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ABOUT THE TURNING DELAY OF THE MOTOR VEHICLE WITH HYDRAULIC SERVO-STEERING

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Abstract: *It is studied the hydraulic servo-steering dynamics for automotive with constant flow rate volume pump, linear hydrostatic motor and slide valve. It is established the time of the dynamic coupling through the sum of partial times (in connection with the slide valve control displacement, the working fluid compressibility, the elastic propagation of the pressure shock, and all), like a delay of the wheel control for motor vehicle turning, that must be anticipated from the driver. The practical possibility to emphasize determinant factors permits to take efficient measures for the dynamic improvement of hydraulic servo-steering beginning with design, construction and exploitation.*
Keywords: *hydraulic servo-steering, hydraulic shock, celerity, turning delay.*

1. INTRODUCTION

The servo-control, known that servo-steering, represents a command system subservient to amplify the power, destined to lighten the motor vehicle drive (tractors, autotrack, agricultural machinery, utilitarian and special vehicles) by reducing the necessary effort on the steering-wheel, damping shocks transmitted from the direction wheels to the steering wheel, increasing the maneuverability and the security of the movement.
Works [1], [2] and [3] carried out the dynamic analysis of this system, the composition, and the stability. Concerning the force at the wheel there are recommended works [3], [4] and [5].
However, because some phenomena there are a turning delay that cannot be neglected for an accurate and careful analysis. This study attempt to establish the turning delay by computing the time of the dynamic coupling through the sum of partial times due to the slide valve control displacement, the working fluid compressibility, the elastic propagation of the pressure shock, and all.

2. PROBLEM DESCRIPTION

The control by wheel of the road vehicle bend uses a power hydraulic amplifier which the structural scheme includes (Fig. 1): the control device (the wheel, V and the steering column mechanism, Mc); the error transducer (slide valve, Ds); energy source (constant flow rate pump PDC, provided with maximal valve, VM and oil reservoir Rz), the output regulator (output hydrostatic motor MH, usually double action hydraulic cylinder), the controlled device (direction wheels RD by means of a trapezoid steering mechanism, TRD), a feedback bundle, LR, hydraulic and/or mechanic in order to realize the effort sensation - “road sensation”.
The constructive scheme of the hydraulic servo-steering for road vehicles, structured as above is presented in Figure 2 (with the additional notation LD - the steering lever).
The pump with flow rate Q feeds at the pressure p, the linear hydraulic motor having the acting surfaces $S = \pi d^2/4$, (in which d is the inner diameter of the hydraulic cylinder) through a distributor with slides and multiple plunger (slide valve). By introducing a displacement x of the slide valve leads to the out of idle pressures balance, which pursues to the displacement of the linear hydraulic motor with the value y, in the presence of force F_M.

