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# *A Principal Diagram and a Mathematical Model Development of the Lateral Stabilization System Operation of a Steep-Stabilized Small Tractor*

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**Abstract – The article presents the theoretical studies results to determine the mathematical model of the lateral stabilization system operation of the position of the steep small-sized wheeled tractor platform control.**

**Keywords – Steep-Sided Small-Sized Tractor, Control Platform, Bogie, Automatic Stabilizer, Lateral Stability, Leveling Mechanism, Lateral Stabilization System, Mountain And Foothill Slopes.**

## I. Introduction

The agricultural production of our republic employs 55-60% of the country's population, while in developed countries it is 5-10%. At the same time, the population of the republic is growing rapidly (by 450-550 thousand people a year), and land, as a means of production, does not expand, does not renew or change. At the same time, an important reserve for food production, both in our country and abroad, is the further development of mountainous regions for agricultural land. Of the total 44,410.3 thousand hectares of the republic, 23.4% falls on the foothill zones. In mountainous conditions, special tractors with increased lateral stability are used - slopes [1,2,3].

A characteristic feature of mountainous regions is that with an increase in the steepness of slopes, the areas of cultivated areas decrease, which often do not exceed 0.5 ... 1.5 hectares. The use of high-performance agricultural machinery in such "small-contour" areas is either not economically justified or technically impossible. Therefore, an increase in food production on the basis of these lands can be carried out only with the widespread use of small-sized agricultural machinery, which will lead to a sharp (tens of times) increase in labor productivity of the population employed in these jobs [4,5].

On the ST operator it is possible to stabilize the MP of the alignment of the management platform (MP) relative to the truck. MP is part of the ST steep structure, consisting of a seat, operator, and control system, pivotally connected to the frame. A trolley is a part of a structure, consisting of a frame, wheels, an engine and a transmission [6,7].

## II. Materials and Methods

The lateral stabilization system (LSS) of the lateral oscillations of the steep-slope ST is designed to maintain the vertical position of the MP tractor during its operation on slopes, the schematic diagram of which is shown in Fig. 1 [8].

The LSS operation principle is as follows. When the tractor MP is tilted, the pendulum 4 under the action of gravity, trying to maintain a vertical position, moves the distributor spool. The oil flow from pump 2 through the distributor and shut-off valves enters the opposite cavities of the hydraulic cylinders and moves the hydraulic cylinder rods in different directions. Oil from adjacent cavities of the hydraulic cylinders returns to the hydraulic tank. The rods of the hydraulic cylinders rotate MP around the horizontal axis until it assumes a vertical position.

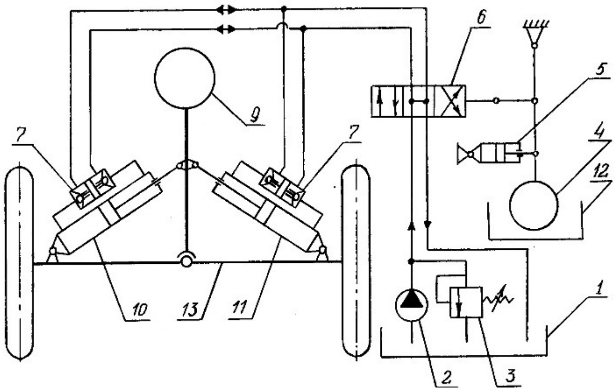


Fig. 1. Schematic diagram of СПС ПУ steeply sloping ST

1-hydraulic tank; 2-hydraulic pump; 3- bypass valve; 4-pendulum; 5- damping device; 6- distributor; 7 and 8-locking devices; 10 and 11-executive hydraulic cylinders; 12-MP; 13-frame.

Fig. 2

The LSS described above can be represented in the form of a block diagram shown in Fig. 2. Данная система является системой автоматического регулирования по отклонению, в которой измеряется отклонение $\varphi'$ угла крена МР $\beta$ от заданного значения $\varphi_o$, и в функции от измеренного отклонения, подается такое воздействие на регулирующий орган крутосклонного ST, которое уменьшает величину отклонения. All elements shown in the block diagram, with the exception of the control object 4, form an automatic regulator, consisting of a comparison and shaping device 1, a hydraulic booster 2, an executive hydraulic cylinder 3 and a roll sensor 5.

