U.P.B. Sci. Bull., Series D, Vol. 75, Iss. 1, 2013 ISSN 1454-2358

CONTRIBUTION TO THE STUDY OF U-650 M TRACTOR'S POWER STEERING BEHAVIOR IN DIFFERENT OPERATION CONDITIONS

Stelian ANGHEL[1]

Acest studiu işi propune să analizeze solicitările servodirecţiei tractorului U-650 M, în condiţiile deplasării pe diferite tipuri de teren şi cu raze de viraj diferite. În acest mod se face o legatură directă între condiţiile de lucru şi forţa de rezistenţă care apare la tija pistonului servomotorului hidraulic. Bazat pe modelul dinamic echivalent al tractorului şi utilizând un program de simulare se poate urmări evoluţia principalilor parametri ai servodirecţiei în cele mai dificile condiţii de lucru şi se pot trage concluzii cu privire la comportamentul tractorului în exploatare.

This study aims to review the loads for tractor power steering U-650 M, according to travel on different types of terrain and different turning radius. In this way there is a direct link between working conditions and the force of resistance that occurs in the hydraulic actuator piston rod. Based on the dynamic equivalent model of the tractor and using a simulation program, we can follow the main parameters of the power steering in the toughest working conditions and can draw conclusions about the behavior of the tractor in operation.

Keywords: steering wheels, resistance force, hydraulic actuator, critical speed, rolling resistance coefficient, coefficient of adhesion, dynamic stability**.**

1. Introduction

According to [14] the effort of adapting a hydraulic mechanism acting at the direction of a vehicle follows the choosing of constructive optimal parameters, which ensures stable operation, accurate imposed. This choice should be made taking into account all factors that influence the functionality of the system.

To establish a functional conditions of a hydraulic servosystem means to determine the maximum steering angle of the steering wheel (the condition that the minimum turning radius of the vehicle), minimum turning time, the opposing forces and the turning moments .

The hydrostatic motor of the power steering should provide the turning of steering wheels between the two extreme positions, beating the resistant torque, given by the friction between tire and ground, due to their balancing, the torque of

[1] PhD Student, University POLITEHNICA of Bucharest, Faculty of Biotechnical Systems Engineering, Romania, e-mail: stelica_anghel@yahoo.com

inertia of the wheels in balancing movement and friction forces in the steering system bearings and joints.

Depending on the conditions of turning, it highlights three characteristic values of the torque resistance can be highlighted: maximum torque occurs at turning the steering wheel in an extreme position to another, on a concrete or asphalt dry clothing, if the tractor is fully charged and immobile; same torque, but the steering wheels in the direction of drift angle limits to start slipping and the torque meeting in normal working conditions. The greates torque is the first, lowest - the third.

A large tractors due to heavy loads on the steering wheels, elevated forces are required from the driver exceeding the allowable limit, especially when moving on deformable terrain, or passing over the bumps, [1].

U-650 M tractor is a medium power tractor equipped with a diesel engine of *65* hp at *1800* rev / min with direct injection and electric start .

The wide range of use of the tractor is provided by independent or synchronous PTO , high speed range (*2.63...27.32 km/h*) and a possibility of defeating the effort while working through the transition from a high range lower gear to another without stopping the tractor, using the torque amplifier.

Reduction of the tractor driving effort was solved by adopting a hydraulic steering servo.

If the pump system does not work, the steering is mechanical, the piston-rack is moved in the cylinder due to screwing (or unscrewing) the threaded shaft in the piston.

Steering system elements were calculated either from the maximum force applied by driver to wheel or considering the maximum resistance that occurs when turning the steering wheels, which can now be determined based on empirical relationships.

In this study we aimed at finding a relationship between the resistant torque of steering wheels and the resistance force that occurs in hydraulic actuator, in different working conditions.

We used this study to simulate the tractor power steering on different terrain and to adopt constructive solutions for its improvement .

