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CONTROL SYSTEM OF THE HYDRAULIC CYLINDERS SYNCHRONIZATION WITH THE USE OF ARITHMETIC MEAN OF THEIR POSITIONS

SUMMARY

The aim of the control system is to ensure the effective synchronization of the cylinders motion and simultaneous minimization of volume loss in the throttling hydraulic systems. In the proposed control system, arithmetic mean of the synchronized cylinders displacements is the command value to the control subsystems where identical controllers are used. The number of the control subsystems is equal to the number of the synchronized cylinders. Developed control system enables effective motion synchronization of arbitrary number of the hydraulic cylinders with the pressure of the working liquid resulting from the current cylinders load.

***Keywords**: control system, controller, motion synchronization, hydraulic cylinder*

UKŁAD STEROWANIA SYNCHRONIZACJĄ SIŁOWNIKÓW HYDRAULICZNYCH Z WYKORZYSTANIEM ŚREDNIEJ ARYTMETYCZNEJ ICH POŁOŻEŃ

Celem układu sterowania jest zapewnienie efektywnej synchronizacji ruchu siłowników wraz z minimalizacją strukturalnych strat objętościowych, jakie występują w dławieniowych układach hydraulicznych. W proponowanym układzie sterowania, średnia arytmetyczna przemieszczeń synchronizowanych siłowników jest wartością zadaną do podsystemów sterowania, w których zastosowano identyczne regulatory. Liczba podsystemów sterowania jest równa liczbie synchronizowanych siłowników. Opracowany układ sterowania umożliwia efektywną synchronizację ruchu dowolnej liczby siłowników hydraulicznych przy ciśnieniu cieczy roboczej wynikającym z bieżącego obciążenia siłowników.

***Słowa kluczowe**: układ sterowania, regulator, synchronizacja ruchu, siłownik hydrauliczny*

1. INTRODUCTION

Throttling systems for the hydraulic cylinders motion synchronization are most frequently used in the industry. The control of such systems is based on the idea that one of the cylinders functions as the leading assembly and the rest as following assemblies (Stefański 1999, Sikora 2004).

The aim of the first system with the leading cylinder (Fig. 1) is to accomplish in the possibly shortest time the set position with the minimal static deviation, while the aim of the control system with the following cylinder is to track the leading cylinder with the minimal dynamic and static deviation. It is of an utmost importance to highlight that the system with the leading cylinder is fixed during the control system design process and it does not change during the cylinders motion synchronization. Such a structure of the control system requires an operation of the synchronization system with the working liquid pressure allowing to overcome the highest potential working load of any cylinders which may appear. Therefore, “the capacity reserve” of the feed pump needs to be maintained which allows to increase the flow through the hydraulic control valve of any arbitrary set in the case of the load increase in that set. This means that during standard work the hydraulic valves cannot be completely open. “The capacity reserve” is uselessly carried away to the tank which generate volume losses in the system.

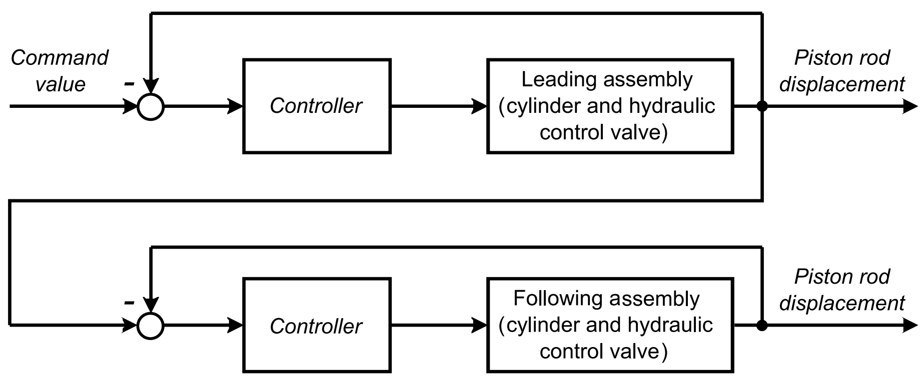

Fig. 1. Schematic diagram of the control system with the constant leading cylinder (Stefański 1999)

