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SHAPING THE STARTING OF A HYDROSTATIC TRANSMISSION WITH PROPORTIONALLY CONTROLLED ELEMENTS

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Abstract: *This paper focuses on selected adverse phenomena accompanying the starting of a hydrostatic transmission with a linear motor. Some practical ways of attenuating the process of starting the hydrostatic transmission are presented. A hydraulic system, being in a transient process state and incorporating a proportional control valve, was analysed. The aim of the investigations was to reduce dynamic pressure excesses and to determine the effect of this factor on the noisiness of the hydrostatic system in the transient state.*

Keywords: Hydrostatic transmission, Starting, Pressure excess, Noise.

1. Introduction

The current trends in the development of hydrostatic drives consist in, among other things, minimizing energy losses and weight and increasing the transmitted power, and so increasing the power/weight ratio. One of the limitations on improving this ratio is the fact that the noisiness of hydrostatic systems and elements increases as the generated or transmitted power is increased. It emerges from the available literature and the authors' own experiments that one of the principal causes of the excessive noisiness of hydraulic systems are pressure fluctuations due to flow rate oscillations and pressure excesses during hydraulic system starting or braking.

The starting process (pressure buildup) can be attenuated by introducing additional leakages through the use a starting valve (e.g. an adjustable throttle valve or a specially designed starting valve). But the incorporation of a throttle valve between the hydraulic motor's feed conduit and its drain conduit to increase leakages results in lower efficiency, whereby it is necessary to install greater power for the performance of the drive function.

The starting process can also be attenuated by introducing proportional control valves.

2. Tested system

The initial system shown in Fig. 1a was adopted to examine the possibilities of the effective attenuation of hydraulic transmission starting. During the experiments special attention was given to the process of starting the hydraulic system. In order to attenuate this process the structure of the system was modified by introducing a proportional distributor into the pump shunt conduit, Fig. 1b.

3. Measuring apparatus and methodology

The measured quantities were:

1) pressure (points 6 and 7 (Figs 1a and 1b)), using SPAIS PT5101 extensometric pressure transducers with a measuring range of 0-6 MPa with an SPAIS AT-5210N instrumentation amplifier and an SPAIS BZ5202 power supply unit;

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2) the equivalent level (L_{Aeq} [dB]), the maximum level (L_{Amax} [dB]) and the minimum level (L_{Amin} [dB]) of sound A, using a sonometer and a type 2250, series 2519832 modular B&K sound level meter with a type ZC 0032 preamplifier (serial number 4112), a type 4189, series 2519832 microphone and BZ 5503 software;
3) the displacement of the proportional distributor, using an induction displacement sensor integrated with the proportional distributor.