2. Material and methods

Dynamic stability of tractor

Total weight of the tractor is determined by the relationship [1]:

$$G_t = m_t \cdot g \ [N] \tag{1}$$

where m_t is the operation tractor mass ($m_t = 3380\ kg$); g – gravitational acceleration [m/s^2].

Because I wanted to study the dynamics of the tractor and default steering conditions in difficult situations, I have chosen for the calculations $g = 10\ m/$.

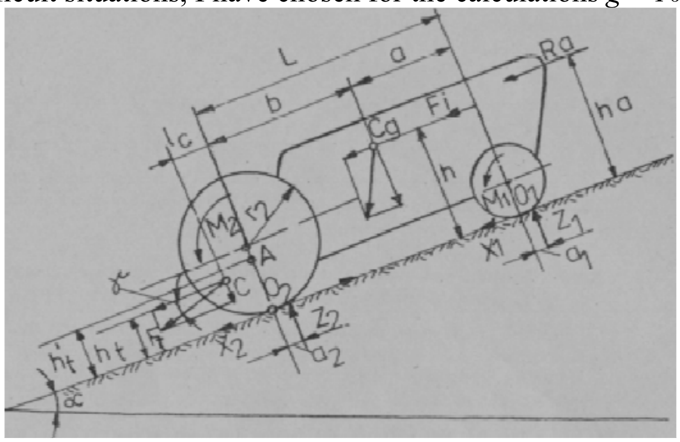


Fig. 1. Model of the calculation for tractor

Coordinates of center of gravity :

$$a = G_2 . L / G_t ,\ m \tag{2}$$

where G_2 - rear axle load, N.

$$a = 2210.2,43 / 3380 = 1.58\ m \tag{3}$$

$$b = L - a = 2.43 - 1.58 = 0.85 m \tag{4}$$

Center of gravity height is determined experimentally, for the formula 4x2 wheeled tractors is between *0.9...1.1 m,* [3]. We choose $h = 1\ m$ for calculations.

Critical speed of the tractor rollover on a horizontal road is determined with the relationship :

$$v_{cr} = \sqrt{\frac{g.E.R}{2.h}}\ [m/s] \tag{5}$$

where: R – turning radius, m; E – average track, $E = (E_1 + E_2) / 2$ – front and rear wheels track, m; h – height of center of gravity of the tractor, m.

$$E = (1,32+1,4) / 2 = 1,36\ m \tag{6}$$

$$R_{min} = 3,64\ m; \quad h = 1\ m$$

Critical slip speed of the tractor on a horizontal road is determined with the relationship :

$$v_{ca} = \sqrt{g.\varphi.R}\ [m/s] \tag{7}$$

Calculation of the hydromechanical steering of U-650 M tractor

The gear ratio of forces is defined as the ratio between the amount of resistance forces F_r acting on the two steering wheels, the point of contact with the road surface, at the distance c from the point of intersection with the road surface pivot axis, and the force F_v applied to steering the wheels [1]:

$$i_F = \frac{F_r}{F_v} \tag{8}$$

Steering force applied F_v can be expressed depending on the torque applied for steering wheels M_v and steering wheel radius r_v:

$$F_v = \frac{M_v}{r_v} \tag{9}$$

We note „r_v” the steering wheel radius so it cannot be confused with „R_v”- turning radius.

The steering wheel radius for U-650 M tractor is r_v = 225 mm.

Resistant torque for turning the wheels when the tractor is at rest (torque required to be applied to the steering wheel pivots) is determined with the relationship:

$$M_r = M_{r_1} + M_{r_2} \tag{10}$$

where M_{r_1} is the resistant torque of steering wheels and M_{r_2} the friction torque of the wheels.

$$M_{r_1} = G_1 . f_1 . c \tag{11}$$

G_1 -static load on front axle;

f_1 - rolling resistance coefficient of steering wheels;

c - deported wheel; for existing tractors $c = 100...300\ mm$.

