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Calculation and analysis hydraulic system for paint mixing machine

ALMIR OSMANOVIĆ, ELVEDIN TRAKIĆ & EDIN OMERAGIĆ

Abstract Paint production and for that colour mixing is an important process which has wide applications in several field. There are various kinds of colour mixing that can be done. It can either be additive colour mixing or subtractive colour mixing. Generally, there are various kinds of paint mixing machines available in the market. They vary in their size, shape, technology and methodologies. The aim of this paper is to present a calculation and analyse of hydraulic system for paint mixing machine. The paint mixing machine has the task of mixing the oil paint components in vessel until the finished mass is obtained. As oil paint is a typical representative of rheological materials where the viscosity depends on the shear rate, it is necessary to mix different paint structures during processing at different speeds - the number of revolutions of the propeller must be changes.

Keywords: • hydraulic system • analysis • simulation • mechatronic • paint mixing machine •

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1 Introduction

The paint mixer has the task of mixing the oil paint components in vessel A until the finished mass is obtained. Oil paint is a typical representative of rheological materials in which the viscosity depends on the shear rate, it is a necessary to mix different paint structures during processing at different speeds. The container with the paint components is pulled on the cart and placed under the propeller (Figure 1). After placing the vessel in the required place, the piston of the hydraulic cylinder C1 drives the mechanism for clamping the vessel with the mixture, and then the piston of the cylinder C2 immerses the propeller in the mixture. The propeller of the mixer is driven by a hydraulic motor of one-way action (M) with regulation of the number of revolutions. The hydraulic system consists of two separate circuits: for the drive of the hydraulic motor M and the system for providing the control drive of the cylinders C1 and C2 (Figure 2).
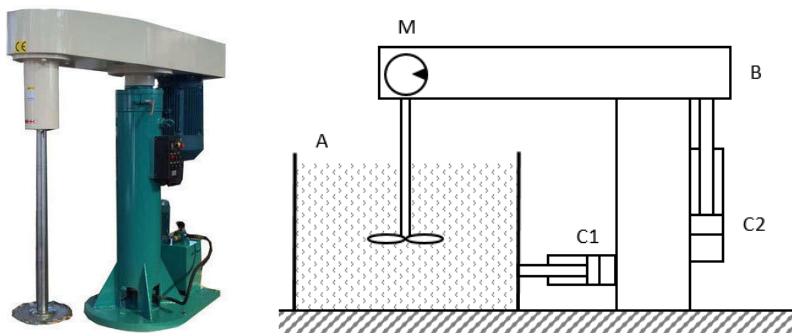
Figure 1: Paint mixing machine and hydraulic components.

Pump P2 with hydraulic regulator drives the hydraulic motor M. The capacity of the pump depends on the pressure of working fluid so that in parallel with its increase. Controlling the position of the regulator is provided by a special branch from the small capacity pump P1. The required pump capacity P2 is controlled by reading the achieved engine speed on the TM tachometer. When the required engine speed is reached, the distribution valve (2) is brought back to the neutral position, and the non-return valve maintains the required pressure in the pump regulator. The pump capacity is kept constant during the further processing of the mixture. If it is necessary to reduce the speed with the production technology, the distribution valves (2) are brought to position (a), the non-return valve is opened, and the control circuit is relieved to the required pressure.
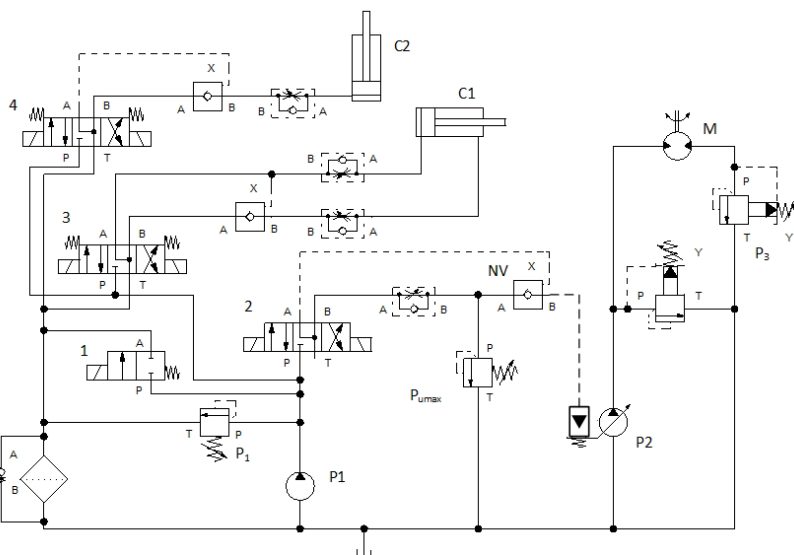
Figure 2: Hydraulic circuit diagram.