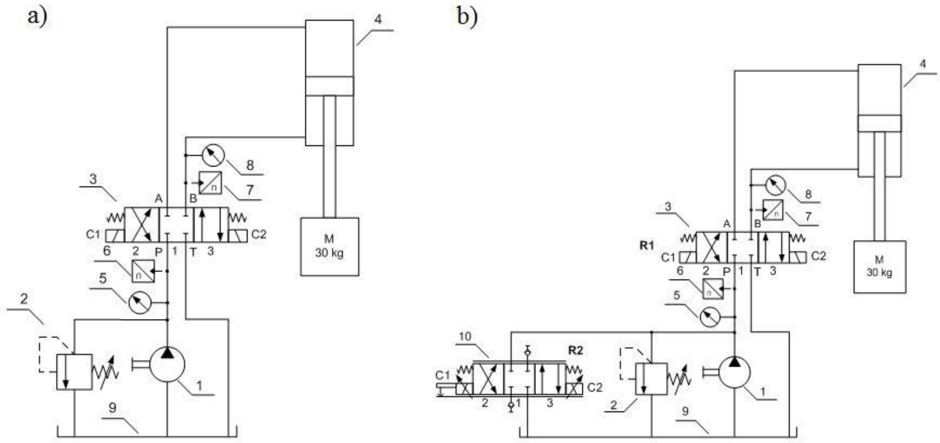

Fig. 1: a) diagram of hydraulic system of hydrostatic transmission with linear motor: 1 – displacement pump (Q_{pt} = 6.5 dm^3/min at 1450 rpm); 2 – pressure relief valve; 3 – conventionally electrically controlled 4/3 distributor; 4 – linear hydraulic motor (piston diameter 25 mm, piston rod diameter 16 mm, stroke 200 mm); 5, 8 – manometer; 6, 7 – pressure transducer; 9 – hydraulic oil tank; b) diagram of modified hydraulic system of hydrostatic transmission with linear motor: 1 – displacement pump (Q_{pt} = 6,5 dm^3/min at 1450 rpm); 2 –relief valve; 3 – conventionally electrically controlled 4/3 distributor; 4 – linear hydraulic system (piston diameter 25 mm, piston rod diameter 16 mm, stroke 200 mm); 5, 8 – manometer; 6, 7 – pressure transducer; 9 – hydraulic oil tank; 10 – 4/3 proportional distributor.

The signal from the pressure measuring circuit was fed into a four-channel digital oscilloscope with an FFT analysis module (Tektronix TDS 224) and then into a measuring computer. The signal from the sonometer was transmitted directly to the measuring computer. The sound level meter can simultaneously record the time history and perform a frequency analysis. During the experiments the sonometer was situated at a distance of 1 m from the source. The whole measuring circuit was calibrated before and after the measurements.

The hydraulic system schematically shown in Fig. 1a and b was subjected to tests. As the system shown in Fig. 1a was being tested, coil C1 of the 4/3 distributor was surgewise (in accordance with the step function) energized at instant t_0. This resulted in the abrupt connection of the actuator's piston rod chamber with the operating hydraulic feeder, forced by an abrupt change in the rate of flow. The pressure (in points 6 and 7 acc. to Fig. 1a) and the accompanying acoustic signal were recorded. As the system shown in Fig. 1b was being tested, distributor 10 was overdriven. Distributor 2 was constantly in position 2, while distributor 10 was in initial position 2. In this case, the feature of proportional control valves, enabling one to gradually increase or decrease the electrical signal fed into the coil of the proportional electromagnet, was exploited. This feature is referred to as "*Rampenzeit*" in the German-language literature and as "*time of delay*" in the English-language literature on the subject. As a result, the proportional distributor could be gradually moved from position 2 towards neutral position 1.

4. Test results

Traces of: the pressure at the pump, the proportional distributor slide displacement and the sound level during the starting of the investigated hydraulic systems were obtained from the tests.

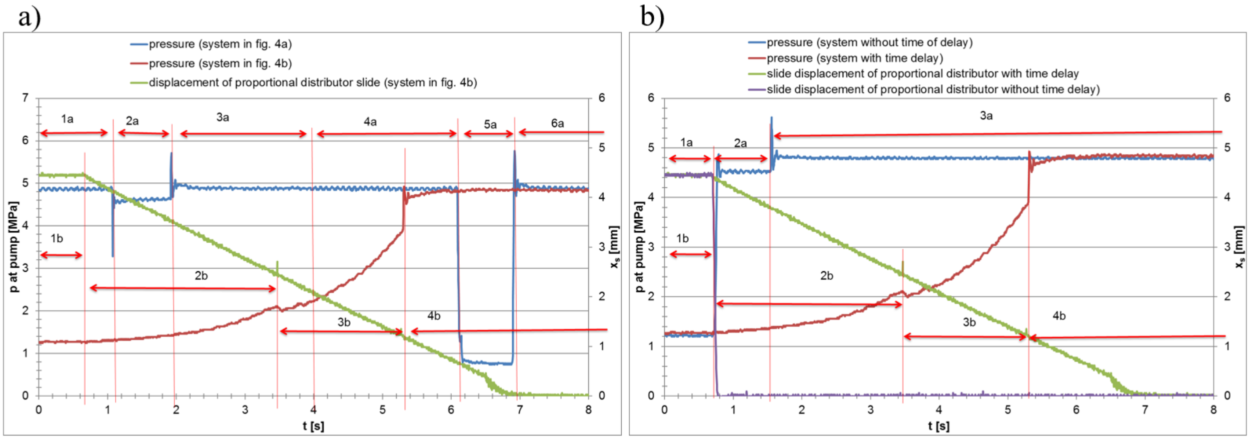

Fig. 2: Pressure at pump and slide displacement traces for tested hydraulic systems.