From the paper [15], we obtain the following values of the coefficient of rolling resistance, from experiments performed in different driving conditions:

Table 1

Coefficient of resistance for tractors

State land	Coefficient of resistance for tractors	
	wheels with tires	Caterpillar
Field dry road	0.03 – 0.05	0.05 – 0.07
Compact celery,meadow	0.05 – 0.07	0.06 – 0.07
Stubble	0.08 – 0.10	0.07 – 0.09
Plowing	0.15 -0.18	0.09 – 0.11
Land harrowing	0.16 – 0.19	0.09 – 0.11
Muddy	0.25 – 0.30	0.10 – 0.25
Trodden snow road	0.03	0.06

The friction torque of wheels, M_{r_2} is calculated with :

$$M_{r_2} = 0.14.G_1.\varphi.r_{1_S}, \tag{12}$$

where φ is coefficient of adhesion of the wheels with the way, and r_{1s}, static radius of steering wheel.

From [15], the following values of the coefficient of adhesion result:

Table 2

Coefficient of adhesion for tractors

State land	Coefficient φ for tractors	
	wheels with tires	Caterpillar
Compact celery	0.6 – 0.7	1.0 – 1.2
Stubble	0.5 – 0.7	0.8 – 0.9
Plowing	0.3	0.4 – 0.5
Land prepared for sowing	0.4 – 0.6	0.6 – 0.7
Muddy	0.1	0.4 – 0.5
Tamping snowy road	0.2 – 0.3	0.6 – 0.7

As the gear ratio of forces is higher, both drive wheel requires less effort (i_F = *100...300*).

Simulation of U-650 M tractor steering

For simulation we used the AMESim program. It offers a complete 1D simulation of intelligent systems and predicts their performance .

The model components are described by analytical models representing the behavior of subsystems hydraulic, pneumatic, electrical or mechanical parts of the system and overall system behavior.

The simulation that follows after the development scheme includes the following steps:

- mathematical modeling of components associated to icons ;
- defining the parameters of components;
- execution of numerical integration;
- interpretation of results describing the behavior of the system by appropriate graphics whith the possibility to export the resulting data.

In addition to the regular hydraulic scheme that includes common elements (pump, control valve, hydraulic cylinder, valves, tank, pipes, etc.), the simulation scheme allows comparison of amplified control signal for moving the piston rod to report any deviation of normal stroke (actually the feed-back that transmits via servo levers any deviation to the control valve for correcting it) and implementation of the resistance force actuator piston rod through a force signal and a power converter which transforms the signal in force (Fig. 2).

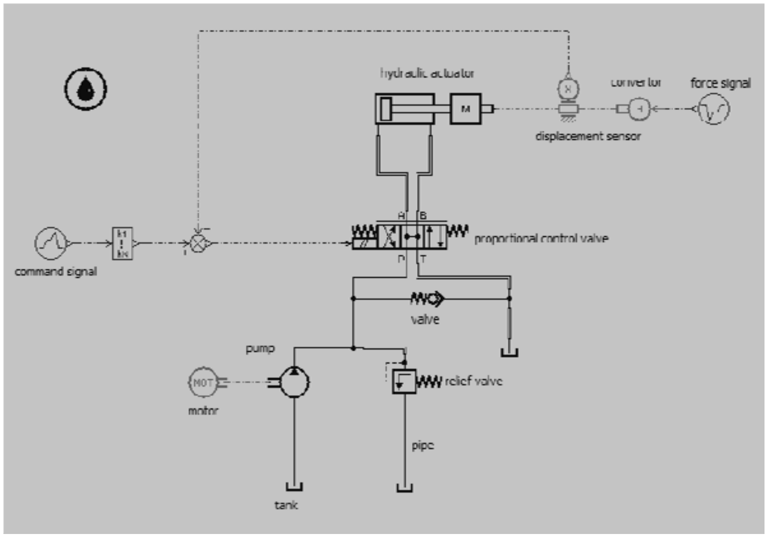


Fig. 2. The U-650 tractor power steering scheme in AMESim

3. Result

According to the relationships *5.6* and *7*, and , where φ = *0.7* the coefficient of adhesion (on dry roads , beaten or asphalt).We chose this ratio because the steering system following calculations, when the greatest resistance occurs at maximum adhesion coefficient (*0.7*) .

