storage capacity. The proposed accumulator consists of a typical piston accumulator additionally equipped with a gas regulator and several control valves. In [22], was used a hydraulic accumulator to study its influence on pressure stabilization in the electro-hydraulic pressure regulation system in the gear-shifting mechanism using a wet clutch. The increasing demands placed on modern hydraulic systems and the growing competition from other types of drives (electric, pneumatic, hybrid) are forcing designers to eliminate the disadvantages hitherto encountered in hydraulic systems: excess dynamic pressure and excessive working noise. Among the many possible ways of reducing these parameters, the use of a hydropneumatic accumulator is a cheap and effective way. The use of simulation techniques makes it possible to make predictions about the effects obtained in the form of a reduction in dynamic pressure excesses.

2. Research objects and methodology

The tests were carried out on a laboratory hydrostatic drive system with a piston engine, the hydraulic diagram of which is shown in *Fig. 2*. There are two slide distributors in the system, Nr. 5 and 6. Manifold Nr. 5 cuts off the pump from the rest of the system, shorting it to the tank in its neutral position. In its neutral position, distributor Nr. 6 cuts off the hydraulic cylinder from the system and also serves to reverse its movement. The outflow of liquid from the lower chamber of the actuator is dictated by the reduction of flow losses in the system in the case in which the reversal of the actuator movement is carried out thanks to the control of distributor Nr. 6 and the action of gravity on the external load of the actuator. Multi-piston pump Nr. 2, type PTOZ-40R, driven by electric motor Nr. 1, type SZD-114 (power N = 11 kW, revolutions 1400 rpm) sucks oil from tank Nr. 11 through suction filter Nr. 3 and feeds it through electromagnetically controlled distributors Nr. 5 and 6, type RSR -12 to hydraulic cylinder Nr. 12 (D = 50 mm, d = 25 mm). The cylinder was loaded indirectly with the weight of the platform with a weight using a rope system. The system was protected against overload by a maximum valve Nr. 4, type ZP-42, whose high opening pressure eliminated its operation in the start-up dynamics tests. Moreover, the research system was equipped with a hydropneumatic accumulator Nr.7 (p_0 = 2.0 MPa, V_0 =2.5 dm^3), which could be disconnected from the installation using a shut-off valve Nr. 13.



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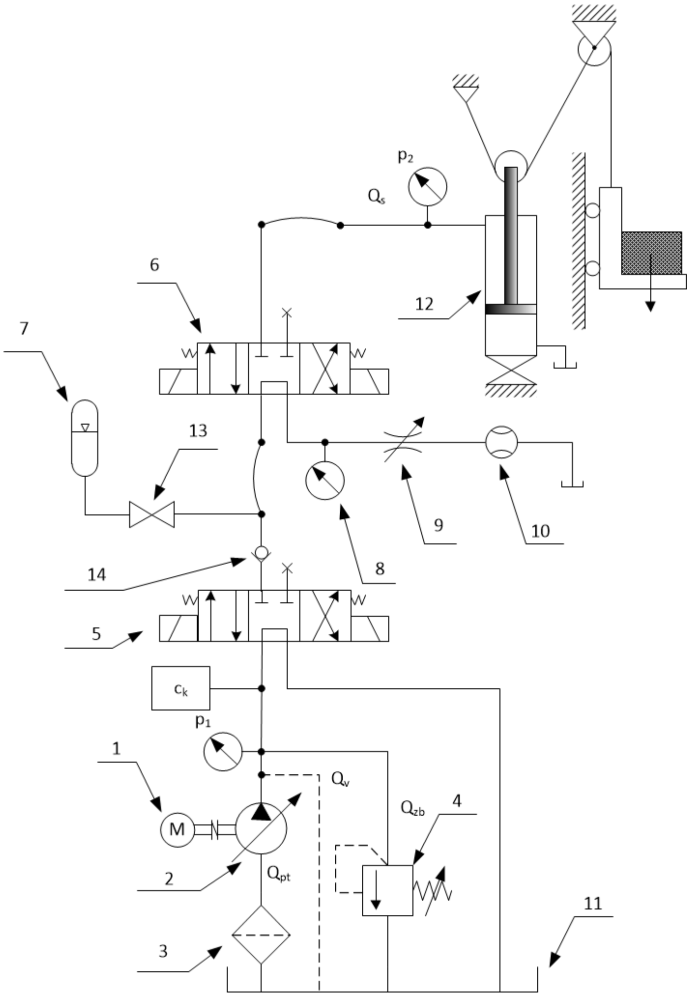


