Laboratory tests of inlet pressure changes of electrohydraulic vibrations generator

***Roman Korzeniowski*[1] *and* *Janusz Pluta*[2]**

Laboratórne testy zmien vstupného tlaku elektrohydraulických vibrácii generátora

In the paper laboratory researches are presented of the multi pump hydraulic power unit used for supplying pressure to the hydraulic systems, which characterise frequently occurring dynamic states of operation. It concern supplying e.g. electrohydraulic servodrive working in the capacity of shakers generating mechanical vibrations. Researches are carried out in order to characterise the mutual interaction of the hydraulic power supply and its receiver working under dynamic states. The authors investigate how real pressure course in the outlet of the hydraulic power supply is differing from the set up value stabilised by pressure relief valve. The influence of the hydraulic power unit surplus output flow and following pressure supply changes on the servomechanism operation is being investigated as well. Most interesting results of researches are illustrated by the time characteristics.

***Key words**: dynamic state, hydraulic power supply, laboratory tests, servodrive*

Introduction

Many hydraulic control systems draw power from the hydraulic power supplies, described as the constant pressure source. It concerns mainly the throttling control hydraulic systems where hydraulic fluid supply should be realised with surplus of output flow in relation with receiver absorptivity. It concerns mainly the systems based on proportional and servovalve technique. This type of systems is used in mechanical vibration generators, active and semiactive vibration reduction systems, and also in other systems where way of working is characterised by occurrence of frequent dynamic states. Important difficulties with stabilisation of pressure supply may occur in such conditions. Many factors can be mentioned, depending on pressure stabilisation accuracy. Consequently this may influence working factors of the hydraulic receiver supplied from such a power supply. As it was presented in publication (Jaracz at al., 2004), above-mentioned hydraulic systems show sensitivity to pressure supply changes but their mathematical models are mainly created under assumption, that ideal constant pressure sources are applied (Korzeniowski at al., 2005; Kwaśniewski at al., 2003; Koňařík, 2006; Strážovec at al., 2004; Pluta at al., 2003). It may lead to considerable discrepancies between results of simulation tests based on such model and results obtained from real object. To state in what way the real constant pressure hydraulic power supply acts under dynamic states of receiver work suitable laboratory researches have been carried out.

Description of the system

Presented in the Figure 1 laboratory stand using for researches consist of multi pump hydraulic power unit 1 piped up with electrohydraulic servodrive 2. Simplified scheme of the laboratory stand was presented in the Fig. 2. Hydraulic power unit is equipped with three fixed displacement pumps where pumps HP1 and HP3 have the same output flow rate (16 $l.min^{-1}$) and the pump HP2 has less output flow rate (10 $l.min^{-1}$). In the hydraulic power unit there are used popular fixed displacement piston pump with swash plate where each of one have seven pistons. Cross section of such pump is presented in the Fig. 3. Electrohydraulic servodrive consist of double acting hydraulic cylinder HC and two stage servovalve SV with mechanical feedback. The presented system enables supply cylinder HC either from one, two or three pumps. Flow rate of the fluid current, supplying from one or both pumps cylinder servovalve SV, can be decreased by leading part of flow current to the tank by flow regulators FR1, FR2 or FR3. If the hydraulic power supply is working with pumps surplus output flow rate in proportion to receiver absorptivity, fluid pressure is stabilised on their outputs by pressure relief valves PV1, PV2 or PV3 (they can work two or three at the same time). Work of the hydraulic power supply can be additionally assisted by hydraulic accumulator HA. Laboratory stand was equipped in displacement transducer DT, pressure transducers PT1 and PT2, as well as flow meter FM.

[1] *dr inż. Roman Korzeniowski,* Department of Process Control, Faculty of Mechanical Engineering and Robotics, AGH University of Science and Technology, Krakow, Poland, korzerom@agh.edu.pl

[2] *dr inż.* Janusz Pluta, Department of Process Control, Faculty of Mechanical Engineering and Robotics, AGH University of Science and Technology, Krakow, Poland, plutian@agh.edu.pl

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Fig. 1. View of the laboratory stand: 1 – hydraulic power supply, 2 – electrohydraulic servodrive.

