MODELLING OF THE CRANE JIB LOWERING SYSTEM WITH THE FUZZY LOGIC CONTROLLER

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Abstract: In this work application of the fuzzy logic controller in the hydraulic lowering system of the self-propelled crane jib, with a proportional relief valve used as a counterbalance valve is presented. First, the mathematical model of the hydraulic system was prepared, and then simulations were carried out, using the proper computer program. The fuzzy logic controller was applied as the pressure regulator in the supply line of the hydraulic system. Application of the fuzzy controller gave satisfactory results of the control process.

Keywords: self-propelled crane, crane jib lowering system, proportional relief valve, fuzzy logic controller, computer simulation program

1. INTRODUCTION

Application of the elements controlled with the aid of the digital-circuit engineering allows to increase the precision of the machines work motion. It is indispensable to apply the proper digital controllers in these systems. In this paper a digital simulation method was used to examine the fuzzy logic controller. Simulations were carried out using the proper computer program. The controller operated the force of the electromagnet in the proportional relief valve. The valve was installed in the control system of the crane jib lowering velocity.

2. MATHEMATICAL MODEL

2.1 Diagram of the hydraulic system

The diagram of the crane jib lowering system with a proportional relief valve used as a counterbalance valve is presented in Figure 1. A proportional relief valve was taken into consideration during building the mathematical model (figure 2).

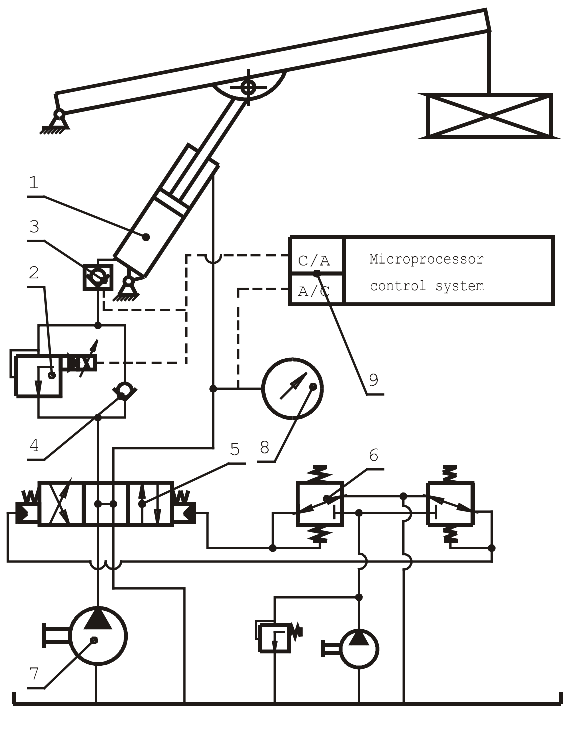

Figure 1. Diagram of the crane jib lowering system with a proportional relief valve : 1-hydraulic cylinder, 2-proportional relief valve, 3-hydraulic lock, 4-check valve, 5-control valve, 6-hydraulic controllers, 7-pump, 8-pressure transducer, 9-converters and microprocessor system

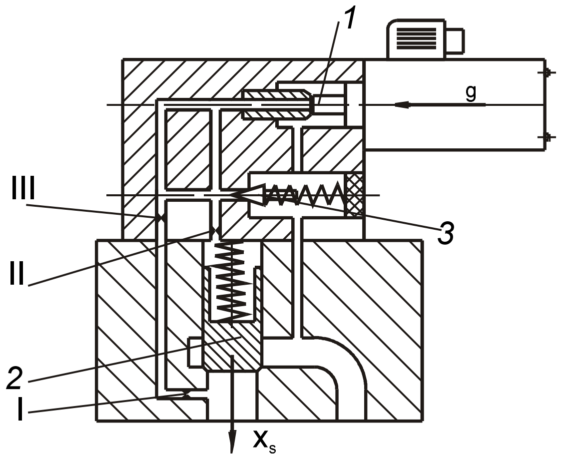

Figure 2. Proportional relief valve : 1 - pilot, 2 - slide, 3 – poppet valve, I, II, III – throttle nozzles

2.2 Crane jib geometrical dimensions

Geometrical dimensions of the crane were taken from the self-propelled crane DST-01841 produced by FAMABA Głogów (Figure 3).