b)

Fig. 2. Laboratory hydrostatic drive system with a piston engine (1 - electric motor; 2 - multi-piston pump; 3 - suction filter; 4 - safety valve; 5,6 - electromagnetically controlled distributor; 7 - hydropneumatic accumulator; 8 - pressure gauge; 9 - throttle valve; 10 – flow meter; 11 – tank; 12 – hydraulic cylinder; 13 – cut-off valve; 14 – non-return valve): a) View of stand in laboratory conditions (some elements not visible); b) Hydraulic diagram of the experimental system with a hydraulic cylinder.

The literature [3, 19, 23, 29] provides mathematical models enabling analytical or numerical determination of the course of fundamental quantities during the start-up period of the hydrostatic system when forced by a step flow rate corresponding to a quick switching of the distributor. These models can be divided into models with lumped parameters and distributed parameters. Models with distributed parameters assume a finite time velocity of pressure wave propagation and describe pressures as a function of two parameters: time and distance, using partial differential equations. The solution of such equations is complicated and often implemented using the method of characteristics (MOC). A separate group of models are models with lumped parameters in which the system's features are concentrated in points. Such models are built based on ordinary differential equations, and the variables (pressure, velocity) of these models are time functions. The presented considerations assume modelling based on a system of ordinary differential equations. The first equation was the equation for the balance of forces acting on the actuator piston, and the second equation was the equation for the balance of the flow rate, considering the flow caused by the system's capacitance concentrated at the points of the system. The general integral of a system of differential equations represents a family of integral curves. The special integral is obtained after assuming the initial conditions.

Before building a mathematical model for the start-up of a hydrostatic transmission, simplifying assumptions were made: the speed of the engine driving the pump is constant and does not depend on the load; clearances in the drive system elements do not change during the operation of the mechanism and the working medium has unchanged physical properties; lumped parameters are assumed; leaks occurring in the drive system components were mapped with laminar flow; the effect of pump pulsation is ignored; the switching of the distributor, i.e., the connection of the pump with the actuator, takes place in steps in time Δt =0; the viscous nature of damping is assumed, the system is vented and that there are no cavitation phenomena in the considered system, the influence of position of oil filter on damping coefficients is negligibly small (Fig. 3).

The equation for flow continuity in the pump discharge port is:

$$Q_{pt} = Q_s + Q_v + Q_{zb} + Q_c, \tag{1}$$

where, Q_{pt} – theoretical efficiency of the pump; Q_s – actuator absorption; Q_v – leaks in the system; Q_{zb} – flow through the safety valve; Q_c – flow caused by compressibility.

It was assumed that the system start-up takes place without the participation of the safety valve. Therefore, Q_{zb} = 0 m^3/s.

For a piston engine, the absorption capacity is expressed as:

$$Q_s = A \cdot v, \tag{2}$$

where, A – piston surface; v – piston speed.

Leakage losses depend linearly on the pressure in the system, so we can write the following:

$$Q_v = a \cdot p, \tag{3}$$

where, a – proportionality coefficient, the value of which can be determined based on the static characteristics of the elements from which the system is assembled; p – nominal pressure value in the system.

The flow caused by the compressibility and deformation of elements is determined according to the following relationship:

$$Q_c = c_k \cdot \frac{dp}{dt}, \tag{4}$$

where, c_k - compressibility capacity of the system (capacitance).