The control of the position of the distribution valve (2) is performed, if necessary, by pressing the button placed on the control panel. Cylinder C1 and C2 are supplied by pump P1. The direction of movement of their pistons is controlled by bringing the distribution valves 3 and 4 to the required position, and keeping the pistons in the required position by hydraulically operated non-return valves. The kinematics of the functioning of the hydraulic system is defined by the Table 1 of the solenoid valves.

Table 1: Kinematic functioning of the hydraulic system

System function	1a	2a	2b	3a	3b	4a	4b	
Clamping the container with paint - C2	+	-	-	-	+	-	-	
Lowering the mixer - C1	+	-	-	-	-	+	-	
Increasing the speed - M	+	-	+	-	-	-	-	
Keeping the constant speed - M	-	-	-	-	-	-	-	
Reducing the speed - M	+	+	-	-	-	-	-	
Lifting the mixer - C1	+	-	-	-	-	-	+	
Paint container release - C2	+	-	-	+	-	-	-	
Mixer standstill	-	-	-	-	-	-	-	

2   Calculation of the hydraulic system

Calculation of hydraulic cylinder C1 - for the required clamping force F = 1000 [N] and the adopted cylinder diameter of D = 50 [mm] and connecting rod diameter d = 28 [mm], the required pressure in the cylinder is:

$$F = pA = p_1 A_1 = p_2 A_2 \tag{1}$$

$$p_1 = \frac{4F}{\pi D^2} = 0.5093 \text{ [MPa]} = 5.093 \text{ [bar]}$$

$$p_2 = \frac{4F}{\pi (D^2 - d^2)} = 0.7420 \ [MPa] = 7.420 \text{ [bar]}$$

$$Q = \frac{V_1}{t} = \frac{A_1 l}{l/v} = \ 0.196 \ [l/s] \tag{2}$$

Calculation of hydraulic cylinder C2 - cylinder C2 should enable lifting of platforms of mass m = 500 [kg], speed v = 0.1 [m/s]. Let it takes time to reach this velocity t = 1 [s].

$$F = F_u + G = \ a \cdot m + m \cdot g = \frac{v}{t} m + m \cdot g = 5000 \text{ [N]} \tag{3}$$

The diameter of the piston D = 100 [mm] is adopted.

$$p_{c2} = \frac{F}{A} = \frac{F}{\pi D^2 / 4} = 0.637 \text{ [MPa]} = 6.37 \text{ [bar]} \tag{4}$$

Calculation of the required pump capacity:

$$V = Al = 11{,}779{,}500 \text{ [mm}^3\text{]} = 11.7795 \ \text{[l]} \tag{5}$$

$$Q_p = \frac{V}{t} = 0.784 \text{ [l/s]} = 47[\text{l/min}] \tag{6}$$

In order to select a pump of the appropriate specific capacity, it is necessary to select the number of revolutions of the electric motor and the coefficient of volumetric efficiency of the system (n = 1450 [rpm]; η_{vol} = 0.9).

Specific pump capacity:

$$q_p = \frac{Q_p}{n \eta_{vol}} = 36 \ [\text{cm}^3/\text{ob}] = \frac{36}{2\pi} [\text{cm}^3/\text{rad}] = 5.72 [\text{cm}^3/\text{rad}] \tag{7}$$

A standard axial- piston pump of specific capacity is selected q=36 [cm³/rev] and total capacity $Q = 870$ [cm³/s].

