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CONSIDERATIONS CONCERNING THE DYNAMIC PERFORMANCES ANALYSIS OF A SPECIAL VEHICLE STEERING SERVOMECHANISM

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Abstract. *The paper presents an analysis using modeling and simulating techniques of a special vehicle steering servomechanism. The modern simulation software allows studying the complex automotive systems; at the same time allows the fast variation of the system's parameters and the study of all influences that occur within the working system.The working characteristic features of a process can be synthetically expressed as a mathematical model. For the technical systems, this model can be obtained based on the mathematical expressions that describe the working way of the system's components. The mathematical model can be used to analyze the dynamic performances of the system or a proper way to improve them, as far as the actual evolution of the system is accurately enough described. Some of dynamic charts are presented and improvements of the dynamic performance methods are given.* ***Keywords:*** *Mathematical model, simulation, steering servomechanism, dynamic performances.*

1. GENERAL APPROACH

The steering mechanism that is subject to this paper provides the steering of special tracked vehicle. The planetary gearing systems [1] work as drivelines for the gearbox's output to the side gears, using either normal (1:1 ratio) or low (1:1.44 ratio) driving modes. Acting together with the stopping brakes, they achieve steering of the vehicle and a short increase of the traction effort of the vehicle's tracks (without shifting the gears inside the gearbox). They also provide a speed decreasing or even stopping the vehicle, keeping it still on a gradient without pressing the braking pedal.

The vehicle steers by driving the tracks at different speeds. The vehicle will steer towards the slower track. The planetary steering mechanisms have their own clutches, placed between the gearbox and the final (side) gears.

When analyzing the working modes of the planetary steering gears, their hydraulics should be also taken into account, i.e. the steering distributor (fig. 1). Its command is achieved from the steering handlebar, acted by the driver.

When the vehicle moves in straight line, the handlebar is horizontal and the lever of the "low ratio" gear of the final drive is in its "up" position (i.e. "normal gear engaged, low gear disengaged"). In this mode, the acting rods of the steering distributor are completely retreated by their restoring springs; the locking clutch is engaged. The speed-decreasing brakes are disengaged and the planetary steering mechanism provides a 1:1 driving ratio.

To achieve the steering, the handlebar is turned to the left (counter-clockwise) or to the right (clockwise). The plungers of the steering distributor achieve the variation of the steering radius of the vehicle. The plungers change their positions and gradually build-up the oil pressure within the cylinders of the locking clutch and the speed-decreasing brakes.

The planetary steering mechanism consists of a planetary gear 11, a locking clutch 9 of the sun gear and a speed-decreasing brake (7 or 14). On the stopping brake (towards the side gear) there is a brake drum the brake is acting on.