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The volume structural losses may be decreased in some range of applications in which the speed values is not important but there is a need to maintain the same speed to all synchronized cylinders. The speed value in these applications is the feed pump capacity and the surface of the piston in the cylinders dependent. Various loads of each cylinder and such phenomena as leaks affect the speed. The proposed solution is to set a displacement value which is obtainable by the piston rod of each cylinder with the liquid pressure resulting from current load of all cylinders. Therefore, it must be conditioned by the possible displacement of the most overloaded cylinder or a cylinder in which other phenomena such as leaks are taking place. These phenomena result in a decrease in speed. The signal control based on value variable of the displacement, causes the closure of the hydraulic valves of cylinders whose pistons have higher speed, and the opening of the valves of cylinders whose pistons have lower speed. Therefore, the cylinders move with the speed resulting from the feed pump capacity. The aim of control is to divide the pump capacity in a way that the piston positions in any given moment are the same. In this way the continuous draining of liquid to the tank through the relief valve is eliminated.

However, the hydraulic cylinders motion synchronization system becomes then the object consisting of the mutual coupled control system of the synchronized cylinders. Control of such object is difficult because of the cross-coupling among the control system of each cylinders. In the literature the problem of the cross-coupling in such systems is solved by implementing an additional controller whose aim is to minimize the difference between the displacement of the cylinders by generating additional ingredient signals steering individual cylinders (Chen 2008) or by the modification of the command value (Vasiliu 2004). An exemplary control system is presented in Figure 2.

However, control systems which are proposed in the literature enable the motion synchronization of the two cylinders without the indication of the way in which generalization of these systems may be done for more than two cylinders synchronization. It is important to indicate that for the motion synchronization of two cylinders three regulators are usually implemented. A following question arises, how many controllers are required to synchronize more than two cylinders.

2. PROPOSED CONTROL SYSTEM

2.1. Limitations of the basic quantities

The throttling surface is a function of the slide movement in the valve. The slide movement value results from physical structure of the control valve. The limitation is expressed by the inequality:

$$0 \le y_{zi}(t) \le y_{z\max i} \tag{1}$$

where: $y_{z\max i}$ is the maximal slide's movement in the hydraulic control valves no. i [m], $y_{zi}(t)$ is the slide's movement in the hydraulic control valves no. i [m].

The increment limitations of the movement value of the slide in the hydraulic control valve result from the signal range frequency of the control valve. It is conditioned by the element which moves the slide in the control valve.

The displacement limitations of the piston rods. A displacement of the piston is limited by the length of the cylinder. The limitation is expressed by the inequality:

$$0 \le y_i(t) \le l_c \tag{2}$$

where: $y_i(t)$ is a displacement of the piston rod of the cylinder no. i [m], l_c is the cylinder length [m].

2.2. Structure of the control system

In this structure all the cylinders follow the motion of the virtual leading cylinder, whose motion trajectory is a result of the motion of all synchronized cylinders (Fig. 3). This trajectory is assigned as the arithmetic mean of the piston rods movement (Grzybek, Kowal 2009):

$$y_z(t) = \frac{\sum_{i=1}^{n} y_i(t)}{n} \tag{3}$$

where: n is a number of the synchronized cylinders [–].

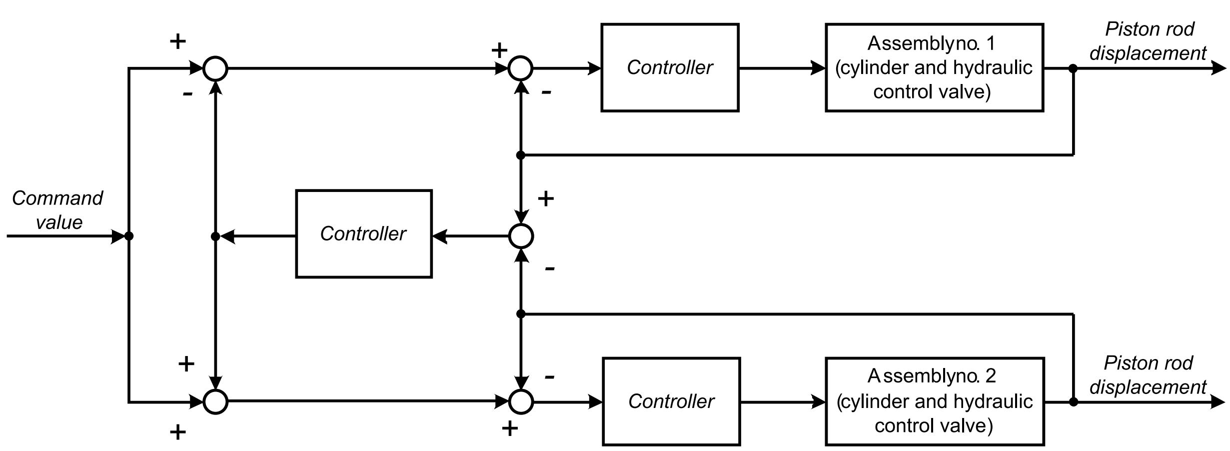

Fig. 2. Schematic diagram of the control system without the leading cylinder (Vasiliu 2004)