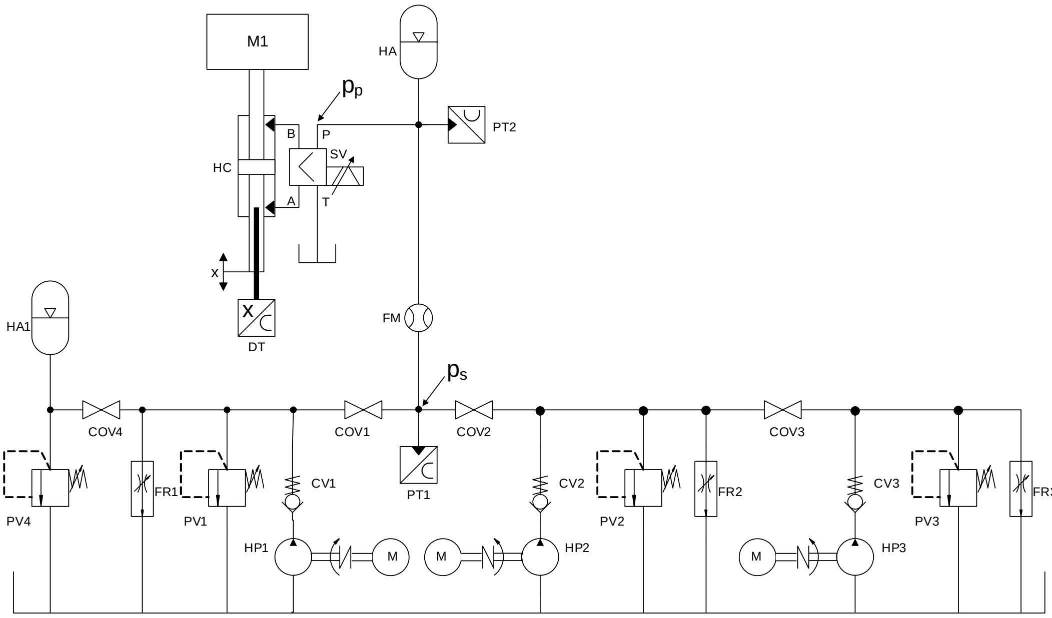

Fig. 2. Functional scheme of the hydraulic power supply loaded by electrohydraulic servodrive: HC – hydraulic cylinder, HA and HA1 – gas-loaded accumulators, HP1, HP2 and HP3 – hydraulic pumps, PV1, PV2, PV3 and PV4 – relief valves, FR1, FR2 and FR3 – flow regulators, SV – electrohydraulic servovalve, P, A, B and T – servovalve ports, CV1, CV2 and CV3 – check valves, COV1, COV2, COV3 and COV4 – shut-off valves, DT – displacement transducer, PT1 and PT2 – pressure transducers, FM – flow transducer, M – electrical motors, M1 – load, p_s and p_p – pressures, x – piston displacement

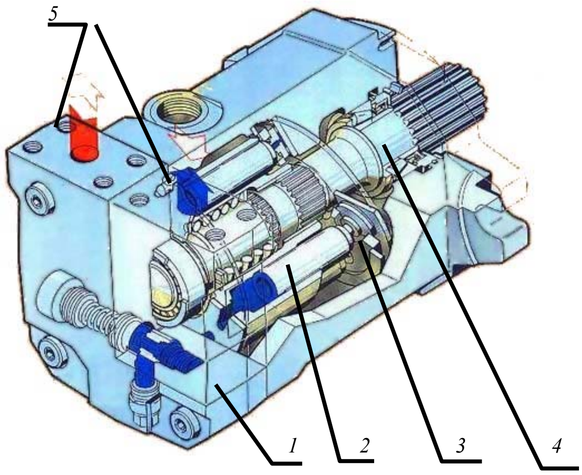

Fig. 3. Fixed displacement piston pump section: 1 - housing, 2 - piston, 3 - swash plate,4 - shaft, 5 - inlet and outlet.