Taking into account the relationships Eqs. (2)-(4) and that $Q_{zb} = 0$ m^3/s, the flow rate balance can be written as:

$$Q_{pt} = A \cdot v + a \cdot p + c_k \cdot \frac{dp}{dt}. \tag{5}$$

The balance equation of forces acting on the actuator piston is as follows:

$$m_{zr} \cdot \frac{dv}{dt} + f \cdot v + F = p \cdot A \tag{6}$$

where, m_{zr} - reduced mass of the external load, moving parts of the actuator (piston, piston rod) and working fluid; f - coefficient of fluid friction determined based on the mechanical efficiency of the actuator and the hydraulic efficiency of the circuit; F - external load; A - active surface area of the piston; t - time.

Equations (5) and (6) allow, after solving, the determination of the pressure course during the start-up of the analysed system. The accumulator's participation in the system starting process is described in the mathematical model by the system capacitation value.

If the pressure course is considered over time, two stages of system start-up can be distinguished. For a system without an accumulator and in which the actuator is loaded with the force of gravity of the mass element rigidly connected to the actuator piston rod, the first stage of start-up, counted from the moment of forcing the step flow to the moment of obtaining the pressure value occurring in steady motion p_{stab}, is described by the following equation [14]:

$$p_I = \frac{Q_{pt}}{a} \cdot \left(1 - e^{-\frac{a}{c_k} \cdot t}\right), \tag{7}$$

where, t – time; a – leakage coefficient; c_k – capacitance; Q_{pt} – theoretical pump efficiency.

Equation (7) was obtained for the initial conditions: t = 0; v = 0; p = 0. The duration of the first stage, i.e., the time after which the pressure in the system reaches the steady-state pressure is:

$$\tau = \frac{c_k}{a} ln \frac{Q_{pt}}{Q_{pt} - p_{stab} \cdot a}, \tag{8}$$

The final conditions of stage 1 (index 1 k), which are also the initial conditions (index 2 p) for stage 2, will be obtained for $p = p_{stab}$, taking into account the relationship (Eq. 8) in Equation (5). These conditions are: $p_{1k} = p_{2p} = p_{stab}$.

$$\left(\frac{dp}{dt}\right)_{1k} = \left(\frac{dp}{dt}\right)_{2p} = \frac{Q_{pt} - p_{stab} \cdot a}{c_k}. \tag{9}$$

After Laplace transformations and solving the system of Eqs. (5) and (6), the pressure course in the second stage of start-up is described by the following equation [16]:

$$p_{II} = p_{stab} + \frac{Q_{pt} - p_{stab} \cdot a}{c_k \cdot \omega} \cdot e^{-\zeta \cdot \omega_0 \cdot t} \cdot sin(\omega \cdot t). \tag{10}$$

The individual symbols in Equation (10) mean:

- circular frequency of undamped natural vibrations:

$$\omega_0 = \frac{A}{\sqrt{c_k \cdot m_{zr}}} \tag{11}$$

- circular frequency of damped vibrations of the system:

$$\omega = \omega_0 \sqrt{1 - \zeta^2} = \sqrt{\frac{A^2}{m_{zr} \cdot c_k} - \left(\frac{m_{zr} \cdot a + f \cdot c_k}{2 \cdot m_{zr} \cdot c_k}\right)^2} \tag{12}$$

- reduced damping coefficient:

$$\zeta = \frac{m_{zr} \cdot a + f \cdot c_k}{2 \cdot A \cdot \sqrt{c_k \cdot m_{zr}}} \tag{13}$$

The next research object was the Hydropax ZY25 linear hydrostatic drive simulator. The hydraulic simulator is a research device for the reciprocating drive system. This device reflects the actual operating conditions of devices with this type of drive. This simulator consists of three main parts: hydraulic part; control device; control program.