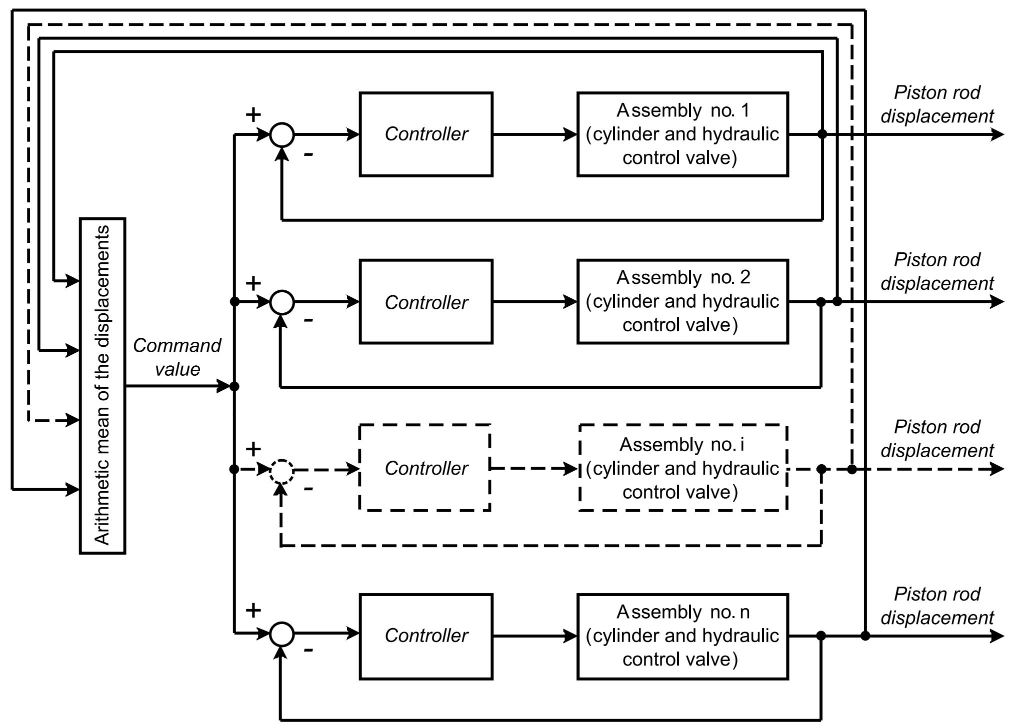

Fig. 3. Schematic diagram of the control system with the arithmetic mean of the synchronized cylinders displacements as the command value