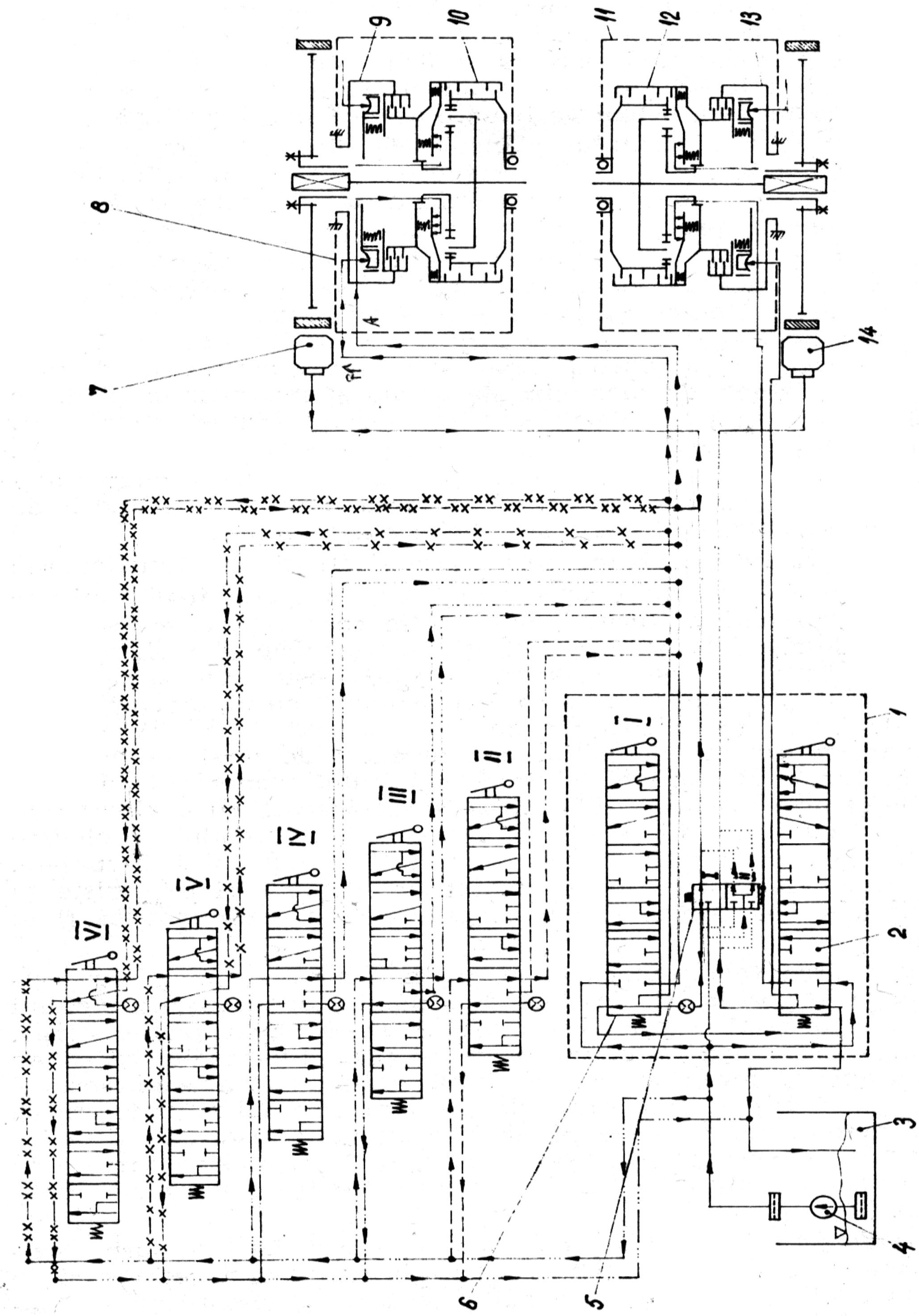


Figure 1 1– hydraulic steering distributor; 2–right-hand distributing valve; 3-gear casing; 4–oil pump; 5–brakes distributing valve; 6–left-hand distributing valve; 7–left-hand band brakes; 8–left-hand planetary gear steering mechanism; 9–steering mechanism brake; 10–clutch of the left-hand planetary gear steering mechanism; 11–right-hand planetary gear steering mechanism; 12-clutch of the right-hand planetary gear steering mechanism; 13–brake of the right-hand planetary gear steering mechanism; 14–right-hand band brakes.

2. MATHEMATICAL MODEL

The mathematical model of the hydraulic servomechanism [2], [3] has been issued taking into account the following equations:

- the continuity equation, corresponding to the non-permanent motions within the acting hydraulic systems;

- the equation of the displacement - flow rate - pressure;
- the equation of the hydraulic engine piston's movement.

The hydraulic servomechanisms are mounted using assembling elements. The elasticity of these elements influences both the stability and the positioning accuracy of the servomechanism. If not considering the finite stiffness of these components, then the servomechanism is as good as ideal mounted (the stiffness of all the mounting elements to the supporting structure and to the actuated system is considered to be infinite). In real life, the stiffness is finite. Moreover, it slowly decreases due to the loads the mounting components have to bear.
Considering Q the flow through a stream tube [1], [2], [3], the continuity equation can be written as follows:

$$\frac{A\cdot\rho}{E}\cdot\frac{dp}{dt}+\rho\cdot\frac{\partial Q}{\partial s}=0 \tag{1}$$

If further processed, it leads to:

$$\frac{dp}{dt}=-\frac{E}{A}\cdot\frac{\partial Q}{\partial s}$$

$$\frac{dp}{dt}=\frac{E}{\vartheta}\cdot(Q_1-Q_2) \tag{2}$$

where:
- Q_1 – liquid flow through first (inlet) section;
- Q_2 - liquid flow through second (outlet) section;
- A – flow cross-section;
- υ - liquid system's volume;
- s – displacement (travel);
- t- time;
- p – liquid system's pressure;
- E – elasticity modulus;
- ρ - oil density