Selection of pressure and return pipe diameters – according to the pressure, the speed of the fluid in the pressure part of the pipeline is adopted: $v = 3.5$ [m/s]. The diameter of the pipeline is:

$$Q = vA = v\frac{d^2\pi}{4} \rightarrow d = 1{,}78 \ [\text{cm}] \tag{8}$$

A pipeline measuring 25x2.5 [mm], internal diameter $d_u = 20$ [mm] is adopted.

$$v = \frac{Q}{A} = \frac{4Q}{d^2\pi} = 2.76 \ [\text{m/s}] \tag{9}$$

Calculation of the required pump pressure - the required pump pressure is calculated based on the sum of the total pressure losses:

$$p_r = \sum \Delta p_1 + \sum \Delta p_2 + \sum \Delta p_3 + \sum \Delta p_4 + \sum \Delta p_5 + \sum \Delta p_6 + \sum \Delta p_7$$

$$p_v = 1.1 \cdot p_r \tag{10}$$

Where are:
$\sum \Delta p_1$ -line resistances in the pipeline from the pump to the cylinder,
$\sum \Delta p_2$ - local resistances in the pipeline from the pump to the cylinder,
$\sum \Delta p_3$ - resistances in hydraulic components from pump to cylinder,
$\sum \Delta p_4$ - line resistances in the pipeline from the cylinder to the tank,
$\sum \Delta p_5$ - local resistances in the pipeline from the cylinder to the pump,
$\sum \Delta p_6$ - resistances in hydraulic components from cylinder to tank,
$\sum \Delta p_7$ - opposing resistances of external forces to the movement of the piston.

The pressure drop of pipeline is equal to:

$$p_{lin} = \sum \Delta p_2 + \sum \Delta p_5 \tag{11}$$

If you want to make an accurate calculation, it is necessary to calculate the pressure drops in all branches in each operation. An accurate calculation is needed to calculate the total heat balance. Often, pressure drops are not calculated in detail, because it is a complicated calculation, especially with complex hydraulic systems. In such situations, only the critical branch is calculated, in which the pressure drop is the largest, and the entire hydraulic system is dimensioned in relation to it.

The pressure drop when lifting the platform is:

$$\Delta p_{lin} = \lambda \cdot \frac{l}{d} \cdot \frac{v^2}{2} \rho = 25084 \; [\text{N/m}^2] = 0.25084 \; [\text{bar}] \tag{12}$$

Local pressure drop - From pump to cylinder we have the following local losses: manifold pressure drop (4) 4/3; pressure drop on throttle valve with non-return valve; pressure drop on non-return valve with hydraulic control. Based on the Q-Δp diagram for the manifold model (Figure 3), the pressure drop for $Q = 50 \; [\text{l/min}]$ is $\Delta p_a = 0.21$ [MPa] = 2.1 [bar].
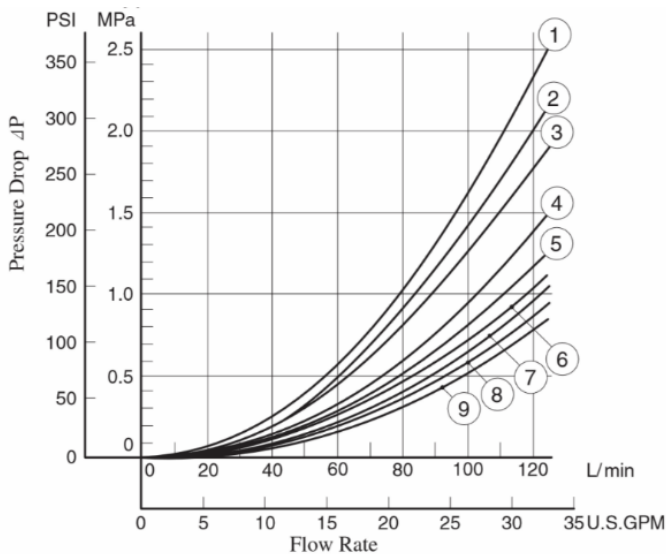

Figure 3: Dependence of manifold pressure drop on flow.

The pressure drop in the non-return valve is given by the diagram Q-Δp (Figure 4) (Δp_b = 0,06 [MPa] = 0,6 [bar]). The pressure drop on the non-return valve with hydraulic control can be determined from the diagram (Figure 5) and for the model the valve model selected is: (Δp_c = 0,3 [MPa] = 3 [bar]).
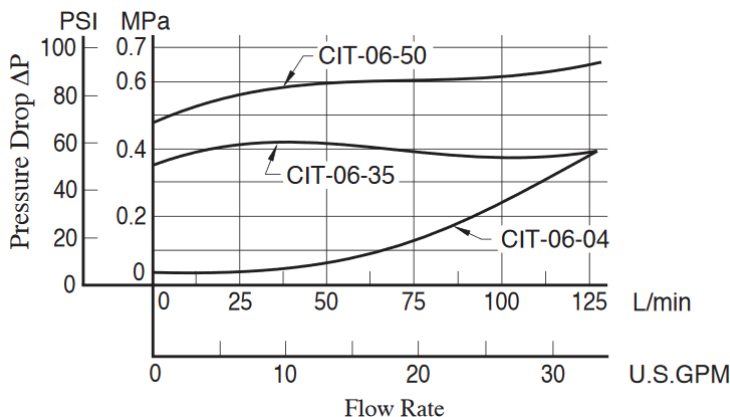