The time intervals corresponding to the working phases of the systems are marked in Figs. 2a and 2b. In Fig. 2 one can distinguish the following time intervals: interval 1a – the coils of distributor 3 (Fig. 1a) are not powered and the liquid fed by the pump returns via pressure relief valve 2 to the tank. At the end of this interval coil C1 of distributor 3 (Fig. 1a) was surgewise energized (24 V), whereby the latter was abruptly overdriven, the actuator (the piston rod chamber) was connected to the supply and actuator 4 was started. Interval 2a corresponds to the movement of actuator 4, which subsequently reaches its upper extreme position and stops abruptly (the piston strikes against the bottom). Interval 3a corresponds to the situation when the actuator remains in the upper extreme position and is connected with the feed pump; this is also the working phase of the relief valve. At the end of this interval the electric circuit of coil C1 of distributor 3 was opened and the distributor slide returned to the neutral position, while the pump continued pumping the whole liquid via over overflow valve 2 to the tank. This state lasted over interval 4a. Then by surgewise energizing coil C2 distributor 3 was overdriven into the other extreme position. Interval 5a corresponds to the return motion (downwards) of the actuator, which performing its full stroke, reaches the extreme position and stops abruptly. The pressure increases and the pump again begins to return the liquid to the tank via relief valve 2 – interval 6a. A similar analysis of time intervals can carried out for the modified system shown in Fig. 1b. Interval 1b corresponds to the situation when proportional distributor 10 is overdriven into extreme position 2 and the whole pumped liquid finds it way, via this distributor, to the tank (relief valve 2 remains closed). Interval 2b corresponds to the movement (using the time of delay) of the slide of distributor 10 from position 2 to (neutral) position 1. As a result, the throttle gap of distributor 10 diminishes and pressure slowly builds up, while the actuator remains motionless (occupying the lower extreme position). It starts to move at the beginning of interval 3b. The gentle movement of the actuator continues to the end of this interval. Subsequently, the actuator stops abruptly as it reaches its upper extreme position. Then pressure in the system rises to the relief valve opening value. This corresponds to interval 4b in Fig. 2a. In a similar way one can describe Fig. 2b, where the systems from Fig. 2b, in which the slide was overdriven with and without the time-of-delay function turned on. Interval 1a corresponds to the situation when proportional distributor 10 is overdriven to extreme position 2 and the whole pumped liquid passes through it to the tank (relief valve 2 remains closed). Interval 2a corresponds to the movement of actuator 4. Then the actuator reaches its upper extreme position and stops abruptly (the piston strikes against the bottom). Interval 3a corresponds to the situation in which the actuator remains in the upper extreme position and is connected with the feed pump. This is also the working phase of the relief valve. Time intervals 1b – 4b in Fig. 2b are analogous to the ones in Fig. 2a.

In order to determine the effect of the particular starting methods on the acoustic climate the measured acoustic quantities were compared for the actuator's two working phases: standstill and stopping (Tab. 1). In the case of phase 1, i.e. cutting off hydraulic supply to the actuator and directing the working medium from the pump via the relief valve or the proportional distributor to the tank, the acoustic parameter values measured for the considered systems are comparable, amounting to 62 dB. In both cases, the character of the spectra is similar in most of the thirds, with characteristic broadband noise in the range of average and high frequencies and with a tonal component at 500 Hz. Larger differences are observed in the third with the midband frequency of 250 Hz and 1 kHz, which in the case of proportional control have a tonal character. The tones are present also in the other working cycles of the system with proportional control. This means that they appear when the proportional distributor is incorporated into the system. As the impedance of the system changes, pressure and dynamic pump shaft torque fluctuations increase to 250 Hz. The next displacement pump pressure fluctuation harmonics are 500 Hz and 1 kHz. No dominant components were observed in the range of harmful infrasounds.