Laboratory tests

Transient states of hydraulic power supply working come mainly from absorptivity changes of its receiver, controlled in the presented system by servovalve SV. As the result of absorptivity changes there is pressure relief valves interaction, where changing flow rate is directed and determined by pump (or pumps) output flow rate surplus. In extreme cases pressure relief valve is completely closed or lets in the whole flow current generated by pump. Construction features (e.g. valve head mass, spring stiffness, viscosity dumper parameters) strongly influence pressure value stabilised by pressure relief valve. Many experiments were carried out, where working receiver parameters and its loading, amount of working pumps and pressure relief valves, set of relief valve opening pressure, together with accumulator where changed. During the researches pressure ps was measured on the outlet of the hydraulic power supply (transducer PT1) and pp on the inlet of the servovalve SV (transducer PT2) as well as piston displacement x of the cylinder HC (transducer DT). Giving suitably chosen piston displacement course from the control system above-mentioned physical quantities were registered. At first triangle control signal with suitable frequencies and amplitudes selection has been chosen. Registered and presented in Fig. 4 charts show hydraulic power supply response to the disturbance of impulse character. This disturbance appears at the moment of piston velocity sign changes, what is visible in the pressure diagrams.

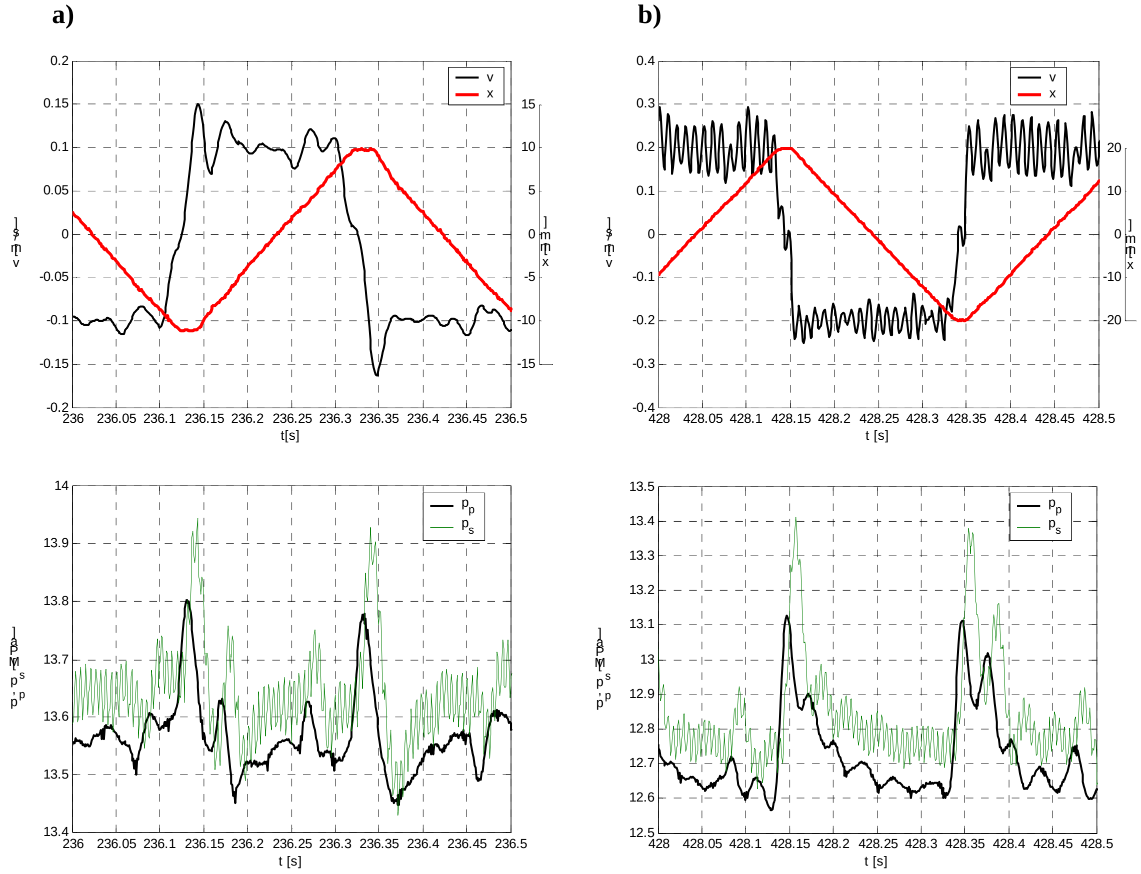

Fig. 4. Time diagram of the piston velocity v and displacement x as well as pressure p_s and p_p for triangle control signal with frequency 2.5 Hz and amplitudes: a) 10 mm, b) 20 mm.

During this laboratory test pump HP1 with the efficiency of 16 $l.min^{-1}$ was working and pressure relief valve PV1 was set to opening pressure 14 MPa (with flow rate 16 $l.min^{-1}$). Lumped mass M1 was 25 kg. Presented results in Fig. 4a concern the situation, where servodrive was controlled by triangle signal with frequency 2.5 Hz and amplitude 10 mm. Piston was moving with the mean velocity 0.1 $m.s^{-1}$ what corresponds to absorptivity 6.92 $l.min^{-1}$. For such defined cylinder motion parameters, mean pressure value measured in the outlet of the hydraulic power supply was 13.64 MPa, and in the inlet of servovalve 13.57 MPa. Range of the pressure value changes measured on hydraulic power supply and servovalve were corresponding to the values 0.56 MPa and 0.41 MPa. During the second experiment piston motion amplitude was changed to 20 mm. Piston was moving with the velocity of 0.2 $m.s^{-1}$ (Fig. 4b) what corresponds to absorptivity 13.84 $1.min^{-1}$. For such defined cylinder motion parameters, mean pressure value measured in the outlet of the hydraulic power supply was 12.84 MPa, and in the inlet of servovalve 12.73 MPa.