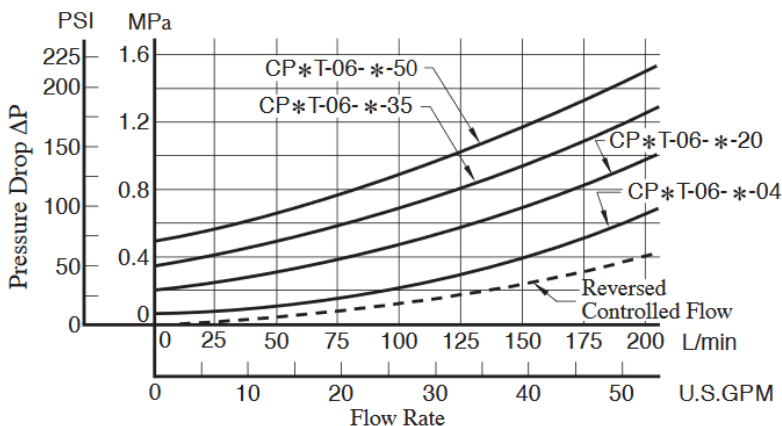Figure 4: Dependence of non-return valve pressure drop on flow.

Figure 5: Dependence of pressure drop of non-return valve with hydraulic control.

Pressure drop is:

$$\Delta p = \Delta p_{lin} + \Delta p_a + \Delta p_b + \Delta p_c = 5.95 \text{ [bar]} \tag{13}$$

The total pressure is the pressure drop and the pressure required to overcome the external resistance: $p_r = \Delta p + p_{c2} = 12.32$ [bar] . The pump pressure increases by 10 % and is: $p_p = 1.1p_r = 13.55$ [bar]. Based on the calculated pressure, the pump with the highest pressure $p_{max} = 20$ [bar] is selected. The pressure relief valve will be set to a pressure of $p_{vop} = 15$ [bar].

Electric motor power calculation - electric motor should enable the pump to start with the following characteristics: maximum pump operating pressure $p_{max} = 20$ [bar], pump capacity $Q = 52.2$ [l/min]. The power of the electric motor of the pump is equal to ($\eta_v = 0.9$-volumetric efficiency coefficient; $\eta_m = 0.85$- mechanical efficiency):

$$N = \frac{p_v Q}{600 \eta_v \eta_m} = 2.27 \text{ [kW]} \tag{14}$$

2.1   Calculation of pump and mixer motor

The estimation of the mixer shaft power (P) required for mixing is based on the application of the following dimensionless correlation:

$$N_p = f(Re_M, Fr_M, \frac{D}{d_m}, \frac{H}{d_m}, \frac{d_3}{d_m}) \tag{15}$$

The installation of the breaker prevents the occurrence of unfavourable vortices and thus eliminates the influence of Freud's number.

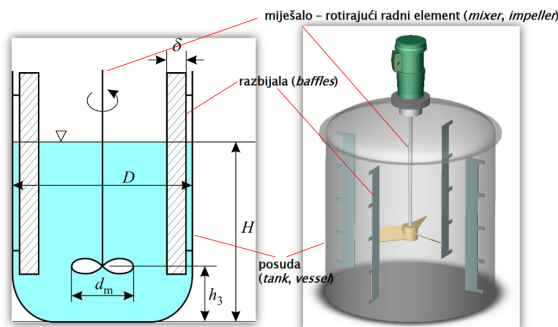$$N_p = f(Re_M, \frac{D}{d_m}, \frac{H}{d_m}, \frac{d_3}{d_m}) \tag{16}$$

Figure 6: Mixer parameters.