The flows Q_1 and Q_2 can be written as follow [3]:

$$Q_1=c_{ip}\cdot(p_1-p_2)+c_{ep}\cdot p_1+A_p\cdot\frac{d(z+u)}{dt}+\frac{\vartheta_1}{E_e}\cdot\frac{dp_1}{dt} \tag{2}$$

$$Q_2=A_p\cdot\frac{d(z+u)}{dt}+c_{ip}\cdot(p_1-p_2)-c_{ep}\cdot p_2-\frac{\vartheta_2}{E_e}\cdot\frac{dp_2}{dt} \tag{3}$$

where:
- c_{ip}– internal leakage motor's coefficient;
- c_{ep} – external leakage motor's coefficient
- p_1 – pressure of the intake motor's chamber;
- p_2 - pressure of the exhaust motor's chamber;
- Ap - effective cross-section of the motor's piston;
- z – piston's rod displacement;
- u – servomechanism's body displacement;
- υ_1, υ_2 – flows through the motor's chambers and the coupling pipes;
- E_e - equivalent modulus of elasticity.

For a symmetrical hydraulic distributor, if using the hydraulic stiffness of the hydraulic motor, R_h, [1] one could write:

$$Q=K_l\cdot P+A_p\cdot\frac{d(z+u)}{dt}+\frac{A_p^2}{R_h}\cdot\frac{dP}{dt} \tag{4}$$

where K_l – total flow coefficient throughout the distributor;
P - pressure drop on the hydraulic motor.

For a symmetrical dividing, paired-slots, equal flowing type distributor, the flow provided to the engine by the distributor can be written using the following equation:

$$Q_m=Q_m(x,P)=c_d\cdot A(x)\cdot\sqrt{\frac{(P_s-P)}{\rho}} \tag{5}$$

where:
- Q_m – the flow taken by the engine;
- A(x) – distributor's slot cross-section;
- c_d – flow coefficient of the distributor's slot;
- P_s - fluid's pressure at the distributor's inlet port.

Using a Taylor developing procedure for equation (5) around the working plot "o" one can get the following relation:

$$Q_m - Q_{mo} = \Delta Q = \left(\frac{\partial Q_m}{\partial x}\right)_o \cdot \Delta x + \left(\frac{\partial Q_m}{\partial P}\right)_o \cdot \Delta P + \ldots \tag{6}$$

where the above written factors are as follows:

a) amplifier flow factor of the distributor:

$$K_{Qx} = \frac{\partial Q_m}{\partial x} \tag{7}$$

b) flow – pressure coefficient:

$$K_{Qp} = -\frac{\partial Q_m}{\partial P} \tag{8}$$

c) pressure – displacement coefficient:

$$K_{px} = \frac{\partial P}{\partial x} = \frac{K_{Qx}}{K_{Qp}} \tag{9}$$

Thus, equation (6) becomes:

$$\Delta Q = K_{Qx} \cdot \Delta x - K_{Qp} \cdot \Delta P \tag{10}$$

The pressure acting on the servomechanism's piston moves it with a motion law described by:

$$F = m_p \cdot \ddot{z} + K_{fv} \cdot (\dot{z} + \dot{u}) + R_C \cdot (z - v) + D_C \cdot (\dot{z} - \dot{v}) \qquad 11)$$

where

- R_c – the stiffness of the connecting mechanism between the servomechanism and the load (actuated element)
- D_c – the damping coefficient of the connecting mechanism;
- z – displacement of the actuated element;
- K_{fv} – viscous friction coefficient;
- m_p – piston's mass

3. SIMULATING THE SERVOMECHANISM'S WORK

The steering mechanism's distributor work was simulated using the Simulink module of Matlab programming environment

Perturbation R (the force needed to engage the locking clutch and the force needed to apply the speed-decreasing brake respectively) has been described as having an exponential variation as $R = a \cdot e^{b \cdot z}$, where the coefficients have the values a=5; b=20

The results of the simulation [4] as well as the influence of different parameters over the working performances of the system are given in the following pictures.

4. CONCLUSIONS

1. The system is stable in its work. It shows a fast damping of the pressure oscillations while the response time of the system keeps at very low values. Therefore, we could say that the system quickly follows the applied command of the driver. We performed the simulation using the step signal (corresponding to a sudden action of the handlebar). Under the given circumstances, the time response is lower than 0.1 seconds.
2. Analyzing the charts depicted in fig. 2 -4, we could reveal the influence of the flow coefficient K_{Qx} over the dynamic performances of the system. The higher the value of K_{Qx} (i.e. using distributors with larger slots), the higher the oil flow. Hence, the filling speed of the cylinder increases in the first stage. After that, we could notice a decrease of the filling speed due to the leaks of the incoming oil. The moment of the filling speed decrease debut matches the moment of the overpressure protection valve opening.
3. The higher K_{Qx} is, the higher the pressure increase speed is (i.e. the nominal value is faster reached). On the other hand, pressure waves occur at the starting point. They are due to the increase of the accelerations of the actuated mass. Nevertheless, these oscillations are rapidly damped, since the system has a stable working way.
4. The response time of the system (when excited with the step signal) decreases when the flow through the distributor increases. The speed and acceleration of the piston oscillations also increase. The system rapidly reacts to the flowing cross-section's variations. All in all, this phenomenon could also be noticed in the

depicted time histories, when crossing the points the overpressure valve opens (the second wave of oscillations).