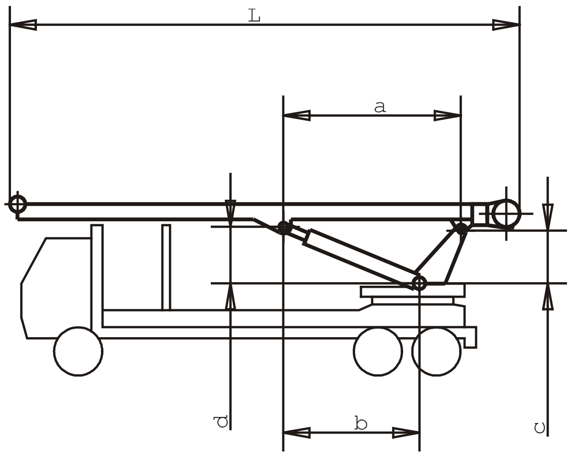

Figure 3. Geometrical dimensions of the self-propelled crane

2.3 Notations

For the mathematical description of the slide and the pilot motion in the relief valve, and the hydraulic cylinder, an uniform notations with indexes were used. For the relief valve „s" index is used for parameters connected with the slide of the valve, „g" index is connected with the valve head, and hydraulic cylinder parameters are indexed with „cyl".

m_s, m_g, m_{cyl},- mass,

x_s, x_g, x_{cyl}, - displacement,

x_{sw}- preliminary spring tension,

α_s, α_g, α_{cyl},α- resistance of motion coefficient,

α_{fs}, α_{fg}, - jet angle,

c_s,, - coefficient of stiffness,

S_s, S_g, S_b, S_c – area of cross-section of the slide and the valve head,

S_{d1}, S_{d2}, S_{d3} - areas of flow sections appropriately for throttle nozzles I, II and III,

S_{cyl}, S_{cylt} – area of the active face of the hydraulic cylinder,

S_{pcyl} – area of the service line of the hydraulic cylinder,

μ_s, μ_g, μ_{cyl} – coefficients of the flow rate,

D_s, D_g - diameters of the sockets,

R_s, R_g – coefficient of restitution,

V_1, V_w, V_s ,V_z,V_g,V_{cyl} – volumes of working chambers,

D_r, D_k, - control valve slide diameter, control valve inlet channel diameter,

Q_I, Q_{II}, Q_{III} – volumetric flow rate appropriately for nozzles I,II,III,

Q_s, Q_g - volumetric flow rate by the gap,

p_1, p_s, p_g, p_w, p_{cyl} , p_z – pressures in chambers: V_1, V_s, V_g,V_w, V_{cyl}, V_z,

p_{zl} – pressure in the sink line,

p_{zad}, – expected pressure in the supply line,

p_{zm} – measured pressure in the supply line,

p_0 – starting pressure,

Δp, Δp_{max}– difference between expected and measured value of the supply pressure, maximum allowable value of the difference,

ρ - liquid density,

B_1, B_s, B_g, B_w, B_{cyl} - substitute coefficients of bulk modulus in corresponding volumes,

F_{obc}, F_{el}, Fel_0- load of the hydraulic cylinder, electromagnet force, starting value of the electromagnet force,

Δt_{pocz}, t_{konc} - starting computing step, end of simulation time,

2.4 Equations of the model

Force of inertia, motion-resisting force, spring force, hydrostatic force and hydrodynamic force act on the valve slide:

Equation of the slide motion can be written in the following way:

$$m_s \cdot \frac{d^2 x_s}{dt^2} + \alpha_s \cdot \frac{dx_s}{dt} + c_s \cdot x_s = S_s \cdot (p_w - p_s) - 0.36 \cdot Q_s \cdot \sqrt{2 \cdot \rho \cdot (p_w - p_{zl})} \tag{1}$$

Equation of the valve head motion is as follows:

$$m_g \cdot \frac{d^2 x_g}{dt^2} + \alpha_g \cdot \frac{dx_g}{dt} = (p_g - p_{zl}) \cdot S_g - 0.36 \cdot Q_g \sqrt{2\rho(p_g - p_{zl})} - F_{el} \tag{2}$$

Equation of the piston rod motion:

$$m_{cyl} \cdot \frac{d^2 x_{cyl}}{dt^2} + \alpha_{cyl} \cdot \frac{dx_{cyl}}{dt} = (p_{cyl} \cdot S_{cyl} - p_{zl} \cdot S_{cylt}) - F_{obc} \tag{3}$$

In the analysed system, control flows inside of the relief valve: Q_{s1}, Q_{s2}, Q_{s3}, Q_g have been distinguished. The main flows have also been taken into consideration: flow through the relief valve Q_s and the volumetric flow rate to the hydraulic cylinder Q_{cyl}.