In this case $Re_M = 243$ and the flow is in the transition region between laminar and turbulent. From the power diagram, the power number is about $Np = 1.5$.
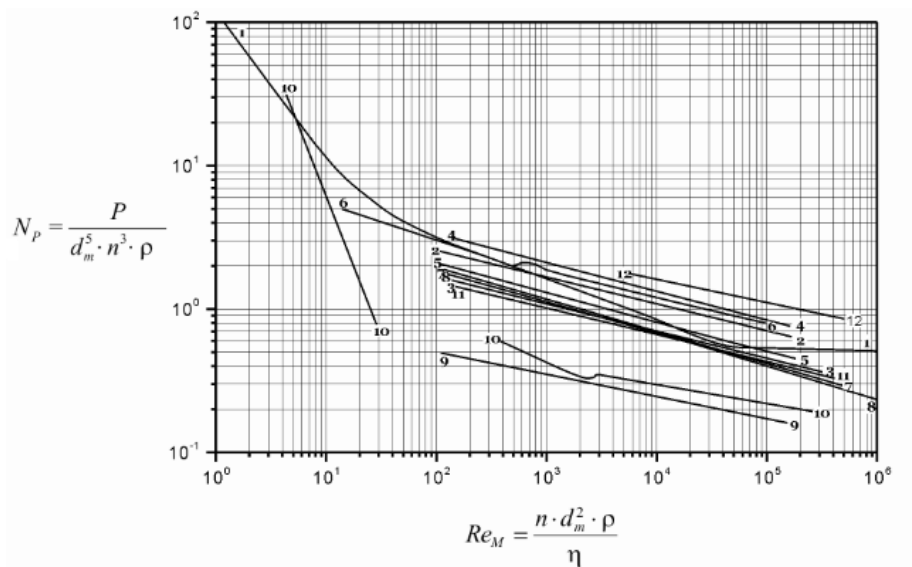

Figure 7: Dimensionless power diagram.

The power on the mixer shaft is:

$$P = N_p \cdot n^3 \cdot d_m^2 \cdot \rho_f = 182 \text{ [kW]} \tag{17}$$

Required torque and specific volume:

$$M = \frac{P}{\omega} = 2896 \text{ [Nm]} \tag{18}$$

$$P = \frac{\Delta p \cdot Q}{600} \rightarrow Q_m = \frac{600 \cdot P}{\Delta p} = 436.8 \text{ [l/min]} \tag{19}$$

$$q_m = \frac{Q_m}{n \cdot \eta_v} = 485 \text{ [cm}^3\text{/ob]} \tag{20}$$

The Rexroth A4FM500 engine is selected based on specific volume, operating pressure and torque. Basic data are given in Table 1.

Table 1: Basic data of A4FM500

Motor		
Specific volume	$q_m = 500 \text{ o/cm}^3$	
Maximum speed	$n = 1800 \text{ o/min}$	
Torque	$M = 2783 \text{ Nm}$	
Pressure drop	$\Delta p = 350 \text{ bar}$	
Volumetric efficiency	$\eta_v = 0.9$	

Fluid flow in the hydraulic circuit of the pump and motor let be $v = 4,5$ [m/s]. The diameter of the pipeline is:

$$Q_m = q_m \cdot n \cdot \eta_{vol} = 450 \text{ [l/min]} = 7500 \text{ [cm}^3/\text{s]} \tag{21}$$

$$d = \sqrt{\frac{4Q}{\pi v}} = 4.6 \text{ [cm]} \tag{22}$$

The pipeline diameter d = 50 [mm] is adopted.

$$v = \frac{Q}{A} = \frac{4Q}{d^2 \pi} = 3.82 \text{ [m/s]} \tag{23}$$

Line pressure drop from pump to reservoir:

$$\Delta p_{lin} = \lambda \cdot \frac{l}{d} \cdot \frac{v^2}{2} \rho = 46754 \text{ [N/m}^2] \tag{24}$$

The pressure drop on the pressure relief valve is given in the Table and for a flow of 500 [l/min] it is Δp_v=16 bar. Working pressure: $\Delta p_r = 266.47$ [bar]. The pump pressure increases by 10% and is: $p_p = 1.1 \Delta p_r = 293$ [bar]. Based on the calculated pressure, the pump with the highest pressure $p_{max} = 350$ [bar] is selected. The pressure relief valve will be set to a pressure of $p_{vop} = 300$ [bar]. Basic data of the used pump are given in Table 2.