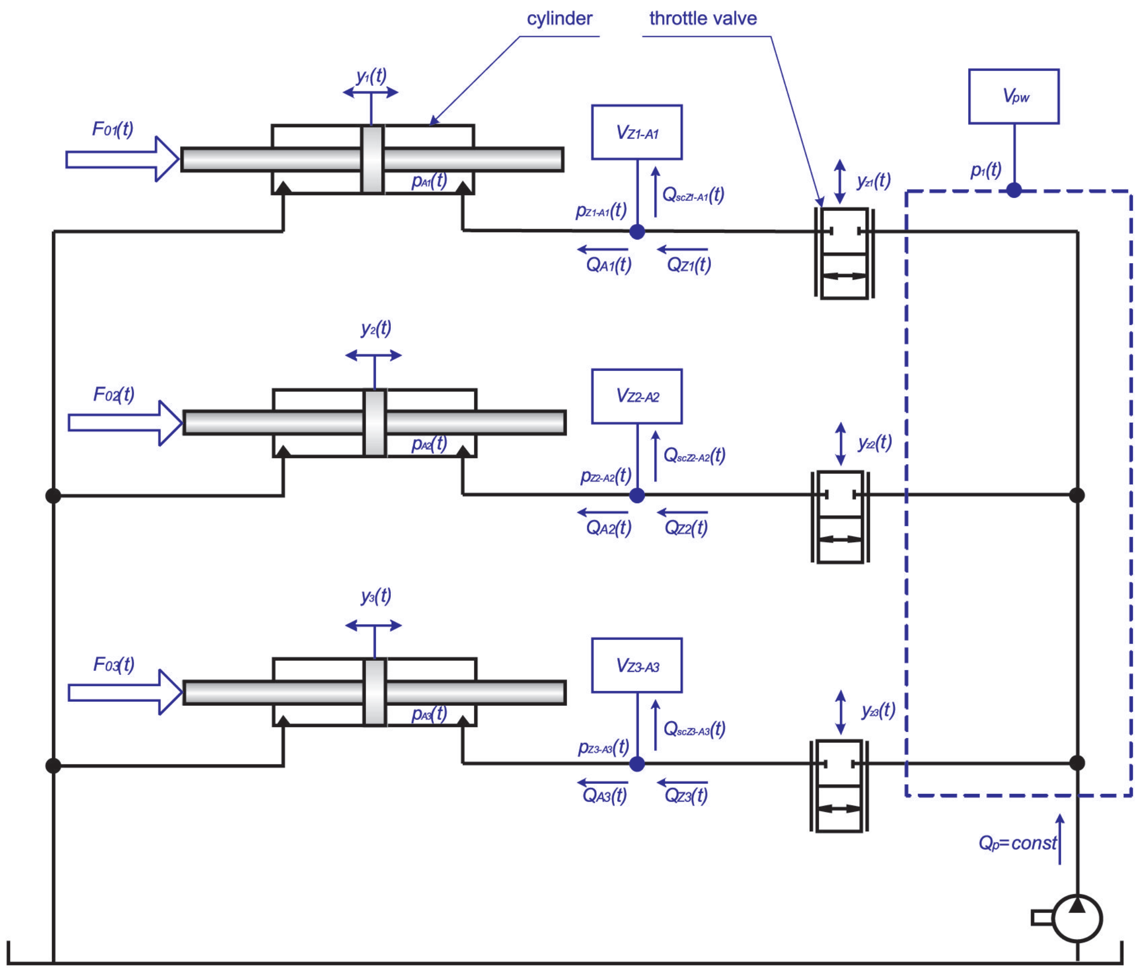

Fig. 4. Computational diagram of the hydraulic structure of synchronization system

3. MATHEMATICAL MODEL OF THE SYNCHRONIZATION SYSTEM

The proposed control system has been tested in the simulation experiment. The applied model of the hydraulic structure of the synchronization system consists of:

- three hydraulic cylinders,
- three throttle valves,
- pump,
- hydraulic conduits.

Computational diagram of the hydraulic structure is presented in Figure 4 [Grzybek, Kowal 2009].

Basic equations describe the throttling synchronization system (Stryczek 1997; Dindorf 2004; Szydelski 1999):

$$\frac{d^2 y_i(t)}{dt^2} = \frac{1}{m_{rs}}\left[A_t p_{Ai}(t) - A_t p_{Bi} - F_{0i}(t) - F_{ti}(t)\right] \tag{4}$$

$$\frac{dp_{Ai}(t)}{dt} = \frac{E_C}{V_0 + A_t y_i(t)}\left(Q_{Ai}(t) - Q_{hi}(t) - Q_{vi}(t)\right) \tag{5}$$

$$Q_{Ai}(t) = \frac{A_{Zi-Ai}}{\rho l_{Zi-Ai}} \int_{t_0}^{t} \left(p_{Zi-Ai}(t) - p_{Ai}(t) - \xi_{RH}\rho \frac{Q^2_{Zi}(t)}{A^2_{RH}} - \lambda_{Zi-Ai} \frac{l_{Zi-Ai}}{d_{Zi-Ai}} \frac{\rho}{2} \frac{Q^2_{Zi}(t)}{A^2_{Zi-Ai}} \right) dt \tag{6}$$

$$\frac{dp_{Zi-Ai}(t)}{dt} = \frac{E_C}{V_{Zi-Ai}}\left(Q_{Zi}(t) - Q_{Ai}(t)\right) \tag{7}$$