The functions of the roll sensor 5, the comparison device and the formation of the control action on the spool 1 are performed by the vertical pendulum sensor device. Blocks 2 and 3 characterize the hydraulic drive elements. Block 2 describes a spool booster, and block 3 describes a two-way hydraulic cylinder. The hydraulic booster 2 controls the oil flow $Q$ entering the hydraulic cylinder 3, which, in turn, causing the forces $F_{ц}$, moves the rod by the value $x_n$, acting on the regulating body of the control object 4. The dynamic characteristics of the hydraulic drive can be obtained in the form of differential equations.

The comparison device is implemented using a pendulum due to its tendency to maintain a vertical position regardless of MP position. In our case, the angle $\varphi_o$ is set by the initial position of the pendulum and has a zero value. The error in setting the angle $\varphi_o$ is determined by the value $\gamma$ and in the case $\varphi_o = 0$, $\varphi' = \varphi$.

## III. Results and discussion

Thus, for a complete description of the automatic stabilization system, it is necessary to obtain a mathematical model of the controlled object, the oscillatory motion of the pendulum and supplement them with a mathematical description of the dynamics of the hydraulic drive.

To derive a mathematical model of the pendulum, we will use the Lagrange equations of the second kind [9] and consider the design scheme of its oscillations, shown in Fig.3. We assume that the swinging plane of the pendulum coincides with the transverse vertical plane of the steeply sloping ST. As a generalized coordinate for the pendulum, we choose the γ angle rotation.

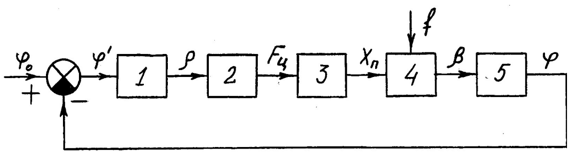


Fig. 2. Block stabilization system diagram of a steep-slope ST

Let us give the pendulum a possible displacement δγ and, calculating the sum of the work of active forces on this possible displacement and taking into account *sinγ≈γ*, we determine the generalized force

$$Q_\gamma = -m_m g l\gamma + \kappa\dot{\gamma} l_\partial^2 + \kappa\dot{\beta} l_\partial^2, \qquad (1)$$

where $m_m$ is the mass of the pendulum; $g$ - acceleration of gravity; $\kappa$ is damping factor [*н·с/м*]; $l_\partial$ -is distance from the suspension axis of the pendulum to the damper connection point; $\dot{\beta}$ -is angular rate MP rotation.

The kinetic energy of a pendulum performing a complex movement, consisting of a portable movement together with the ST cart due to the connection through the suspension and relative rotation relative to the suspension point, is expressed by the formula

$$\begin{aligned} T = \frac{1}{2} m_m (H_1 - H_3)^2 \dot{\theta}^2 + \frac{1}{2} m_m l^2 \dot{\gamma}^2 - m_m (H_1 - H_3)\cdot \\ \cdot l\dot{\theta}\dot{\gamma}\cos\theta - m_m (H_1 - H_3) l\gamma\dot{\theta}\dot{\gamma}\sin\theta, \end{aligned} \qquad (2)$$

where $\dot{\theta}$ -is angular velocity of ST cart oscillations.

After substitution of formulas (1) and (2), as well as the results of partial derivatives: - $\frac{\partial T}{\partial \dot{\gamma}}$, $\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\gamma}}\right)$ and $\frac{\partial T}{\partial \gamma}$, into the Lagrange equation of the second kind, we find the differential equation of motion of the pendulum

$$\begin{aligned} &\ddot{\gamma} + \frac{\kappa l_\partial^2}{m_m l^2}\dot{\gamma} + \frac{(H_1 - H_3)}{l}\left(\frac{g}{(H_1 - H_3)} - \sin\theta\ddot{\theta} - \cos\theta\dot{\theta}^2\right)\gamma = \\ &= \frac{(H_1 - H_3)}{l}\left(\cos\theta\ddot{\theta} - \sin\theta\dot{\theta}^2\right) + \frac{\kappa l_\partial^2}{m_m l^2}\dot{\beta}. \end{aligned} \qquad (3)$$