Electric motor speed $n_{em} = 950$ [rev/min]. The theoretical flow of the pump is:

$$Q = \frac{q_m \cdot n}{1000} = 475 \text{ [l/min]} \tag{25}$$

Table 2: Basic data of used pump type AA4CSG 500

Pump (Rexroth AA4CSG 500)		
Specific volume	$q_m =$ 500 o/cm^3	
Maximum speed	$n =$ 1800 o/min	
Torque	$M =$ 2783 Nm	
Pressure drop	$\Delta p =$ 350 bar	
Volumetric efficiency	$\eta_v =$ 0.9	

Pump electric motor power:

$$N = \frac{p_v Q}{600 \eta_v \eta_m} = 362 \text{ [kW]} \tag{26}$$

3   Modelling and simulation of hydraulic system models

The Figure 8 shows the simulation model of the entire system. According to the principle of operation, it can observe three independent parts, so it will analyse the following subsystems separately: raising and lowering the platform; clamping the paint cart; mixing speed control.
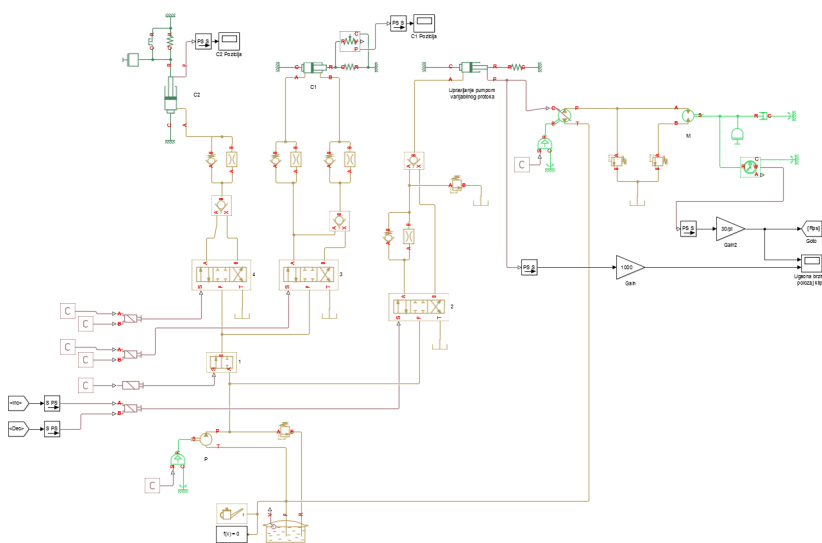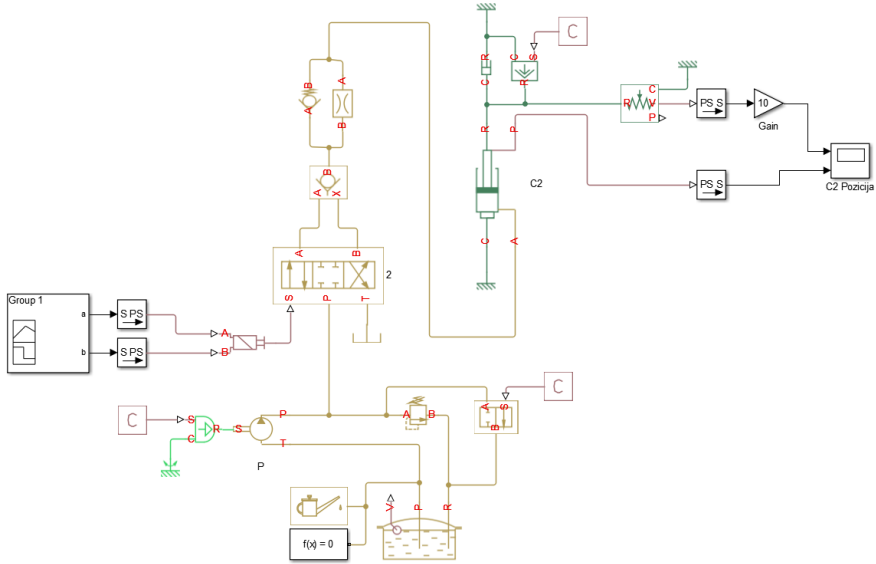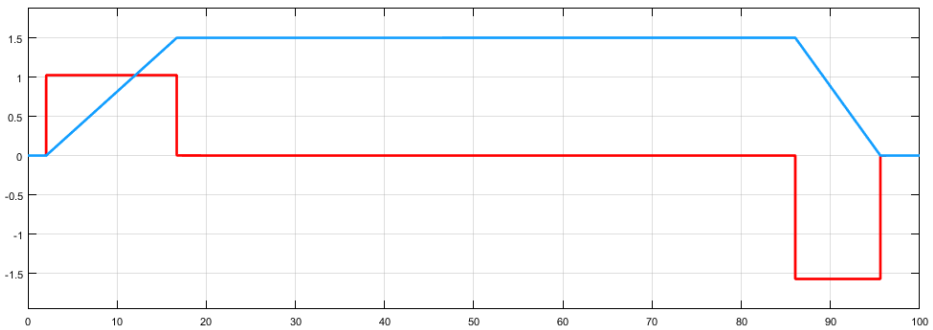