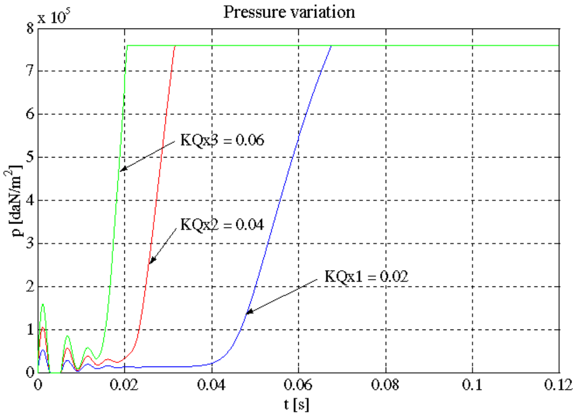


Figure 2 - The influence of the flow coefficient K_{Qx}, on the cylinder’s pressure

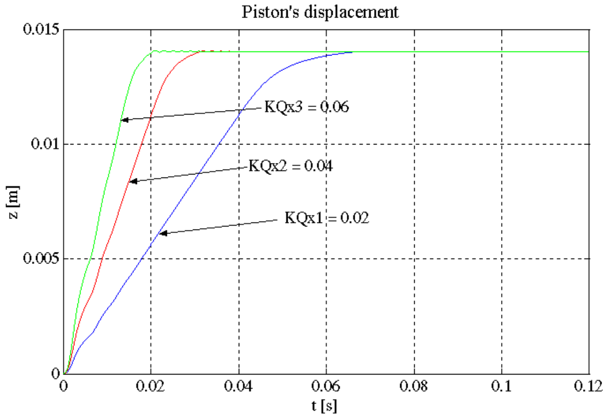


Figure 3 - The influence of the flow coefficient K_{Qx}, on the response (when excited with the step signal)

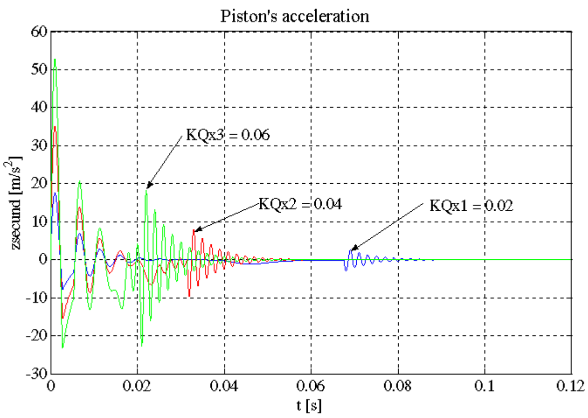


Figure 4 - The influence of the flow coefficient K_{Qx}, on the piston’s acceleration

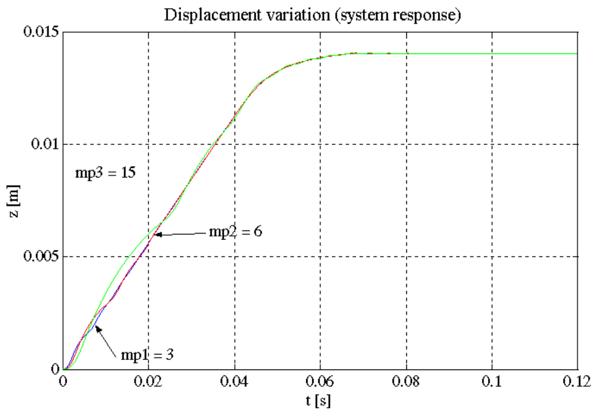


Figure 5 – The influence of the reduced mass on the response (when excited with the step signal)