Tab. 1: Comparison of measured acoustic quantities.

	Type of control					
	conventional		**proportional without time of delay**		**proportional with time of delay**	
Cycle	**denotation**	**L_{AFmax} [dB]**	**denotation**	**L_{AFmax} [dB]**	**denotation**	**L_{AFmax} [dB]**
Standstill	1a	61.8	1b	62.2	1b	62.2
Stopping	3a	82.4	3a	83.5	4b	80.6

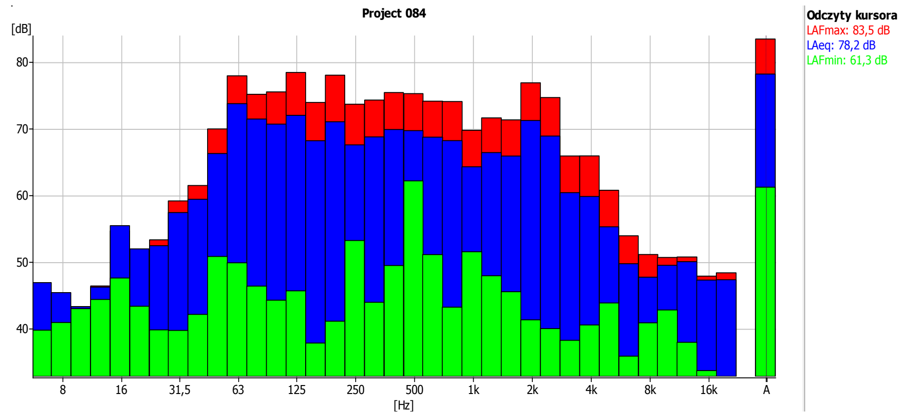

Fig. 3: One-third octave spectrum of average LAeq, maximum LAFmax and minimum LAFmin level of sound A during actuator stopping for proportional-control system without time of delay (stage 4b in Fig. 2b and stage 3a in Fig. 2b).

In the case of both conventional control and proportional control, the acoustic signal recorded for the actuator stopping cycle, without piston final run braking, has a nonstationary character. Considerable differences between the maximum level and the minimum level are observed in a broad band in the 1/3 octave spectrum (Fig. 3). This indicates the occurrence of pulse-like noise produced as the piston strikes against the cylinder bottom. The lower values (2 – 3 dB) of the level of sound A measured for proportional control with time of delay are due to the incomplete overdrive of the slide in the proportional distributor.

5. Conclusion

Dynamic changes in pressure, especially dynamic pressure excesses arising during transient states (e.g. during starting or braking), are to a large extent responsible for the noisy operation of hydraulic systems. Dynamic pressure excesses also result in instantaneous impact or pulse loads acting on hydraulic system elements, contributing to the excitation of vibrations in the elements and to the degradation of their durability.

The adverse effects of dynamic pressure changes (pressure fluctuations, dynamic pressure excesses during the starting or braking of the hydraulic system) can be most effectively reduced through the simultaneous use of several methods, which often have a selective character, i.e. are effective in selected frequency ranges. The results of the experiments show that the dynamic pressure excess during the starting of the investigated transmission was reduced thanks to the use a proportional distributor with the time-of-delay function (Figs. 2a and 2b). However, this is achieved at the cost of longer starting process time. The results for the extreme cases – without time of delay and with the maximum time of delay (amounting to about 6 s) – are reported intentionally, however, the control card of the proportional distributor is equipped with a potentiometer enabling the infinitely variable setting of this time up to $t_{rmax} = 6$ s.