Figure 8: Simulation model of hydraulic system.

Figure 9: Platform lifting and lowering model.

Figure 10: Road and speed diagram.

The following values can be obtained from the diagram:

- Platform boot time T_d = 15 [s].
- Platform lowering time T_s = 10 [s].
- Piston speed when lifting the platform v_d = 0.1 [m/s].
- Piston speed when lowering the platform v_s = 0.16 [m/s].

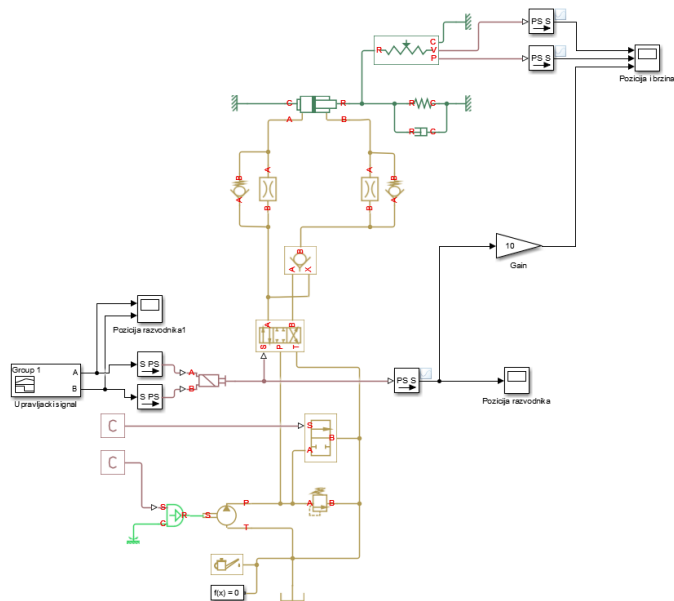The values obtained for simulation correspond to the previously calculated values.

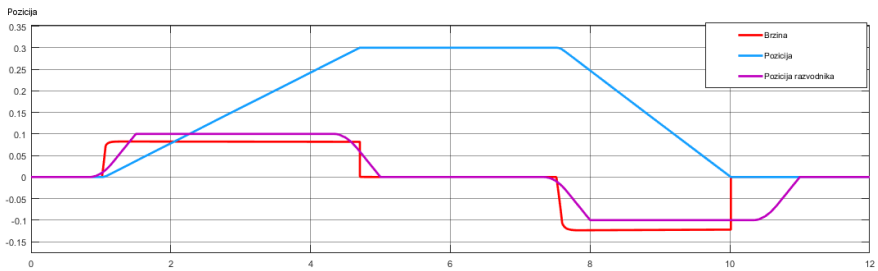**Figure 11: Model of hydraulic circuit for clamping a container with paint.**

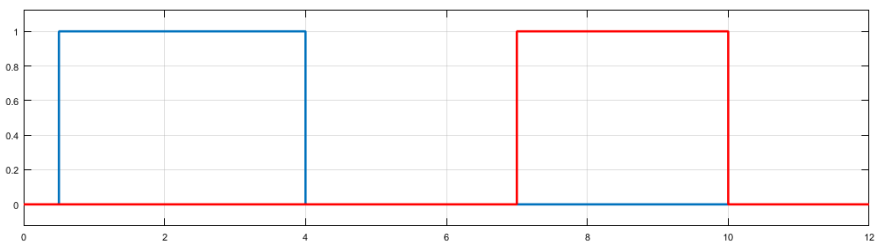**Figure 12: Diagram of the path, speed and position of the valve.**