Range of the pressure value changes measured on hydraulic power supply and servovalve were corresponding to the values 0.88 MPa and 0.64 MPa. Pressure courses registered using transducer PT1 include component represented by pressure pulsation with amplitude about 0.1 MPa and frequency 175 Hz. It is the feature characterised piston pumps operation used in hydraulic power supply. This pulsation is not visible in the diagrams of pressures measured by transducer PT2. Elastic hydraulic pipe connecting power supply with receiver is working as a low pass filter. Where velocity sign is changing, rapid pressure increase appears what is caused by receiver absorptivity decrease, reached as the effect of servovalve SV control. Observed response of the system indicates that investigated hydraulic power supply possesses the features of oscillatory object, connected with dynamic features pressure relief valve. Because of the fact that dumping ratio of the valve head is rather small, step or impulse flow rate change in this valve causes, that valve head achieved equilibrium position in an oscillation way. From the schema presented in Figure 2 follows, that value of this dumping ratio depends on working point of the relief valve. Increasing amplitude or frequency of piston motion leads to the decrease of power supply line mean pressure value. For velocity 0.1 $m.s^{-1}$ supply pressure drop of 2.6 % was observed. When velocity increases twice it leads to pressure drop of 8.3 %. Further velocity increasing lead to even higher supply pressure drop.

On the Figure 5 time diagrams of the piston displacement and velocity of the cylinder HC have been presented as well as pressures p_s and p_p registered in case the servodrive was controlled by triangle control signal with frequency 2.5 Hz and amplitude 30 mm. In that case the system does not achieve established amplitude because absorptivity of the servodrive exceeds output flow of the hydraulic power supply. The main symptom of such situation is supply pressure drop because pump is working with no sufficient output flow stream in relation to absorptivity of the system. In such situation the whole output flow steam of the supply is leading to the servodrive and through relief valve there is no working fluid flow. This valve stops fulfilling a pressure-stabilisation function. Lowered supply pressure cause shut off the relief valve. Since that moment this valve has not taken part in operation of hydraulic system. As a result of such situation hydraulic supply starts working as a flexible source of pressure that characterizes in this way that value of supply pressure adjusts to the actual load of hydraulic cylinder HC. It could be observed that in situation of supply output flow lack amplitude of pressure changes in relation to their mean value is much bigger than in case of the hydraulic unit work with output flow surplus. Lowered supply pressure value is profitable from energetic point of view but dynamic properties of servodrive become worse.