An important element of the simulator is the electrohydraulic amplifier Nr. 6, which controls the operation of the actuator Nr. 7. It is possible to connect the hydropneumatic accumulator Nr. 4 (V_0 = 1 dm^3; p_0 = 1.5 MPa) to the hydraulic system of the simulator using the shut-off valve Nr. 3.

During the tests, the simulator worked as a generator of mechanical vibrations with a fixed frequency not exceeding 100 Hz and of the form $w(t) = w_0 \cdot sin(2\pi f t)$. It is, therefore, a quasi-established work.

The diagram of the simulator's hydraulic system is shown in Fig. 3.

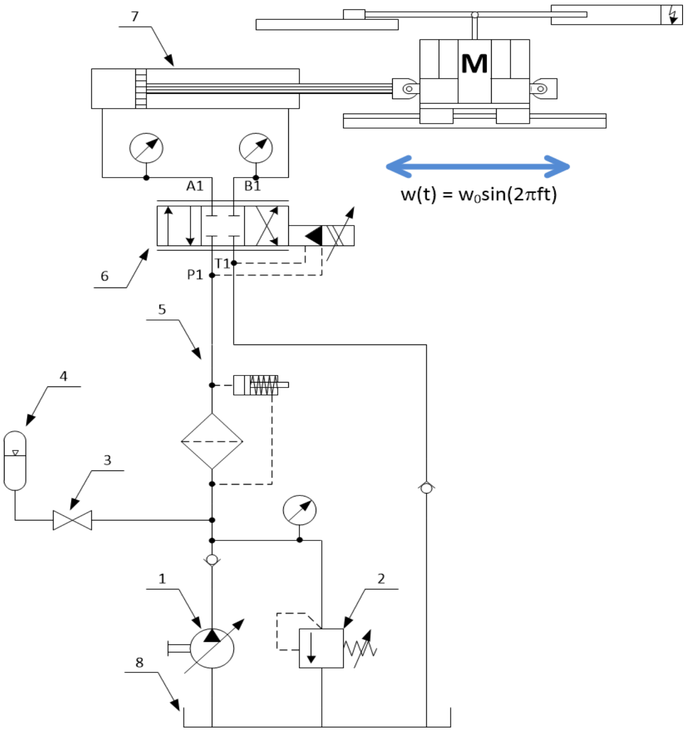


Fig. 3. Diagram of the parts of the hydraulic system of the linear hydrostatic drive simulator: 1 – variable displacement pump; 2 – maximum valve; 3 – shut-off valve; 4 – accumulator; 5 – oil filter; 6 – electrohydraulic amplifier numerous; 7 – setting actuator; 8 - hydraulic oil tank.

3. Experimental results and simulation

In the Fig. 4 shows the actual pressure course during the start-up of the test system in Fig. 2 with an installed hydropneumatics accumulator with initial volume V_0 = 2.5 dm^3 and initial charging pressure p_o = 2MPa. The forcing was a step changeover of the distributor, which resulted in liquid being supplied to the actuator from the pump.

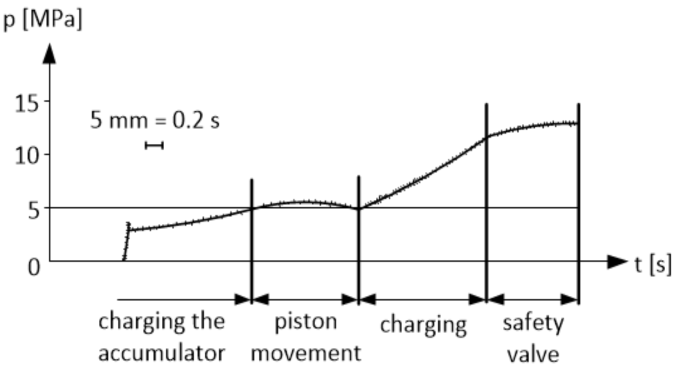


Fig. 4. Actual pressure course during start-up of the test system with the accumulator.