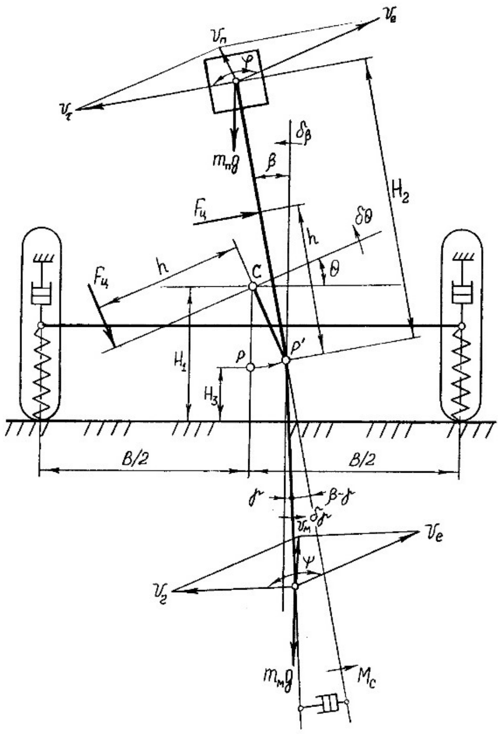


Fig. 3 Design scheme of oscillations of a steeply sloping SM

To derive the equation of motion MP, let us consider the design oscillation scheme for the case when the swing plane MP coincides with the transverse vertical plane ST (Fig. 3).

We give MP a generalized possible displacement $\delta\beta$ and define the generalized force

$$Q_{\beta} = m_n g H_2 \sin\beta - F_{\text{ц}} h - \kappa\dot{\beta} l_{\partial}^2 + \kappa\dot{\gamma} l_{\partial}^2. \qquad (4)$$

The kinetic energy MP rotating around the *z* axis has the form (5). Calculating the partial derivatives of the kinetic energy and using formulas (4), (5) and also taking into account that $\sin\beta \approx \beta$ we obtain the differential equation of motion MP.

$$\ddot{\beta} + \frac{\kappa l_{\partial}^2}{m_n H_2^2}\dot{\beta} - \frac{H_1 - H_3}{H_2}\left(\frac{g}{H_1 - H_3} + \sin\theta\ddot{\theta} + \cos\theta\dot{\theta}^2\right)\beta = $$
$$= -\frac{h}{m_n H_2^2}F_{\text{ц}} + \frac{\kappa l_{\partial}^2}{m_n H_2^2}\dot{\gamma} + \frac{H_1 - H_3}{H_2}\left(\cos\theta\ddot{\theta} - \sin\theta\dot{\theta}^2\right)\cos\beta. \qquad (6)$$

To determine the transverse-angular vibrational motion of the carriage of steeply sloping ST, we give the system a generalized possible displacement $\delta\theta$ and define the generalized force

$$Q_{\theta} = -\frac{RB}{2}(C_1 + C_2)\theta + \frac{RB}{2}(C_1 + C_2)\alpha - RB\mu_1\dot{\theta} + RB\mu_1\dot{\alpha} + +F_{\text{ц}}h - m_n g((H_1 - H_3)\sin\theta - H_2\sin\beta), \qquad (7)$$

where $R = \sqrt{(B/2)^2 + H_1^2}$.

The kinetic energy of the system is determined from the equation

$$T = \frac{1}{2}J_t\dot{\theta}^2 + \frac{1}{2}J_m\dot{\gamma}^2 + \frac{1}{2}J_n\dot{\beta}^2, \qquad (8)$$

where $J_t$ –is inertia moment of the tractor.