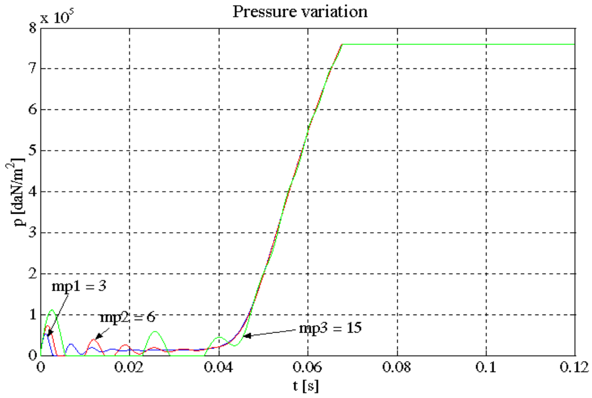


Figure 6 – The influence of the mass reduced on the piston’s rod on the pressure variation

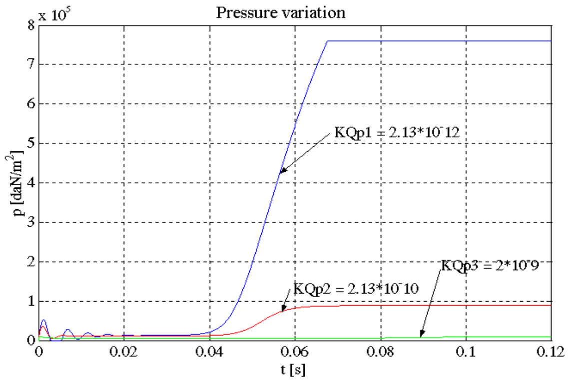
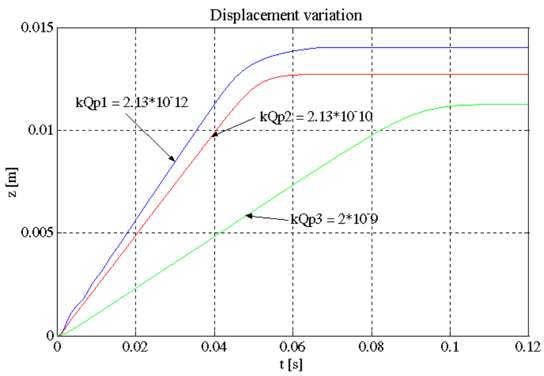


Figure 7 – The influence of the pressure coefficient K_{Qp} on the pressure variation



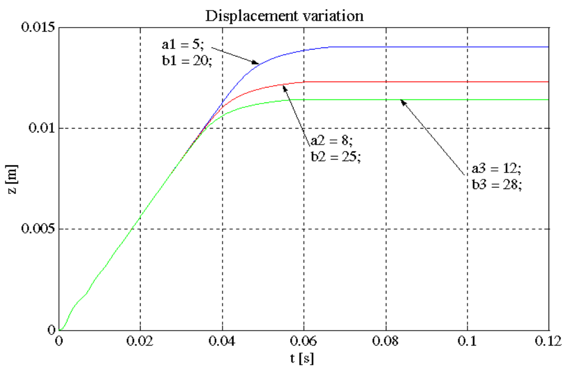


Figure 8 - The influence of the pressure coefficient K_{Qp}, on the response (the step signal)

Figure 9 - The influence of the perturbation (force) on the response (the step signal)

5. Fig. 5 – 6 show that the total mass of the moving parts, reduced to the piston's rod, doesn't significantly influence the system's dynamic performances. The response time remains quite the same, within the limits of the force that is developed at the piston's rod level. Some pressure waves occur from the starting point to the "defeat" of the actuated mass inertia. These pressure waves (determined by the compressibility of the working fluid) generate oscillations of the piston's speed and acceleration.
6. Fig. 7 – 8 show the influence of the pressure coefficient K_{Qp} over the dynamic performance of the system. When this parameter is increased (corresponding to an increase of the losses due to pressure difference between the distributor's chambers), the average flow within the distributor decreases.
7. The higher the losses due to the pressure, the higher the time response. Moreover, the servomechanism's force drops due to the system's pressure drop.
8. Should be also mentioned that the travel of the mechanism is incomplete. The perturbation (force) input in the simulation model estimates the needed force to actuate the engaging elements. The travel of the actuated element is proportional with its coasting force. The conclusion is that a lack in completing the travel leads to smaller actuating forces (that could lead to slipping in clutch's work or inefficient braking process). This situation corresponds to high degrees of wear of the distributor.
9. If the pressure losses increased, the system becomes more stable, i.e. pressure waves disappear and the piston moves more smoothly. Hence, the elements that fix the distributor to the mechanism's case are subject to lower reactive forces with benefic effects on their stresses.

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