The theory of differential equations shows that the nature of the pressure increase during start-up depends on the value of the reduced damping coefficient (Eq. 13) $\zeta < 1$ – oscillatory waveform; $\zeta \geq 1$ - asymptotic course to the value in steady motion. Fig. 5 shows the dependence of the reduced damping coefficient on the capacitance for the test system (Fig. 2) with the accumulator. The assumed values of the system parameters:

- active surface of the piston: $A = 9.5\ 10^{-4}\ m^2$;
- leakage rate: $a = 4.6 \cdot 10^{-12}\ m^5/Ns$;
- system capacitance:

without accumulator: $c_k = 4.33\ 10^{-12}\ m^5/N$;

with accumulator: $c_k = 2.1 \cdot 10^{-10} m^5/N$;

- theoretical pump efficiency: $Q_{pt} = 0.63\ 10^{-3}\ m^3/s$;
- reduced mass: m_{zr} = 1220 kg;
- reduced fluid friction coefficient: f = 3585 Ns/m.

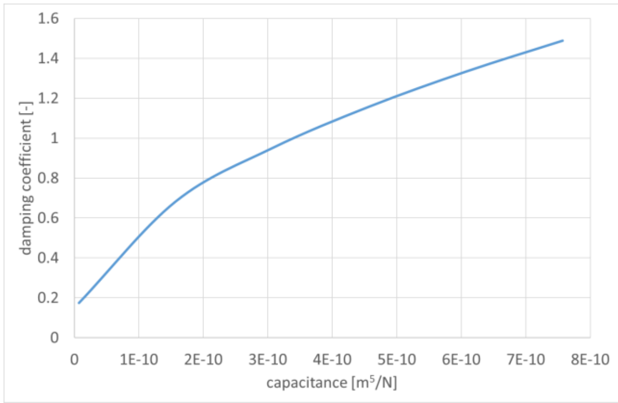


Fig. 5. Dependence of the reduced damping coefficient on the capacitance of the hydrostatic test system (system with the accumulator).

Acoustic tests were also carried out, determining the sound pressure level as a function of the maximum pressure value during the start-up of the test system in Fig. 2. The results of these measurements are shown in Fig. 6. The figure indicates the occurrence of coincidences between these two parameters.

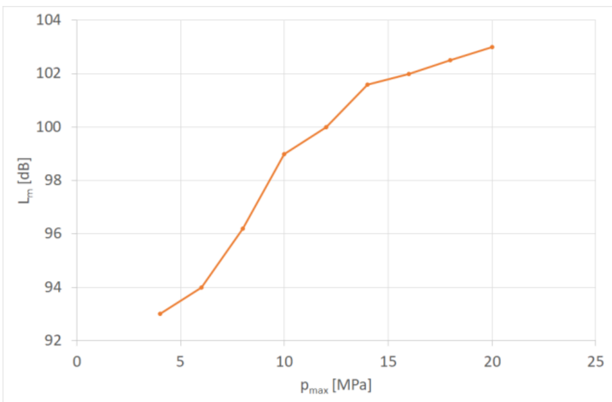
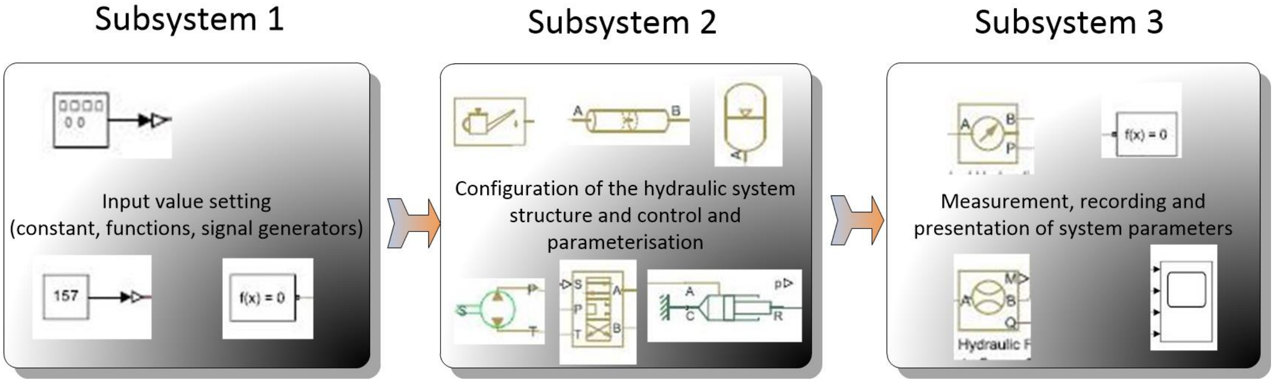