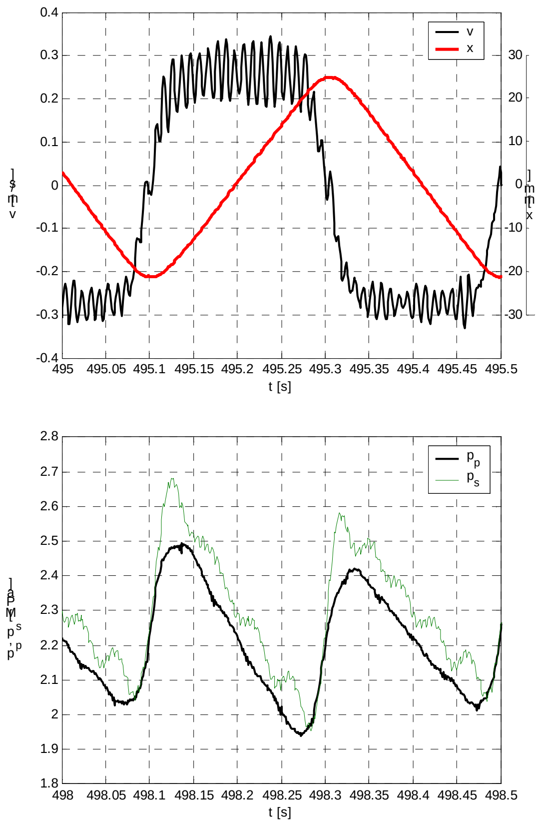

Fig. 5. Time diagram of the piston velocity v *and displacement* x *as well as pressure p_s and p_p for triangle control signal with frequency 2.5 Hz and amplitudes 30 mm*

Next square control signal has been chosen with frequency 2.5 Hz and amplitude 10 mm. In the Fig. 6 time diagrams of appropriate physical quantities registered for two experiments as the response of the system for such control signal have been presented. During first experiment (Fig. 6a) pump HP1 with output flow rate 16 $l.min^{-1}$ was working and pressure relief valve PV1 was set to opening pressure 14 MPa. Lumped mass M1 is 30 kg. During second experiment (Fig. 6b) pumps HP1 and HP2 with joint output flow rate 26 $l.min^{-1}$ and pressure relief valves PV1 and PV2 was both set to opening pressure 14 MPa as well. Pressure changes on the outlet of hydraulic power supply and on the inlet of the servovalve during such experiments are much greater then in case where the triangle control signal has been used. Pressure values registered on the power unit and on the servovalve changes about 3.5 ÷ 4 MPa when the system was supplying from one pump and about 2.5 ÷ 3 MPa when the system was supplying from two pumps. Output flow rate of the hydraulic power unit increased about 60 % as a result of turning on the second pump caused increase of servodrive piston maximum velocity above five times.