$$Q_{Zi}(t) = 2C_d \sqrt{\frac{2}{\rho}} \left(y_{zi}(t)\right)^2 \operatorname{tg}\alpha \sqrt{p_1(t) - p_{Zi-Ai}(t) - \Delta p_{mpw} - \Delta p_{lpm}} \tag{8}$$

$$\frac{dp_1(t)}{dt} = \frac{E_C}{V_{pw}}\left(Q_p - \sum_{i=1}^{n} Q_{Zi}(t) \right) \tag{9}$$

where: F_{0i}(t) is a load of the cylinders no. i [N], $y_i(t)$ is a displacement of the cylinders piston rod no. i [m], $p_{Ai}(t)$ is a pressure of chamber A of cylinder no. i [Pa], $Q_{Ai}(t)$ is a flow to chamber A of cylinder no. i [m^3/s], $Q_{Zi}(t)$ is a flow from the throttle valve no. i [m^3/s], $p_{Zi–Ai}(t)$ is a pressure in conduit between cylinders no. i and throttle valve no. i [Pa], $V_{Zi–Ai}$ is a volume of the conduit between cylinders no. i and throttle valve no. i [m^3], $y_{zi}(t)$ is a slide's displacement in throttle valve no. i [m], $p_1(t)$ is a pressure in the conduits among pump and the throttle valves [Pa], V_{pw} is a volume in the conduits among pump and the throttle valves [m^3], Q_p is the pump capacity [m^3/s], m_{rs} is the mass of the movable elements of cylinder [kg], A_t is the surface of the piston [m^2], $p_{Ai}(t)$ is a pressure of chamber B of cylinder no. i [Pa], $F_{ti}(t)$ is a friction force in the cylinder no. i [N], E_c is an elasticity modulus of the working liquid [Pa], V_0 is an initial volume in chamber A of the cylinder [m^3], $Q_{hi}(t)$ is a liquid flow resulting from absorptivity of the cylinder no. i [m^3/s], $Q_{vi}(t)$ is a liquid flow resulting from leakages in the cylinder no. i [m^3/s], $A_{Zi–Ai}$ is the surface of the conduit between cy-linders no. i and throttle valve no. i [m^2], ρ is a density of the working liquid [kg/m^3], $l_{Zi–Ai}$ is the lenght of the conduit between cylinders no. i and throttle valve no. i [m], ζ_{RH} is a coefficient of the local losses [–], A_{RH} is a surface of the flow through the local obstacle [m^2], $\lambda_{Zi–Ai}$ is a coefficient of the linear losses [–], $d_{Zi–Ai}$ is the diameter of the conduit between cylinders no. i and throttle valve no. i [m], C_d is a flow resistance coefficient [–], α is an angle which depends on the construction of throttle valve [^{o}], Δp_{mpw} is a pressure of local losses [Pa], Δp_{mpw} is a pressure of linear losses [Pa].

4. SIMULATION EXPERIMENTS

4.1. Simulated load of the cylinders

The characteristics of the synchronization error and the movement characteristics of the slides in the hydraulic control valve were determined with the value changes of si-mulated external loads which were treated as disturbances. The simulated values of external loads were commanded according to Figures 5 and 6.

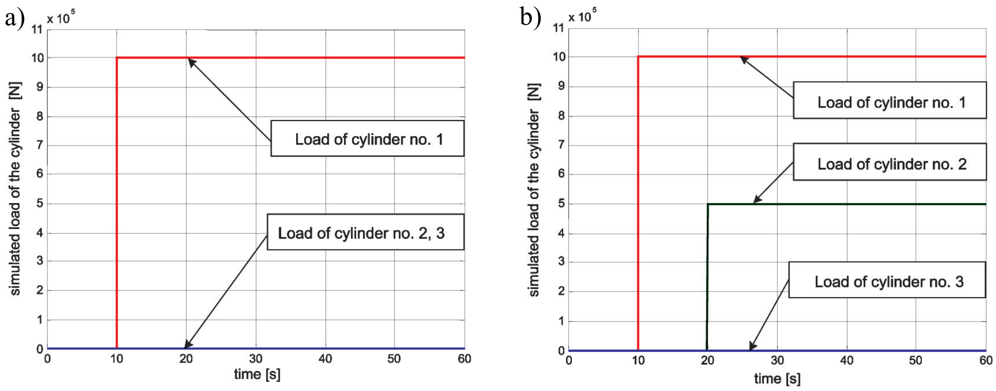

Fig. 5. Step change of the simulated values of external loads: a) for the cylinder no 1; b) for the cylinder no 1 and 2