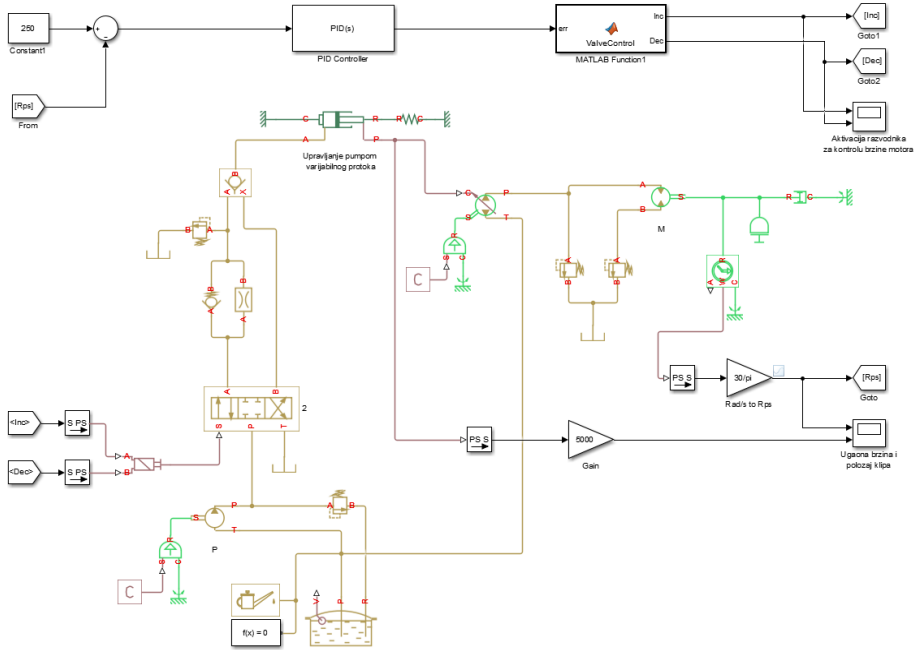**Figure 13: Control signal for valve.**

Figure 14: Model of mixing speed control by control of a variable capacity pump.
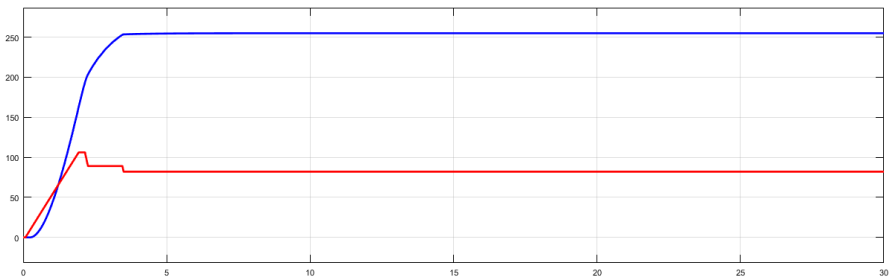

Figure 15: Diagram of angular velocity and position of the connecting rod for pump flow control (PID regulator).

From the values achieved after simulation, it can conclude that the model fully satisfies or that the required parameter is achieved, and that the components, and the model as such can be practically used to create a real system.

3 Conclusion

In this paper work is present calculation and simulation model of the hydraulic system for a paint mixing device. The calculation was performed on the basis of the hydraulic scheme and the principle of operation of the mixing device. The capacity of the mixing device was adopted and on this basis the additional parameters used in the calculation were determined. The outcome of the calculation itself depends on the parameters that are adopted during the calculation, which in a certain way meet the set requirements, e.g. it is possible to reduce the working pressure of the pump by adopting a cylinder with a larger diameter, on the other hand increasing the diameter of the cylinder reduces the speed of the connecting rod which results in choosing a pump with a larger specific volume or increasing the pump speed. Achieving the optimal solution is obtained by setting certain conditions in critical places of the hydraulic system, in this way individual branches of the hydraulic system are solved and the parameters are brought to the working areas. In the second part of the paper, was created model for simulation, which gives a better insight into the behaviour of the whole system. Execution of system simulation is enabled by solving differential equations. Each component of the system is defined by a set of equations, and based on the graph of components. Using appropriate software tools, it provides the ability to simulate the operation of a designed hydraulic system, and it is possible to perform optimization in order to find the most favourable solution, and thus select the components used in the manufacture of hydraulic systems.

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