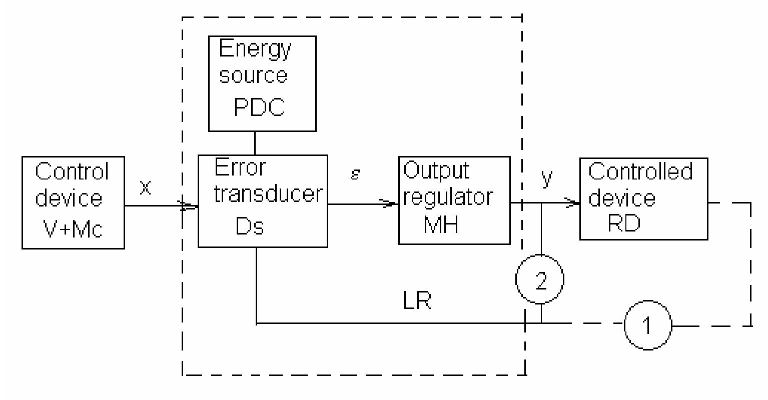

Figure 1: The structural scheme of the hydraulic servo-steering

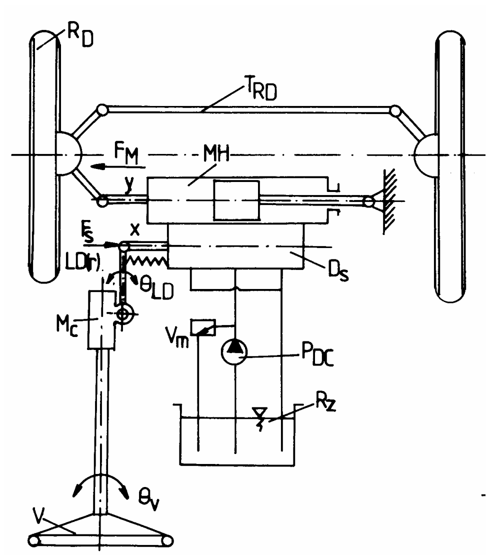

Figure 2: The constructive scheme of the servo-steering

The dynamic study of a mechanic-hydraulic subsystem for pursuit used in automotive servo-steering by a mathematical model is presented in [5]. In the present work is presented the problem of the command transient process duration, the time and distance of acting. The time of the dynamic coupling of the power hydraulic amplifier is a delay time at the control by wheel of the automotive turning, which should be predicted by the driver, because that time the automotive runs a distance function of the speed, certainly significantly.

3. MATHEMATICAL MODEL

The acting time results by summation of times:

- commutation of the distribution valve;
- reaching the start pressure;
- accelerating of actuating masses *m* until reaching the regime velocity;
- crossing the actuating stroke;
- crossing the clearances and elasticity from installation;
- filling some voids V_g from installation t_u;
- crossing time of conduit from the slide valve (Ds) to hydrostatic motor (MH) by the hydraulic shock (preferable indirect, attenuated).

The time of filling some voids is:

$$t_u = \frac{V_g}{Q}\left[\frac{\mathrm{m}^3}{\frac{\mathrm{m}^3}{\mathrm{s}}} = \mathrm{s}\right], \tag{1}$$

in which Q is the volume flow rate entrant in the slide valve. But the installation remains permanently full with liquid by means of single sense valves, so practically it is null. Moreover any existing void shall deteriorate system characteristics, the mixed fluid (hydraulic oil an air) becoming very compressible and the installation cannot perform firm commands.

The time of the dynamic coupling of the power hydraulic amplifier and establish of the force F at the hydraulic motor (adequate to realize the steering wheels deflection), so the adequate pressure values in working cylinder chambers, in conformity with [2], for $p = 50$ bar , $x = y = 0.1$ mm , $F = 200$ daN , is comprised between values $t_F = 0.04...0.08$ s . The reach of the working pressure in the hydraulic motor, in the compressible liquid, is realized in a time t_p, resulted by integration the below relation from the reference overpressure zero to the motive working pressure:

$$Q = \frac{V_l}{\varepsilon}\cdot\frac{\mathrm{d}p}{\mathrm{d}t}\left[\frac{\mathrm{m}^3}{\mathrm{s}}\right], \tag{2}$$

in which $V_l\,[\mathrm{m}^3]$ is the overall liquid volume that should be compressed until the pressure $p\,[\mathrm{Pa}]$, and $\varepsilon\,[\mathrm{N/m}^2]$ - liquid module of elasticity. It can observe that the volume flow rate is supposed a constant because the slide valve and successively the output hydrostatic motor are fed by a constant flow rate pump.

But it can start also from the compressibility coefficient definition:

$$k = \frac{1}{\varepsilon} = -\frac{\mathrm{d}V}{V_l\cdot\mathrm{d}p}\ \left[\mathrm{Pa}^{-1}\right]. \tag{3}$$

The relation (3) in finite differences pursuits to the expression of the displacement error of the plunger, Δs having the stroke $s\,[\mathrm{m}]$:

$$\Delta s = \frac{p}{\varepsilon}\cdot s\ \left[\mathrm{m}\right]. \tag{4}$$

Considering the plunger speed obtained from the continuity equation:

$$v_p = \frac{Q}{A}\left[\frac{\mathrm{m}}{\mathrm{s}}\right], \tag{5}$$

in which A is the surface area of the active face of the plunger, it results the time for reach the working pressure in the hydraulic motor:

$$t_p = \frac{\Delta s}{v_p} = \frac{p}{\varepsilon}\cdot\frac{s\cdot A}{Q}\,[\mathrm{s}]. \tag{6}$$

From the work [1] it results an error of about 0,03 mm due to the oil compressibility in the linear hydraulic motor.

The time t_a for accelerating of actuating masses $m\,[\mathrm{Ns}^2/\mathrm{m}]$ until reaching the regime velocity, $v_p\,[\mathrm{m/s}]$ results from integration of the relation:

$$m\cdot\frac{\mathrm{d}v}{\mathrm{d}t} = A\cdot p - F\,\left[\mathrm{N}\right], \tag{7}$$

$F\,[\mathrm{N}]$ being the sum of resistance forces reduced to the piston.