Minimum adhesion coefficient (φ = *0.1*), leads to the tractor wheel slip, but no effect on resistance when turning [1].

The static load on front axle is $G_1 = 11700\ N$.

The parameter c is considered $c = 200mm$.

Steering wheels tire dimensions are *6.5 – 20 inch*. For these tires, it results [1] :

- section width [mm] – *180*;
- outside diameter [mm] ± *1.5*% - *868*;
- static radius [mm] ± *1.5*% - *408*.

According to relationship *10*:

(13)

Driving force calculation and torque, in the absence of hydraulic amplification, are :

(14)

(15)

where is the angle gear ratio; - mechanical efficiency of the steering (0.95...0.98).

Angle gear ratio (kinematically) i_ω, is the ratio between the angle of rotation of the wheel φ_v, and the average angle of turning of the steering wheels θ:

$$i_\omega = \frac{\varphi_v}{\theta} \tag{16}$$

Angle gear ratio is chosen so that the rotating wheel with 1.5...2.2 revolutions, turning steering wheel maximum does not exceed 35-40^0.

For example, if we consider two revolutions of the wheel and $\theta = 40^0$,

$$i_\omega = \frac{720^0}{40^0} = 18 \tag{17}$$

Driving torque calculation is $M_{va} = \frac{M_v}{k}$; and k is the amplification factor ($k = 2 \cdots 6$).

Considering $k = 4 \Rightarrow M_{va} = \frac{M_v}{4} = \frac{M_r}{4 \cdot i_\omega \cdot \eta_m}$; (18)

From the paper [14] is shown that the hydraulic power amplifier can be calculated with the relationship:

$$P_{ch} = \frac{\pi}{30}\left(\frac{M_r}{i_\omega} - M_{va}\right)\frac{n_v}{\eta_{ma}}\ [W]; \tag{19}$$

where n_v is the wheel rotation speed (RPM) by the driverone may consider $n_v = 60 \cdots 70\ rot/min$, where $\eta_{ma} = 0{,}91 \cdots 0{,}93$ is the mechanical efficiency of the amplifier .

Given the above, the hydraulic power will be :

$$P_{ch} = \frac{\pi}{30}\left(\frac{M_r}{i_\omega} - \frac{M_r}{4 \cdot i_\omega \cdot \eta_m}\right) \cdot \frac{n_v}{\eta_{ma}} = \frac{\pi}{30} \cdot \frac{M_r}{i_\omega}\left(1 - \frac{1}{4 \cdot \eta_m}\right) \cdot \frac{n_v}{\eta_{ma}} \tag{20}$$

The shareholders speed (minimum speed cornering, considered to the work cylinder piston) must not be less than the amplification that can be achieved without the contribution of hydraulic head, as the workload:

$$v_{min} \geq \frac{\pi \cdot n_v \cdot r_v}{30 \cdot i_{v-m}} \tag{21}$$

where i_{v-m} is the gear ratio from the steering wheel to the hydrostatic actuator.

The resistance force to the hydraulic actuator piston rod will be :

$$F_r = \frac{P_{ch}}{v_{min}} = \frac{M_r}{i_\omega} \cdot \left(1 - \frac{1}{4 \cdot \eta_m}\right) \cdot \frac{1}{\eta_{ma} \cdot r_v} \tag{22}$$

Considering $\eta_m = 0.95$; η_{ma} =0.92; r_v =0.225 *m*; i_ω =*18*, we can determine the resistance force to the hydraulic cylinder rod :

$$F_r = 468.f_1 + 117.\varphi\ [\mathrm{N}] \tag{23}$$

Field dry road

According to tables *1* and *2*:

$$F_r = 23.4 + 117 \cdot \frac{v^2}{10 \cdot R} \tag{24}$$

We use the notation *v* instead of v_{ca} for reaseon of simplicity .