The control flows have been described by the following equations:

For the throttle nozzles I, II, III adequately Q_{s1}, Q_{s2}, Q_{s3}:

$$Q_{s1} = \mathrm{sign}(p_w - p_1) \cdot \mu_{d1} \cdot S_{d1} \sqrt{\frac{2(p_w - p_1)}{\rho}} \tag{4}$$

$$Q_{s2} = \mathrm{sign}(p_1 - p_g) \cdot \mu_{d2} \cdot S_{d2} \sqrt{\frac{2(p_1 - p_g)}{\rho}} \tag{5}$$

$$Q_{s3} = \mathrm{sign}(p_s - p_g) \cdot \mu_{d3} \cdot S_{d3} \sqrt{\frac{2(p_s - p_g)}{\rho}} \tag{6}$$

And the volumetric flow rate through the gap of the head valve:

$$Q_g = \mu_g \cdot S_g \cdot \sqrt{\frac{2(p_g - p_{zl})}{\rho}} \tag{7}$$

where: area of the throttle gap S_g depends on the position of the valve head according to the following relationship:

$$S_g = \pi \cdot D_g \cdot x_g \tag{8}$$

The main flow rate through the slide gap Q_s has been described with the formula:

$$Q_s = \mu_s \cdot S_s \cdot \sqrt{\frac{2(p_w - p_{zl})}{\rho}} \tag{9}$$

where: S_s – flow area of the valve slide gap.

After the transformations the formula for the S_s can be calculated as:

$$S_s = \pi \cdot x_s \cdot \sin(\alpha_{sk}) \cdot [D_s - x_s \cdot \sin(\alpha_{sk}) \cdot \cos(\alpha_{sk})] \tag{10}$$

For the individual volumes of the hydraulic system and valves, the flow balance has been drown up, with the fluid compressibility taken into consideration. For the chamber V_1 the following equation is valid:

$$\frac{dp_1}{dt} = (Q_{s1} - Q_{s2}) \cdot \frac{B_1}{V_1} \tag{11}$$

for the V_g chamber:

$$\frac{dp_g}{dt} = (Q_{s2} + Q_{s3} - Q_g) \cdot \frac{B_g}{V_g} \tag{12}$$

and for the V_s chamber:

$$\frac{dp_s}{dt} = (S_s \cdot \frac{dx_s}{dt} - Q_{s3}) \cdot \frac{B_s}{V_s} \tag{13}$$

For the line on the entry to the valve in volume V_w:

$$\frac{dp_w}{dt} = (Q_p - Q_{s1} - Q_s - Q_{cyl}) \cdot \frac{B_w}{V_w} \tag{14}$$

For the hydraulic cylinder:

$$\frac{dp_{cyl}}{dt} = (Q_{cyl} - S_{cyl} \cdot \frac{dx_{cyl}}{dt}) \cdot \frac{B_{cyl}}{V_{cyl}} \tag{15}$$

For the line hydraulic cylinder – counterbalance valve, during the lowering process:

$$\frac{dp_{cyl}}{dt} = (Q_c - Q_s) \cdot \frac{B_c}{V_{cyl}} \tag{16}$$

For the line pump –hydraulic cylinder:

$$\frac{dp_z}{dt} = (Q_p - S_{cylt} \cdot \frac{dx_{cyl}}{dt}) \cdot \frac{B_c}{V_z} \tag{17}$$

2.5 *Control algorithm*

Presented system of the differential equations describes the relief valve motion and flow in the hydraulic system. All equations, completed with the input signals courses and the control algorithm for the proportional valve are the basis of the mathematical model for the simulation of the crane jib lowering control system. A computer programme in Borland Delphi™ 5.0 has been written for solving system of the differential equations. The programme has been named FLC-D5. Schematic diagram of the programme is presented in fig. 4. The programme is composed of the following modules: input data module, differential equations solving module, fuzzy logic controller module and data export module. The mathematical model of the hydraulic system is written in the Pascal notation. A Runge-Kutty change-step method of the IV-th order was used for solving the differential equations.