The hydraulic shock appears like an overpressure Δp at the suddenly closing of the distribution device and is propagated in conduits with length L which connect this device with the hydraulic motor, the wave velocity being a . There are known formulas Jukovski for the maximum pressure due to the hydraulic shock, the real velocity of the hydraulic shock in a conduit, the continuity equation applied for the fluid flow in conduit and the design equation of the conduit wall:

$$\Delta p = \rho \cdot a \cdot v \;[\mathrm{Pa}]; \tag{8}$$

$$a = \sqrt{\frac{\frac{\varepsilon}{\rho}}{1 + \frac{\varepsilon}{E} \cdot \frac{d}{\delta}}} \left[\frac{\mathrm{m}}{\mathrm{s}}\right]; \tag{9}$$

$$v = \frac{4 \cdot Q}{\pi \cdot d^2} \left[\frac{\mathrm{m}}{\mathrm{s}}\right]; \tag{10}$$

$$\delta = \frac{p \cdot d}{2 \cdot \sigma_a} + \text{allowance } [\mathrm{m}]. \tag{11}$$

in which ρ is the liquid density; E – module of elasticity of the conduit material, having the design working stress σ_a; d - inner diameter of the conduit; δ - wall conduit width.

The formula (8) corresponds to the direct, positive hydraulic shock with the maximal intensity, when the liquid flow is closed in a shorter time t_i than the reflection time of the wave, $2 \cdot L/a$:

$$t_i \le \frac{2 \cdot L}{a} \;[\mathrm{s}]. \tag{12}$$

For the technical usually values, $\varepsilon = 1.6 \cdot 10^9 \mathrm{Pa}$, $E = 2.1 \cdot 10^{11}$ Pa, $\rho = 900 \mathrm{\ kg/m^3}$, relations (8) and (9) become:

$$\Delta p = 12.11 \text{ bar}; \tag{13}$$

$$a = \frac{1320}{\sqrt{1 + \frac{1}{131} \cdot \frac{d}{\delta}}} \left[\frac{\mathrm{m}}{\mathrm{s}}\right]. \tag{14}$$

The hydraulic shock in many conduits *n*, in which the liquid flow is closed simultaneously produces an overpressure calculated with relation (8) too, but in which is considered the velocity:

$$a = \frac{L_1 + \ldots + L_n}{\frac{L_1}{a_1} + \ldots + \frac{L_n}{a_n}}; \tag{15}$$

and

$$v = \frac{L_1 \cdot v_1 + \ldots + L_n \cdot v_n}{L_1 + \ldots + L_n}, \tag{16}$$

in which $L_1 \ldots L_n$ are length of the *n* pipes before the distribution closing; $a_1 \ldots a_n$ - velocities of shock propagation through the liquid from conduits (usually $a_1 = a_2 = \ldots = a_n$); $v_1 \ldots v_n$ - velocities of liquids through the *n* pipes before the distribution closing.

The hydraulic shock in a pipe in which the liquid flow is closed in a time $t_i > 2 \cdot L/a$ is indirect, positive and attenuated with the ratio:

$$\frac{\frac{2 \cdot L}{a}}{t_i} < 1, \tag{17}$$

that should correct the relation (8):

$$\Delta p_{att} = \frac{2 \cdot L}{a \cdot t_i} \cdot \Delta p = \frac{t_i}{t'_i} \cdot \Delta p = \rho \cdot 2L \cdot \frac{v}{t'_i} \left[\frac{\mathrm{N}}{\mathrm{m}^2}\right]. \tag{18}$$

It results the acting time, based on the main components:

$$t = t_F + t_p + t_{shock} \ . \tag{19}$$

4. APPLICATION

In order to compute real examples, it is considered a hydraulic-mechanic servo-steering like in Figure 2, the cylinder having: $D = 100$ mm, $s = 100$ mm, $Q = 12\ l/\text{min}$, $p = 50$ bar; the pipe: $d = 10$ mm, $\delta = 1$ mm, $E = 2.1 \cdot 10^{11}$ Pa; the fluid (mineral oil): $\rho = 900$ kg/m^3, $\varepsilon = 1.6 \cdot 10^9$ Pa.
With formula (4) it is calculated the displacement error of the plunger:

$$\Delta s = \frac{50 \cdot 10^5}{1.6 \cdot 10^9} \cdot 100 = 0.031 \text{ mm} \ . \tag{20}$$