We consider a range of values of *v* between *0.7 m/s* (*2.52 km/h*) - the lowest speed the slow group and *7.45 m/s* (about *27 km/h*) - the highest speed the fast group. (Technical Notes of the tractor U-650 M).

From the previous calculation it results, however, that critical speed can not exceed *5 m/s* (*18 km/h*) the minimum turning radius. In these conditions, it follows that resistance to the rod piston of actuator will be lawer.

Since the overthrow of the cross is more dangerous than sliding side, we recommend that the speed limit skid v_{ca} to be less than the speed limit rollover v_{cr}. It is preferable that the side slip appears before the tractor rollover [2].

$$\sqrt{g \cdot \varphi \cdot R} < \sqrt{\frac{g \cdot R \cdot E}{2 \cdot h_g}} \quad (25)$$

or

$$\varphi < \frac{E}{2 \cdot h_g} \quad (26)$$

If we consider E (average track) 1.36 m and h_g =*1 m* (height of center of gravity of the tractor), we get:

$$\varphi < \frac{1{,}36}{2} = 0.68 \quad (27)$$

Note that for a variable $\varphi = 0{,}7$ (field dry roads, asphalt, paved road, etc..), if the condition is not satisfied then we can enlarge the track (we considered the minimum track both the front axle and rear axle), or descend the center of gravity by using ballast on the tractor.

But if we limit the speed to *5 m/s* (*18 km/h*), graphically representing $F_r = f(v, R)$, we see that the force of resistance to the hydraulic actuator decreases by the turning radius, which is normal because the situation remains dangerous to the minimum turning radius when the tractor could rollover.

To maintain stability of the tractor and at the minimum turning radius the resistance force to the hydraulic actuator will have a maximum of *104 N* (fig. *3*).

Muddy field

Similarly we can get these features under other ground conditions, but the worst is moving on a muddy field, where speed must be limited to *1.9 m/s* in order to maintain stability of the tractor. At the minimum turning radius the resistance force to the hydraulic actuator will have a maximum of *152 N* (fig. *4*). We obtain the graphs below ($F_r \rightarrow z;\ v \rightarrow x;\ R \rightarrow y$):

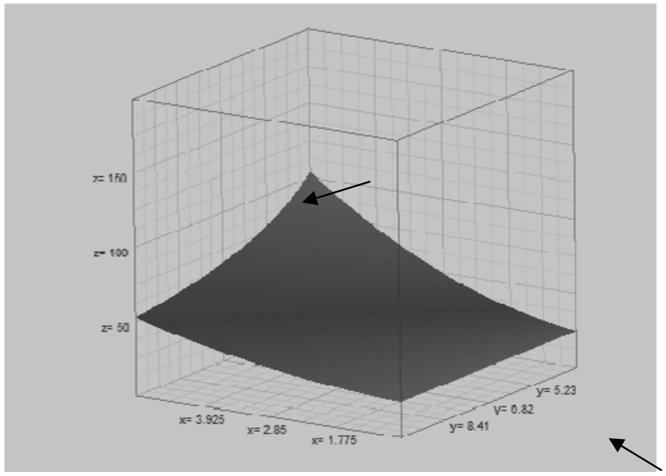


Fig. 3. Force of resistance to the hydraulic actuator depending on the critical speed and the turning radius at the displacement on field dry road