Fig. 6. Dependence of the sound pressure level L_m [dB] on the maximum pressure value p_{max} during the start-up of the test system.

Using the *Matlab-Simulink* package and the *Simscape Fluids library*, the influence of the accumulator volume on the starting process of the hydrostatic transmission shown in Figure 2 was analysed. The simulation graph and results are presented graphically in Fig. 7a and Fig. 7b respectively.



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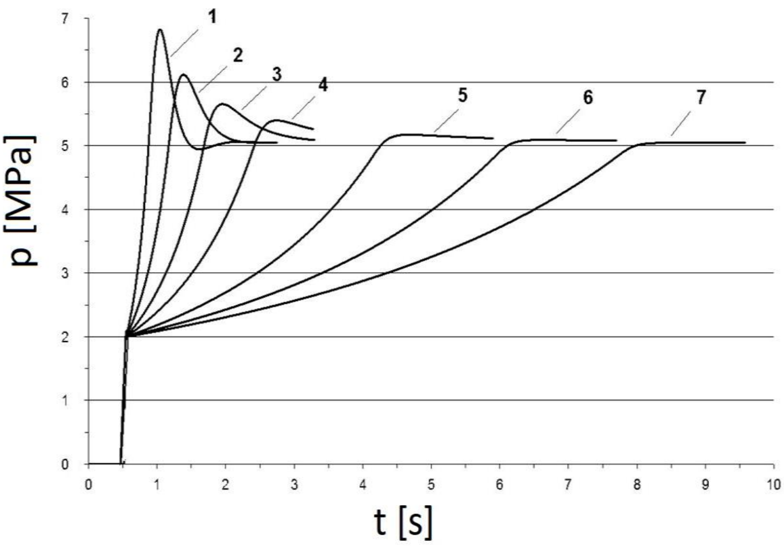

b)

Fig. 7. Simulation part of the research: a) Simscape Fluids graph of simulated hydraulic system with accumulator; b) Influence of the volume of the hydropneumatic accumulator on the nature of the start-up of the hydrostatic system for V_0 :1 - 0.4 dm^3; 2 - 0.8 dm^3; 3 - 1.5 dm^3; 4 - 2.5 dm^3; 5 - 5 dm^3; 6 - 7.5 dm^3; 7 – 10 dm^3.

This software is based on models with lumped parameters described by ordinary differential equations. It does not take into account wave phenomena. In cases where phenomena that require partial differential equations may occur, the package simulation results may differ from experimental measurements. Moreover, with numerical methods of solving equations, the Runge effect may occur, causing distortions in the solution of equations, manifesting themselves in false oscillations of the solution (e.g., pressure course during system start-up) [9, 25]. As a measure of the dynamic surpluses occurring during the transition process, the dynamic surplus coefficient can be used, defined as:

$$\varphi_d = \frac{e_{max} - e_{stab}}{e_{stab}}, \tag{14}$$

where, e_{max} - means the maximum signal value during the analysed transient process; e_{stab} - means the determined signal value after the transient process.