Calculating the partial and total derivatives of the kinetic energy of system (8) along the generalized coordinate $\theta$, as well as after substituting formulas (7) into the Lagrange equation of the second kind, we find the differential equation for the oscillation of the cart around the fixed longitudinal axis passing through its center of mass.

$$J_t\ddot{\theta} = -RB\mu_1\dot{\theta} - \frac{RB}{2}(\boldsymbol{C}_1 + \boldsymbol{C}_2)\theta - \boldsymbol{m}_n g\big((H_1 - H_3)\,\boldsymbol{sin}\,\theta - H_2\,\boldsymbol{sin}\,\beta\big) + +F_{ц}\boldsymbol{h} + RB\mu_1\dot{\alpha} + \frac{RB}{2}(\boldsymbol{C}_1 + \boldsymbol{C}_2)\alpha. \tag{9}$$

Separating the variables in equation (9) and substituting the value of the forced oscillations $\alpha=\alpha_o \sin pt$, $\dot{\alpha} = \alpha_o p \sin pt$ , and also taking $\sin\beta \approx \beta$, we finally find

$$\ddot{\theta} + \frac{RB\ \ _1}{J_t}\dot{\theta} + \frac{R\boldsymbol{B}(\boldsymbol{C}_1+\boldsymbol{C}_2)}{2J_t}\theta + \frac{\boldsymbol{m}_n g(H_1-H_3)}{J_t}sin\,\theta == \frac{\boldsymbol{m}_n gH_2}{J_t}\beta + \frac{F_{ц}\boldsymbol{h}}{J_t} + \frac{RB\mu_1 p\alpha_0}{J_t}cos\,p\,t + \frac{R\boldsymbol{B}(\boldsymbol{C}_1+\boldsymbol{C}_2)}{2J_t}\alpha_0\,sin\,p\,t, \tag{10}$$

Where $\mu_1$- is wheel tire damping factor; $\alpha_o$ – is a disturbing vibration amplitude; $p$ – is the circular frequency of the disturbing vibrations.

As a result of theoretical research, a system of differential equations describing the movement of the pendulum (3), MP (6) and the carriage (10) was compiled.

## IV. Conclusions

The numerical solutions analysis results of the mechanical-mathematical LSS model operation of the MP position of the steep-slope ST and the obtained dependences show that the stabilization system operates stably at average values of the alignment angular velocities up to 5.0 ... 5.6 deg/s. An increase in the oscillation frequency of the trolley $v$ ST leads to an increase in the value of the oscillation angle MP $\beta$. For example, with the oscillation angle amplitude of the $A = \pm 23°$ carriage and the $P = 2$ МПа pressure, an increase in $v$ from 0.08 to 0.024 Hz leads to an increase in $\beta$ from 0.04 rad to 0.113 rad.

## References:


[1] Kereselidze Sh.Ya., Razmadze G.N. Mechanization of mining agriculture and subtropical crops. - Tbilisi, Ganatleba, 1988.—558p.

[2] Alimukhamedov Sh.P., Mamasov Sh.A. Mountain tractors. // Agriculture of Uzbekistan. 2003. №11. 34 p.

[3] Reimer V.V. Substantiation of the methodology for increasing the efficiency of operation of wheeled tractors of class 1.4 when working on an inclined support surface / Abstract of the thesis. for a job. scientific degree of Cand. tech. sciences. Orenburg, 2012.

[4] Behrooz, M. and Hanif, N. Automatic control of a modified tractor to work on steep side slopes. Journal of Terramechanics, 2009 46, c.299–311.

[5] Jileikin M.M., Yagubova E.V., D.A. Chijov. Experimental study of stabilizing effectiveness the movement of a tractor when working on a slope in plowing mode on large-scale models/News of higher educational institutions.Mechanical engineering. Moscow, 2015. No.4.p57-62.

[6] Shigolev S.V. Investigation of lateral stability of self-propelled agricultural machines / Author's abstract of the dissertation for the degree of Cand. tech. sciences. Moscow, 2018.

[7] F. Mazzetto, M. Bietresato, A. Gasparetto, R. Vidoni. Simulated stability tests of a small articulated tractor designed for extreme-sloped vineyards / Journal of Agricultural Engineering 2013; volume XLIV(s2): e133

[8] Provisional patent 3552 (RUz). Mini tractor for work on mountain slopes/N.R. Rashidov, B.S.Suleymanov, Sh.A. Mamasov//Official. bul. -1996.-№2.

[9] Bat M.I., Janelidze G.Yu., Kelzon A.S. Theoretical mechanics in examples and problems: volume II. - 8th ed., Rev. - M.; Science, 1991.– 640p.