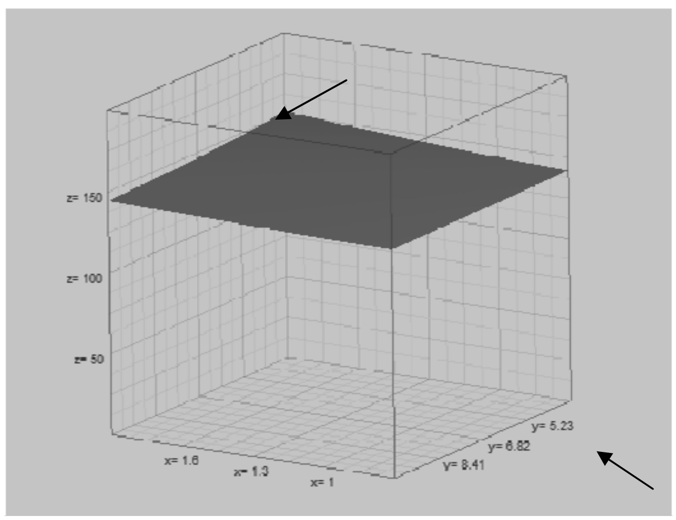
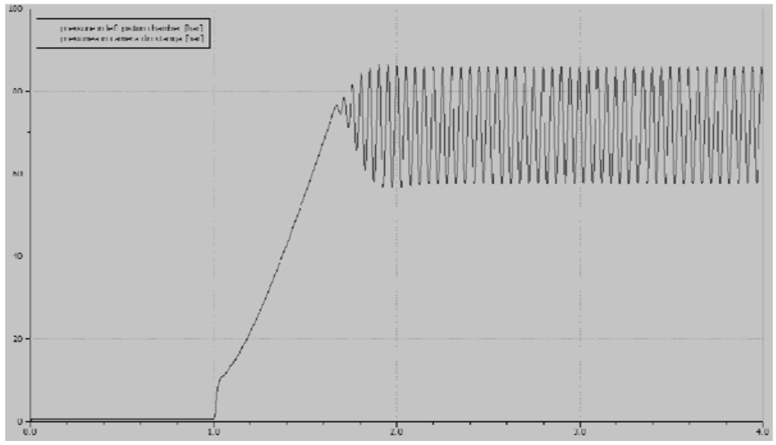


Fig. 4. Force of resistance to the hydraulic actuator depending on the critical speed and the turning radius at the displacement on muddy field

Hydraulic simulation results

Field dry road, $v = 0.7$ m/s, $R = 10$ m, $= 24$ N



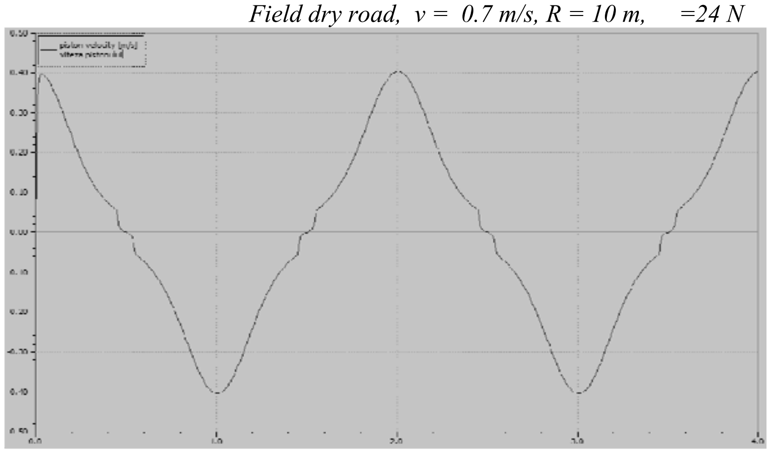
time [*s*]

Fig. 5.The variation of pressure in the left chamber of hydraulic actuator at the displacement on field dry road

Note that the simulation program shows a value of about *7.8 MPa*, which is normal because the maximum pressure can not exceed *8 MPa*.