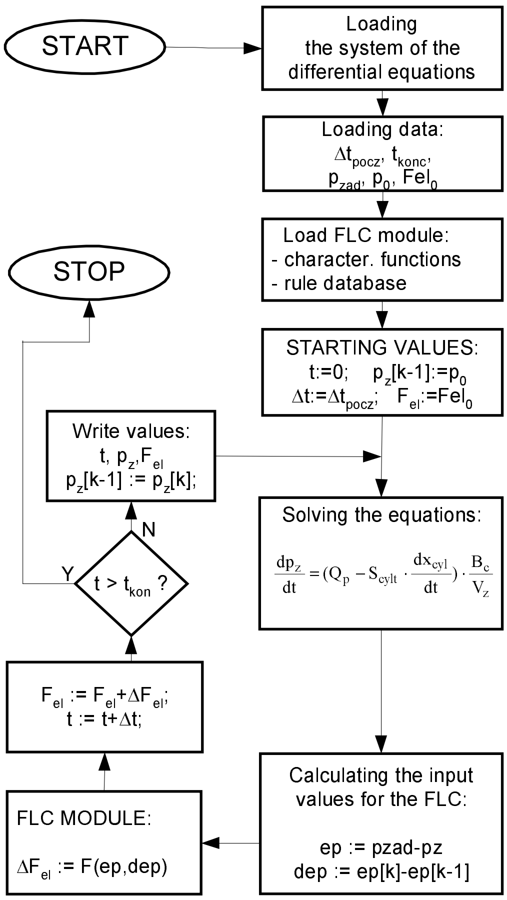

Figure 4. Diagram of the program algorithm

The input function for the system is the supply flow rate Q_p. It was assumed as the step function, however the other functions can be used as well. The controller's main task is generating the control signal for the electromagnet in the proportional valve, so that the expected pressure in the supply line p_z could be obtained. This pressure could be as low as possible, but bigger than the lock pressure of the check valve 3 (fig 2). Thus, the main aim of the simulation was to achieve the expected minimal pressure in the supply line of the hydraulic system. In case of failure of the supply pressure, the regulator has to cut off the outflow from the hydraulic cylinder.

3. FLC CONTROLLER

It has been assumed, that the FLC controller generates the control signal for the electromagnet in the proportional relief valve, on the basis of the actual value of the supply line pressure p_z, and the increment of the pressure since last computation step. A classic controller model with a feedback loop was applied (Figure 5).

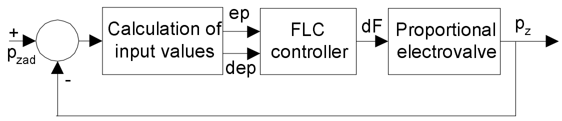

Figure 5. The fuzzy controller model

In Figure 6 is shown the internal structure of the applied fuzzy logic controller. The structure of the controller consists of three parts.

The first one converts the input values into fuzzy values. Triangular characteristic functions were applied inside of the signal ranges, and trapezoidal non-symmetrical on the edges.

The second part calculates the fuzzy value of the output signal on the basis of the data from the rule database.

The last part computes the output signal value. The center of sum (CoS) method was selected for the output signal computation.

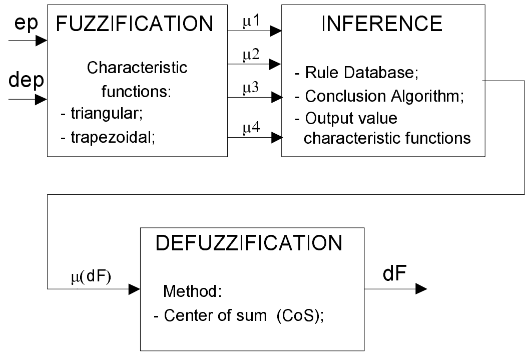

Figure 6. The internal structure of the fuzzy logic controller

4. SIMULATION RESULTS

Simulation of the hydraulic system with the fuzzy logic controller consisted in solving the differential equations (1-17).Geometrical dimensions (Figure 3) were assumed: L=9640mm, a=3272mm, b=2210mm, c=d=661mm. It was assumed that flow rate in the supply line is a step function, and the delivery of the pump is Q_p = 0.5 [dm^3/s]. The initial electromagnet force value was Fel_0 = 18 [N]. The fuzzy logic controller changes the value of the current in the electromagnet coil. It causes the proportional change of the force acting on the electromagnet, and finally change of the pressure in the hydraulic system. Expected pressure value in the supply line was assumed as p_{zad} = 3.5 [MPa]. The simulation time was assumed as t_{kon} = 4.0 [s].