From formula (5) it results the time for reach the working pressure in the hydraulic motor:

$$t_p = 1.18 \cdot 10^{-3} \ \text{s} \ . \tag{21}$$

With formula (9) it results:

$$a = 1280 \ \frac{\text{m}}{\text{s}} \tag{22}$$

and with (8):

$$\Delta p_{\max} = 29.4 \ \text{bar} \ . \tag{23}$$

With formula (10) it results the velocity in a steady state regime:

$$v = 2.54 \ \text{m/s} \ . \tag{24}$$

Considering $t_i = 0.04$ s, the maximum hydraulic shock is obtained for a pipe having a length:

$$L = \frac{a \cdot t_i}{2} = 25.6 \ \text{m} \ , \tag{25}$$

a huge length for the considered case. Thus, the hydraulic shock is indirect.
The slide valve distributor is near the hydrostatic motor and a value of $L = 1$ m for that, pursue to the reflection time of the wave:

$$\frac{2 \cdot L}{a} = 1.56 \ \text{s} << t_i \ , \tag{26}$$

the attenuated hydraulic shock becoming: $\Delta p_{att} = 1.16$ bar, that is acceptable.
The time for running the pipe is:

$$t_{shock} = \frac{L}{a} = 0.78 \cdot 10^{-3} \ \text{s} \ . \tag{27}$$

It results the acting time, based on the main components:

$$t = (0.04...0.08) + 1.18 \cdot 10^{-3} + 0.78 \cdot 10^{-3} = 0.042...0.082 \ \text{s} \ . \tag{28}$$

The maximum value is near the delay time due to the human reflex, $0.1s$.
This time, the automotive having the velocity v_a covers a distance:

$$D = v_a \cdot t \ [\text{m}] \ . \tag{29}$$

Distances covered in the automotive bend for tracks and cars, function of speed is presented in figure 1.
The steering wheels deflection, thus the automotive bend, starts after hydraulic amplifier coupling. The speed of wheel rotation is usually about 10…15 deg/s for tracks, and 90 deg/s for cars, that is a rotating time of 6…8 s in

the 1st case and 2.5…3 s in the 2nd one. Thus, the "delay" expressed in wheel rotation angle is 1.9…4.92 deg for tracks and 5.04…11.8 deg for cars. It can consider a speed at the wheel $n_v = 60...70 \ \text{rev/min}$.

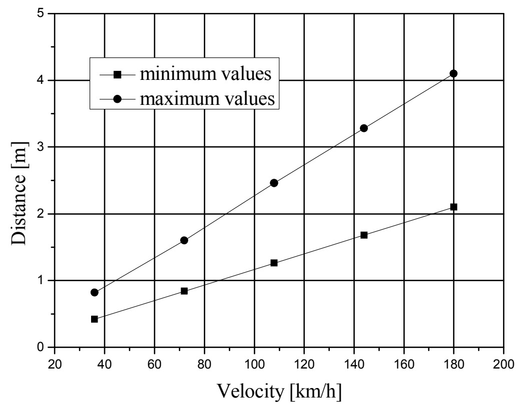

Figure 1: Distance covered in the delay time

Table 1 presents the covered distance in the delay time and, for comparison the covered distance for a single rotation of the wheel function of the velocity for tracks and cars. In the case of high velocities used by cars on highway, the distance is relatively significant.

Table 1: The distance covered in the "delay" time

Velocity [km/h]	36	72	108	144	180
Covered distance in "delay" [m]	0,42-0,82	0,84 -1,64	1,26-2,46	1,68-3,28	2,10-4,10
Covered distance in 6..8 s (tracks) [m]	60-80	120-160	180-240	240-320	300-400
Covered distance in 2.5…3 s (cars) [m]	25-30	50-60	75-90	100-120	125-150

5. CONCLUSION

This study attempt to establish the turning delay, by computing the time of the dynamic coupling through the sum of partial times due to the slide valve control displacement, the working fluid compressibility and the elastic propagation of the pressure shock. It results times having values close the human reflex time and relatively significant values for the covered distances.

Thus, the practical possibility for emphasise determinant factors permits to take efficient measures for the dynamic improvement of hydraulic servo-steering beginning with design, construction and exploitation.

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