The speed of the piston will have the following characteristics (note that *v = 0.4 m/s*):

Field dry road, v = 0.7 m/s, R = 10 m, =24 N

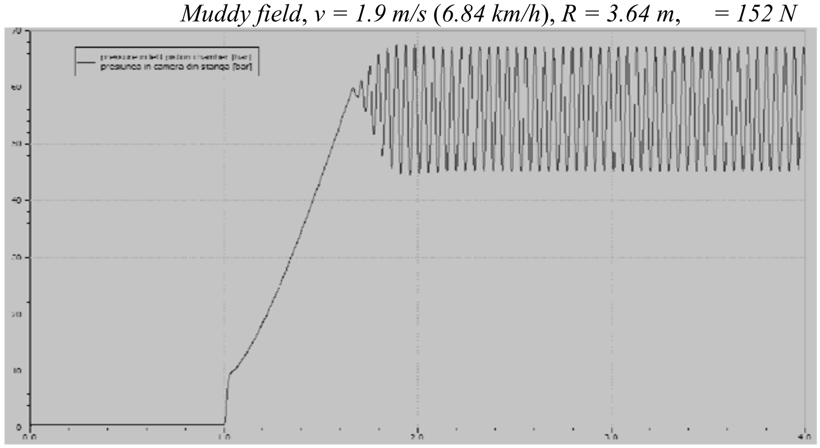


time [*s*]

Fig. 6.The variation of piston velocity at the displacement on field dry road

We now look at the worst case, where the tractor turns to a muddy field with the minimum turning radius and the speed limited to *1.9 m/s* (*6.84 km/h*) , imposed by the conditions of stability. In this case, the resistance force to the hydraulic actuator rod reaches of *152 N*. We see that the pressure in the hydraulic actuator chamber decreases up to *6.5MPa*. We will also observe the lower piston speed is *0.21 m/s* (figs. 7, 8):

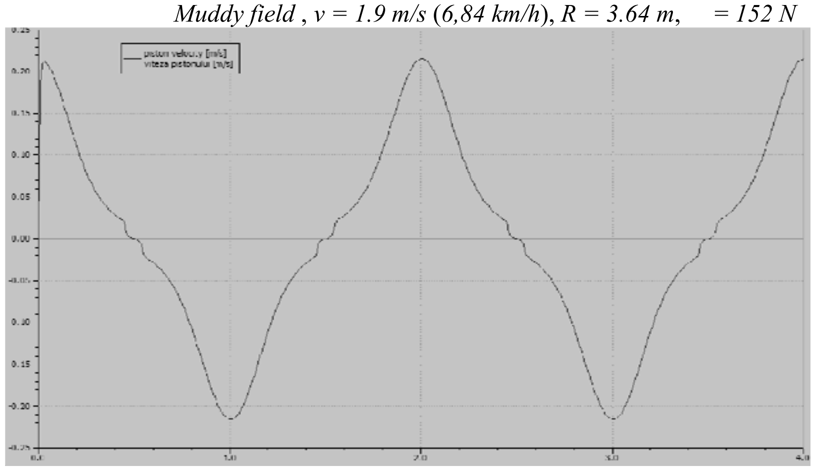
Muddy field, $v = 1.9$ *m/s* (*6.84 km/h*), $R = 3.64$ *m*, $= 152$ *N*



time [*s*]

Fig. 7.The variation of pressure in the left chamber of hydraulic actuator at the displacement on muddy field

Muddy field , $v = 1.9$ *m/s* (*6,84 km/h*), $R = 3.64$ *m*, $= 152$ *N*



time [*s*]

Fig. 8.The variation of piston velocity at the displacement on muddy field

4. Conclusions

That the pressure of the hydraulic cylinder must be between *6* and *8 MPa*, corresponding to the servo system hydraulic resistance encountered in normal operating conditions, it was shown [16]. We see then, at least in the latter case that, the pressure reaches the allowed limit and it is possible that under certain conditions (normal wear steering system bodies, variation coefficient of rolling resistance, etc.) it falls below this value, which means effort especially from the driver to control the tractor and not to lose stability in turning. Do not forget, however, that portions of dry field operation may be interspersed with wet parts, which further hampers the task leader in terms of steering the tractor.