In the case of analysing the dynamic pressure course, the dynamic excess coefficient will describe the relationship between the maximum pressure value and the pressure value during steady-state operation. Table 1 shows the values of the dynamic excess coefficient for the case of the hydrostatic transmission shown in Fig. 2 for different volume values of the hydropneumatics accumulator.

Table 1. Volume values of the hydropneumatics accumulator and the corresponding values of the dynamic excess coefficient.

V_0, dm^3	ϕ_d
0.4	0.366
0.8	0.208
1.5	0.109
2.5	0.069
5	0.02
7.5	0.01
10	0.002

As the accumulator capacity increases, the value of the dynamic excess coefficient decreases, which indicates a decreasing overload of the system. However, this decrease is not linear, which means that a two-fold increase in accumulator capacity is not accompanied by a two-fold decrease in the value of the dynamic surplus coefficient.

Research was also performed to determine the impact of the hydropneumatics accumulator on the maximum pressure values during the quasi-steady operation of the hydraulic simulator. Fig. 8 shows the ratio of maximum pressure amplitudes for selected operating frequencies of the simulator without and using the accumulator.

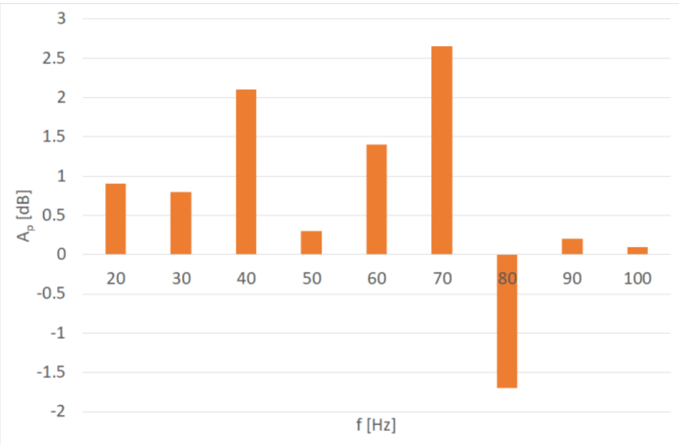

Fig. 8. The ratio of the maximum pressure amplitude for different operating frequencies of the simulator.

The ratio of the amplitudes of the measured quantities (pressures) was written in the following form:

$$A_i = 20 \cdot log\left(\frac{i_{out}}{i_{with}}\right)\ [dB], \tag{15}$$

where, i_{out} , i_{with} – measured quantity: pressure (p), lower indicators "*out*" or "*with*" refer to the test system without or with a hydropneumatic accumulator.

Positive values of the ratio of the amplitudes of the measured values indicate a reduction in levels after including a hydropneumatics accumulator in the simulator system. The negative value presented in Figure 8 (for a frequency of 80 Hz) means that the amplitude of the maximum pressure value of the system without an accumulator was smaller than for the system operating with an accumulator. Such a situation may be caused, among other things, by the mechanical resonance of moving elements of the simulator (e.g. the simulator table).The effectiveness of the accumulator's reduction of maximum pressure values will be greatest around the frequency determined by the following formula [20]:

$$\omega = \sqrt{\frac{\kappa \cdot p \cdot A_{accu}^2}{V_G \cdot m}}, \tag{16}$$

where, κ – polytropic exponent; p – operating pressure; A_{accu} – accumulator surface; V_G – gas volume.

$$m = m_{pA} \cdot \frac{A_{accu}^2}{A^2} + m_A, \tag{17}$$

where, m_{pA} – mass of liquid between the pump and the accumulator; m_A – mass of liquid associated with the accumulator; A – cross-sectional area of the pipe.