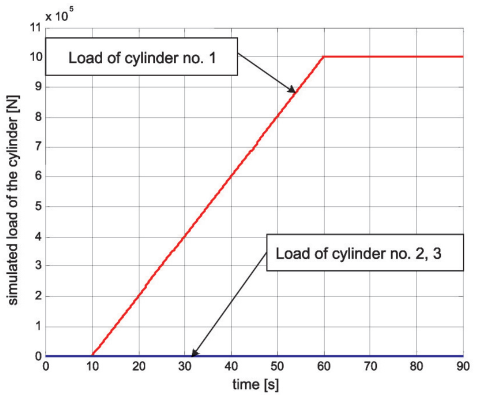

Fig. 6. Linear growing of the simulated values of external load

4.2. Controllers

PID controllers were applied. On the basis of Ziegler-Nichols method controller settings were set: K_p = 1.9 [-], T_i = 2.6 [s], T_d = 0.1 [s]. In each control subsystem identical controllers were applied.

4.3. Results

The characteristics of the simulated synchronization errors are presented in Figures 7a, 8a, 9a and the slide's movement in the hydraulic control valves in Figures 7b, 8b, 9b.

On the basis of these figures, effective synchronization of the cylinders motion is possible for all of the simulated

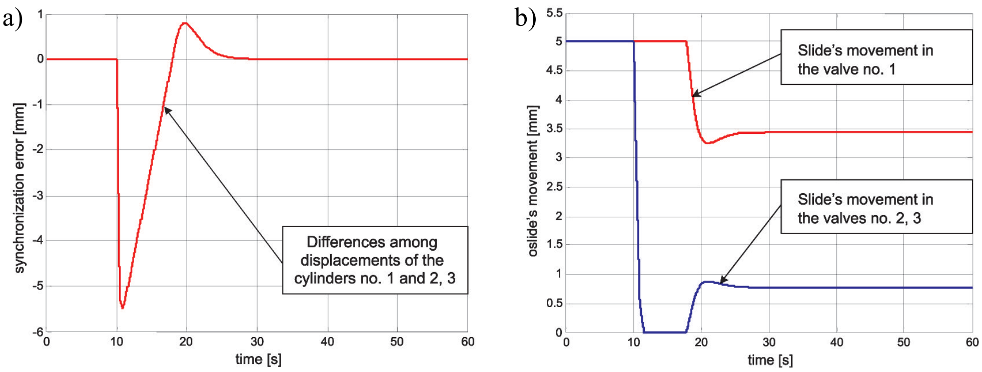

Fig. 7. Response of the system to the step change of the external load of the cylinder no. 1: a) synchronization error; b) slide's movement in the hydraulic control valves

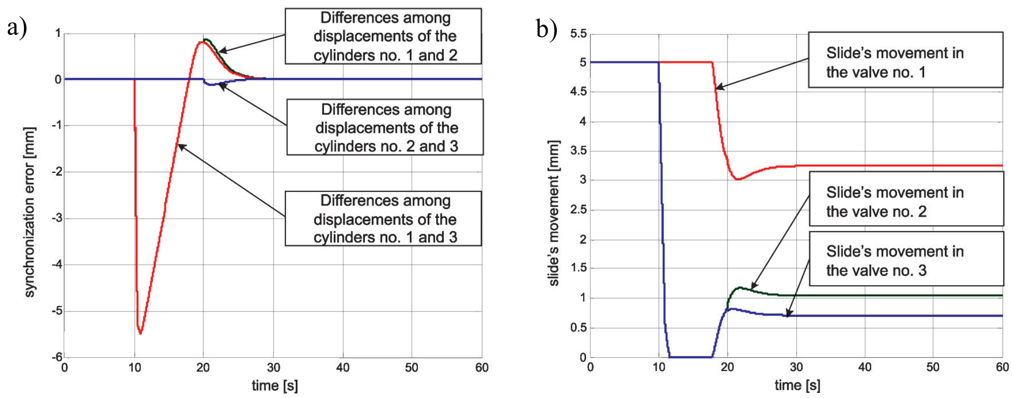

Fig. 8. Response of the system to the step change of the external load of the cylinders no. 1 and 2: a) synchronization error; b) slide's movement in the hydraulic control valves