The input signal (action driving) varies often sharply, in which case the forces of inertia and friction can reach high values. External disturbing signals (shock transmitted from the steering wheel) can cause, under certain conditions, autooscillations of the hydrostatic system. Consideration of these factors is difficult because, unlike the static regime, where the hydrostatic actuator can be considered isolated from the whole tracking system, in the dynamic regime it is necessary to consider the system as a whole.

Therefore the calculation of hydraulic steering take into account the observations mentioned before.

R E F E R E N C E S

[1] *S. Nastăsoiu, C. Andreescu, S. Popescu, Gh. Fraţilă, D. Cristea*, Tractoare (Tractors). Editura Ceres, Bucureşti, 1987; (in Romanian)

[2] *Tiberiu-Nicolae Macarie*, Automobile Dinamica (Cars Vehicle Dynamics). Editura Universitaţii din Piteşti, 2003; (in Romanian)

[3] *St. Tabacu, I. Tabacu, T. Macarie, E. Neagu*, Dinamica Autovehiculelor, Îndrumar de proiectare (Cars Vehicle Dynamics, Design Guidelines), Editura Universităţii din Piteşti, 2004; (in Romanian)

[4] *P. Babiciu V., Scripnic*, Sistemele hidraulice ale tractoarelor şi maşinilor agricole (Hydraulic systems of tractors and agricultural machinery). Editura Ceres, Bucureşti, 1984; (in Romanian)

[5] *R. Rosca, V. Valcu*, Acţionări hidraulice şi pneumatice (Hydraulic and pneumatic drives). Editura "Ion Ionescu de la Brad" Iaşi, 2000; (in Romanian)

[6] *I. Oprean , M. Andreeescu*, Transmisii automate pentru autovehicule (Automatic transmission for vehicles). **Vol. I**. Transmisii hidraulice, Edit. Politehnica Bucureşti, 1997;(in Romanian)

[7] *T.D. Gillespie,* Fundamentals of Vehicle Dynamics, SAE, Warrendale, PA, 1992;

[8] *J. Lennevi, J. Palmberg, and A. Jansson,* Simulation Tool for the Evaluation of Control Concepts for Vehicle Drive Systems. In Proceedings of the 4th Scandinavian International Conference on Fluid Power, Tampere, September 1995;

[9] *J.Schwarzenbach and K.F.Gill*, System Modelling & Control, Edward Arnold, 1992;

[10] * * * - Documentaţie (Documentation) AMESim;

[11] * * * - Documentaţie (Documentation) Parker Hydraulics;

[12] *M.Jelali and A.Kroll*, Hydraulic Servo Systems - Modelling, Identification & Control, Springer, 2003;

[13] *D.DeRose*, Proportional and Servo Valve Technology, Fluid Power Journal, March/April 2003;

[14] *P. Alexandru, Fl. Dudiţă, A. Jula, V. Benche*, Mecanismele direcţiei autovehiculelor (Vehicle steering mechanisms), Editura Tehnică, Bucureşti, 1977; (in Romanian)

[15] *D. Tomescu, Gr. Răileanu, I. Vlădăşel*, Întreţinerea şi repararea tractoarelor (The maintenance and repair of tractors), Editura Ceres, Bucureşti, 1982; (in Romanian)

[16] *D. Tomescu, I. Maraloiu*, Organizarea şi dotarea atelierelor de întreţineri şi reparaţii din agricultură (The organization and equipment maintenance and repair workshops in agriculture), Editura Ceres, Bucureşti, 1982; (in Romanian)

[17] * * * Notiţa tehnică a tractorului U-650 M (TheTechnical Notice U-650 M tractor).