4.1 Pressure courses

The first aim of the simulation was to obtain the pressure courses in the supply line of the system, and in the hydraulic cylinder (Figure 7). The simulations were carried out using different values of the load. Then, the velocity course (Figure 8), and the displacement course (Figure 9) of the piston in the hydraulic cylinder were obtained. On the basis of assumed crane geometry, the course of the crane jib inclination angle was obtained (Figure 10).

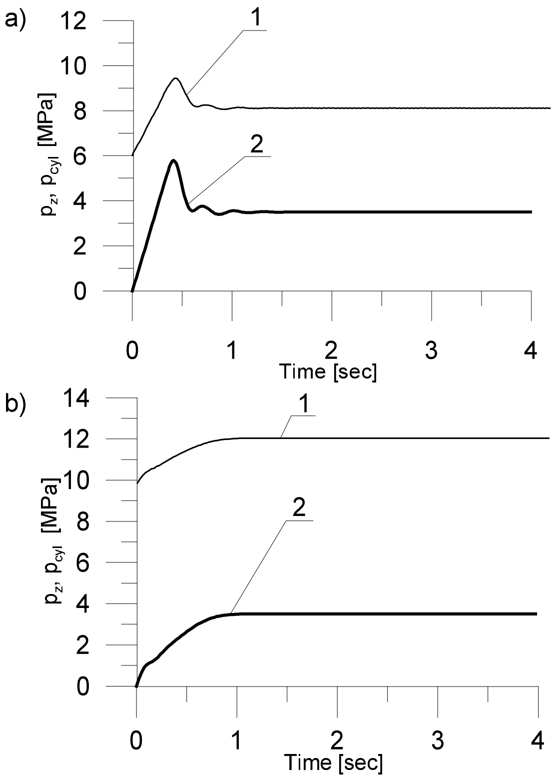

Figure 7. Pressure courses: a) F_{obc} = 300 [kN], b) F_{obc} = 500 [kN]; 1- pressure in the hydraulic cylinder 2- pressure in the supply line.

The pressure courses were obtained using the following cylinder load values: F_{obc1} = 300 [kN], F_{obc2} = 500 [kN]

As it follows from Figure 7, the pressures reached the expected values after about one-second starting time, independently of cylinder load..

The obtained courses of the piston velocity and displacement are shown in the Figure 8, and Figure 9.

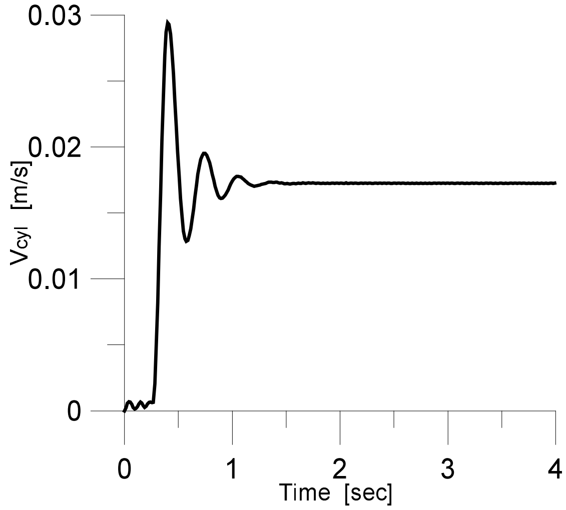

Figure 8. Velocity of the hydraulic cylinder piston with load F_{obc} = 400 [kN]

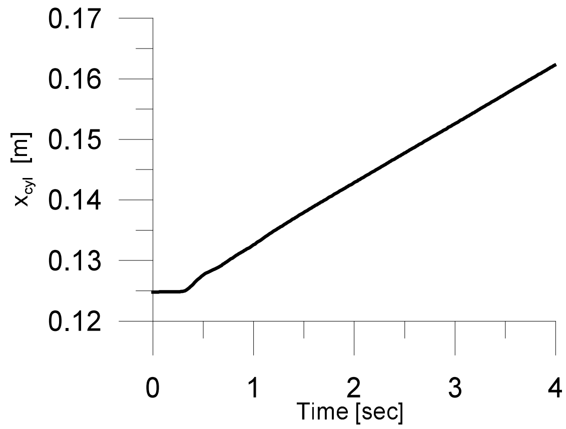

Figure 9. Displacement of the hydraulic cylinder piston with load F_{obc} = 400 [kN]