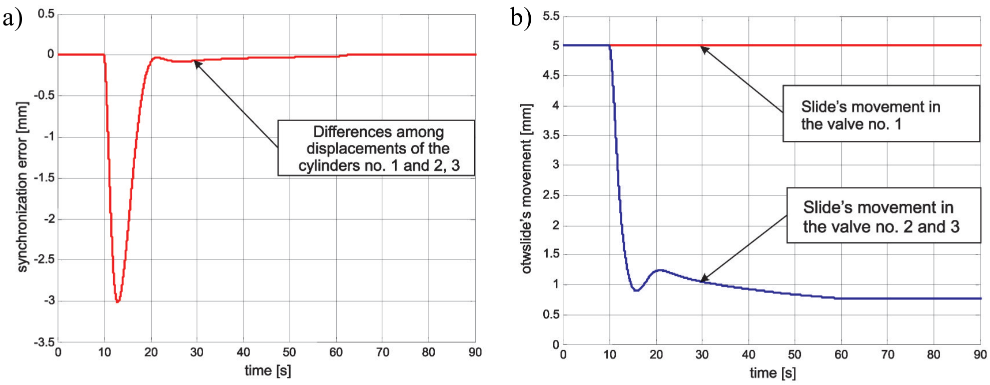

Fig. 9. Response of the system to the linear growing of the external load of the cylinders no. 1: a) synchronization error; b) slide's movement in the hydraulic control valves

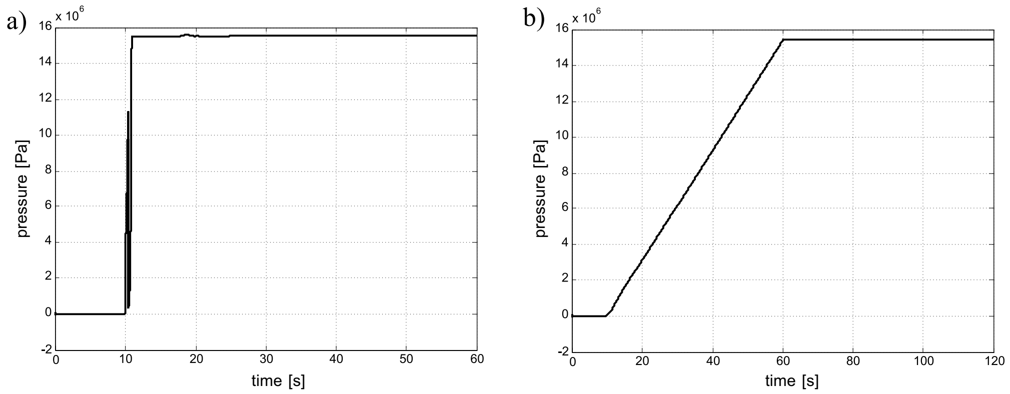

Fig. 10. Working liquid pressure: a) for step change of the simulated values of external loads of the cylinders no. 1 and 2; b) for linear growing of the simulated values of external load of the cylinder no. 1

external loads of the synchronized cylinders. For the simulated linear growing external load, differences among the cylinders displacements is not eliminated (which is visible in Fig. 9a), but its value is contained in the set tolerance and its total value is 0.01 mm. The working liquid pressure (Fig. 10a and 10b) depends on the current load value.

5. CONCLUSIONS

– The effective synchronization of the cylinders motion is possible when the adequate generated command value in the control system is set. This value is determined on the basis of the current state of the synchronization system. It allows to eliminate the volume losses which are the result of the useless draining of liquid to the tank through the relief valve.
– The command value of the cylinder displacement may be calculated as the arithmetic mean of all cylinder displacements. Such a method of the generation of the command value enables to use relatively easy control system in which the number of the control subsystems is equal to the number of the synchronized cylinders. In the particular subsystems the identical regulators can be used.
– Regardless of the load value, the valve that steers the cylinder, which has the higher value of load than the rest of the cylinders, is not completely open. Despite synchronization, it may occur that control valves are almost closed. It may have impact increase the hydraulic losses. Moreover, it can generate the risk of the flow restrain by one or many valves caused by the obliteration phenomena. That is why the next stage of the expanding of the control system is to elaborate the system that will enable the effective motion synchronization with the highest sum of the throttling surfaces in all valves.

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