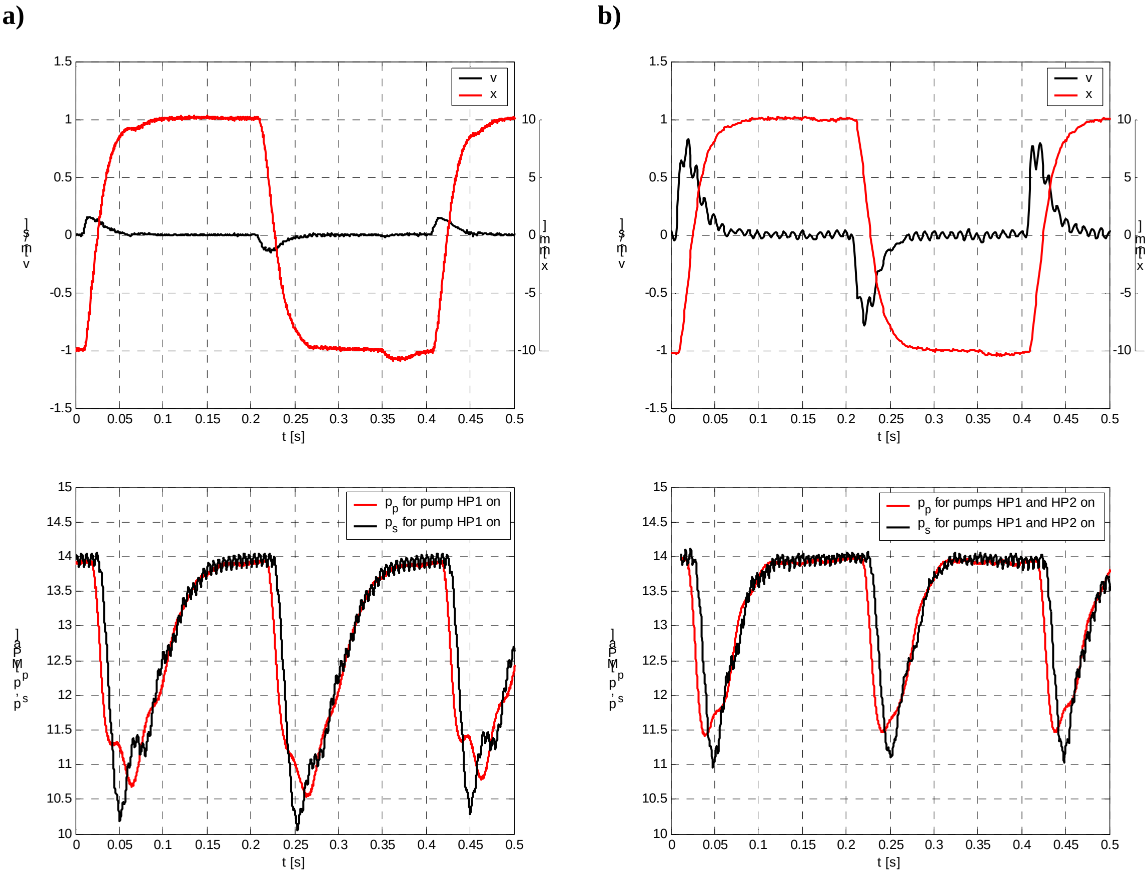

Fig. 6. Time diagram of the piston velocity v *and displacement* x *as well as pressure* p_s *and* p_p *for square control signal with frequency 2.5 Hz and amplitude 10 mm for: a) one hydraulic pump HP1 on; b) two hydraulic pumps HP1 and HP2 on.*

Conclusion

Described in the paper laboratory researches of hydraulic power unit, cooperated with electrohydraulic mechanical vibrations excitator, which is characterised by dynamic absorptivity changes, enable to get interesting results. Obtained results of research indicate clear mutual interactive of those systems. Investigated hydraulic power unit is not always stiff source of constant pressure in dynamic states of operation. For some working parameters hydraulic power supply shows considerable susceptibility to hanges in working conditions. Next the disturbances in supply pressure courses influence into mapping accuracy of control signal shape and dynamic properties of hydraulic receiver. The influence to the supply pressure value possesses among others surplus of the hydraulic power supply output flow, shape and parameters of a control signal used for hydraulic receiver steering, static and dynamic characteristics of the pressure relief valve, cooperation with hydraulic accumulator, parameters of hydraulic supply pipe. Results of laboratory researches lead to the conclusions how hydraulic throttling control systems should be modelled. Simulations of such systems are usually carried out for mathematical models assuming constant pressure supply. It may lead to important discrepancies between results form simulations and real courses of appropriate physical quantities. Therefore for hydraulic throttling control systems working under dynamic states stiff pressure supply source model can not be sufficient. More appropriate and universal, even though more complicated, would be mathematical model of susceptible source of hydraulic pressure.

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