It follows from Figure 8 and Figure 9, that the velocity of the hydraulic cylinder piston is on the constant level, and it allows to obtain the uniform movement of the piston. Thanks to this, during the lowering process neither the oscillation nor the overloads occur.

Displacement of the hydraulic cylinder piston was then converted into the crane jib inclination. Course of the angle of inclination was obtained. This course is shown in Figure 10.

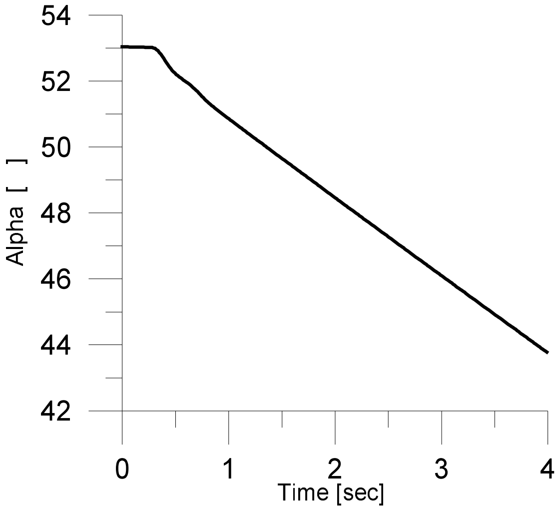

Figure 10. Course of the angle of the crane jib inclination

Application of the FLC controller allows to maintain constant value of the piston velocity, so that the course of the crane jib inclination is almost linear and smooth.

4.2 Disturbance test

The second stage of the simulation concerned the verification of the FLC controller behaviour in case of pressure disturbances in the supply line. The disturbances were applied into the model as step changes of pressure value in the supply line. The pulse amplitude was assumed as $\Delta p_z = 0.5$ [MPa]. In this case the FLC main task was to restore the expected value of the supply pressure as quick as possible.

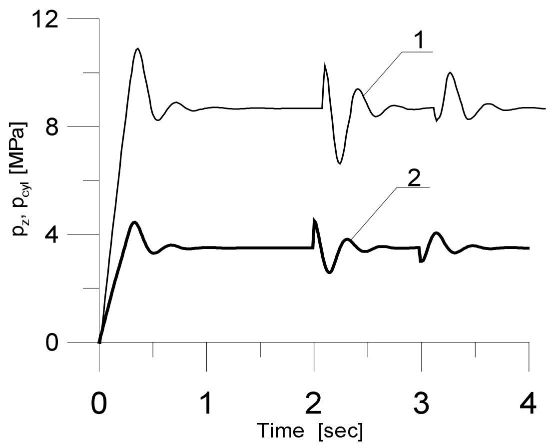

Figure 11. Pressure courses with supply pressure disturbance: (1)- in hydraulic. cylinder (2)-in the supply line

Figure 11 presents the pressure courses in the supply line and in the hydraulic cylinder with the step disturbance of the supply line pressure. In the time t = 2.0 [s] the pressure jump occurred. The pressure value rose up to $p_z = 4.0$ [MPa]. In the time t = 3.0 [s] the supply line pressure value fell down to $p_z = 3.0$ [MPa]. As it follows from Figure 11, the fuzzy logic controller adjusted the control settings, so that the expected value of the pressure could be restored.

5. SUMMARY

In this paper the mathematical model of the hydraulic lowering system of the self propelled crane jib was presented. The velocity of the lowering process was controlled by the fuzzy logic controller. It was assumed, that the fuzzy controller generates the control signal for the electromagnet of the proportional valve. The value of the control signal depends on the current and previous value of the supply line pressure. Applied controller allowed to obtain the expected value of the pressure. Behaviour of the control system in case of occurring the pressure value disturbance was also tested.

The applied mathematical and fuzzy logic controller models confirmed the usefulness in modeling and control of the self-propelled crane working mechanisms.

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