After taking into account the parameters of the simulator's hydraulic system, the frequency value ω is approx. 130 rad/sec. In example case, for a simulator operating at a frequency of 60 Hz, after connecting the accumulator to the system, the sound pressure level was reduced by 1 dB. However, to start the hydrostatic transmission (shown in Fig. 2) by turning on the hydropneumatics accumulator, the sound pressure level was reduced by 6 dB. Ultimately, shaping the starting process of a hydrostatic transmission will depend on damping. Its form is determined by the following characteristic values of the drive system: volume loss coefficient a; system capacitance c_k; resistance to movement and flow of the medium characterized by the factor f; piston surface area A.

Analysing relationship Eq. (13), the real values with which the designer can influence the damping value are the value of the leakage coefficient and the system capacitance c_k. By increasing the leakages, obtained worsen the efficiency of the drive, therefore, it is more convenient to shape the value of c_k. In another example, in the test system without a hydraulic accumulator, at $c_k = 4.33 \cdot 10^{-12}$ m^5/N, the damping coefficient becomes $\zeta = 0.11$, while for $c_k = 2.1 \cdot 10^{-10}$ m^5/N by installing a hydropneumatics accumulator, a reduced damping coefficient was achieved $\zeta = 0.9$. This resulted in a lower sound pressure level during the start-up of the hydrostatic transmission. It should be noted that by increasing the damping coefficient – obtain an increase in the reaction time τ (stage 1). The presented analysing of influencing the dynamics and noise level of the hydrostatic system with hydropneumatics accumulator held for engineering to reduce the risk of noise injury, equipment failure, energy loses or environmental disaster of hydraulic drives using.

4. Conclusion

In the research was investigated influence of the hydropneumatics accumulator on transient and quasi-steady processes of hydraulic systems. The use of an accumulator resulted in lower maximum pressure values and reduced the noise level. Based on the review and research established that there are various ways to reduce dynamic surpluses (e.g., pressure) during transient processes of hydraulic systems, however, the use of a hydropneumatics accumulator is a relatively inexpensive solution and, despite some shortcomings (e.g., extended start-up time, reduction of the system's natural vibration frequency by increasing capacitance), is acceptable in some cases. The influence of the accumulator on the sound pressure level and its relationship with the maximum dynamic values of the working fluid pressure in the hydraulic system is also described. Experimental studies of the dynamics and acoustics of laboratory hydrostatic systems were performed. A dynamic model of a hydrostatic drive equipped with an accumulator was developed, which was used to perform simulation tests illustrating the impact of the accumulator capacity on the system's dynamics. Acoustic tests were carried out for a system with a linear hydrostatic drive simulator.

The use of an accumulator resulted in lower maximum pressure values and reduced the noise level. For a simulator operating at a frequency of 60 Hz, after connecting the accumulator to the system, the sound pressure level was reduced by 1 dB. However, to start the hydrostatic transmission shown in Figure 2, by turning on the hydropneumatic accumulator, the sound pressure level was reduced by 6 dB. Ultimately, shaping the starting process of a hydrostatic transmission will depend on damping. Its form is determined by the following characteristic values of the drive system:

- volume loss coefficient a,
- system capacitance c_k,
- resistance to movement and flow of the medium characterized by the factor f,
- piston surface area A.

The main finding includes a statement - greater the capacitance of the system - the greater the damping. Obtaining a sufficiently high value of compressibility capacity is achieved by installing an appropriately sized hydropneumatics accumulator. In final it was disclosed the possibilities of influencing the dynamics and noise level of the hydrostatic system using a hydropneumatics accumulator in order to reduce the risk of noise injury, equipment failure, energy loses or environmental disaster. The greatest effectiveness of the reduction of excess pressure by the accumulator occurs when its natural frequency coincides with the frequency of the forcing. This is also the case for harmonic excitations. However, if the aim is to mitigate dynamic excess during hydraulic transmission start-up, the system designer faces a dilemma: dynamic excess (e.g. pressure) - start-up time. Increasing the reduction of excess pressure entails increasing the start-up time.

Compliance with Ethical Standards:

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