**Tuijin Jishu/Journal of Propulsion Technology**
**ISSN: 1001-4055**
**Vol. 45 No. 2 (2024)**

# Analysıs of the Stabılıty of the Antı-Erosıon Cultıvator

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***Abstract*:** The stability of the movement of the soil cushion against erosion during basic tillage to the depth of treatment is theoretically analyzed. The methodological approach was evaluated in the works devoted to the stability of trailed and aggregate-suspended versions of tools with various working bodies protecting resources from erosion, and differential equations were compiled taking into account additional exciting forces affecting the movement of the selected option. for research at a given depth in the soil.

***Keywords:*** *Soil erosion, soil protection technology, tillage tool, stability of movement, aggregation scheme, equations of motion*..

**Introduction**

Mountainous and foothill areas make up 60 percent of the territory of the Republic of Azerbaijan. An area of up to 3 million 580 thousand hectares has been eroded to one degree or another [1]. It is this reason that causes a decrease in the yield of major crops in these areas. In a number of regions, the negative impact of the erosion process is manifested as a result of improper agrotechnical measures in cultivated fields, pastures and meadows. It should be noted that in the erosion-prone areas of the country, production is still based on the use of outdated technology and technical means. This has a negative impact on the intensification of agricultural production.

Taking into account the state of the soil, preventing its weakening, and maintaining high productivity regardless of the terrain made the justification of land, resource, and energy-saving technologies and technical means an urgent task of the day.

Currently, world experience has developed a number of technologies for protecting soils and structures of tools against erosion, including recommendations for soil cultivation without turnover and using buffers. However, it should be noted that there is a great need to study the static dynamics of large-width mesh tillage implements with a stronger and faster tractor.

The movement of a tractor with an anti-erosion tool can be considered based on the study of the equation of motion. Because when developing a universal anti-erosion tillage tool for basic tillage with deeply softening, cushioning, chiseled or semi-moving working bodies, it is necessary to know the stability of its movement in depth. The priority of studying the movement of a working machine under the action of specified forces belongs to V. P. Goryachkin [2]. He noted the importance of establishing the correct relationship between the forces acting on the machine, the mass and speed of the machine.

The method of constructing the equation of motion of a machine-tractor unit is reflected in the works of academician P. M. Vasilenko [3]. Here, the movement of an anti-erosion tool attached to a tractor was studied on the basis of the second-order Lagrange equation. The angle of rotation of the car was taken as a generalized coordinate. Here, only the movement of the tillage tool was considered without exciting influences.

The movement of mobile vehicles was studied by A.I.Timofeev [4]. Here are the general equations of solid state dynamics and methods for constructing equations of motion based on the Lagrange equation in generalized coordinates. However, in this study, only the flexible connection of the working body with the tractor by means of a hinge was considered.

In the work of A. B. Lurien [5], a method for deriving the equation of motion of an outboard machine aggregated with a wheeled tractor is presented. Here, taking into account the elasticity property of pneumatic tires, the unit is considered as a dynamic system with six degrees of freedom. The equation of motion is formulated using the quadratic Lagrange equation. A method for determining the kinematic and potential energy of the aggregate, as well as the dissipative function, is given.

The formulation of the equations of motion of a tractor with various tillage tools was considered in the work of A. I. Lyubimov [6]. The equations of motion were formulated using the Lagrange equation of the second degree. The linear displacement of the centers of mass of the tractor and tools, as well as their movement at an angle in the longitudinally vertical plane, has been replaced by linear movement of the corresponding supports in the vertical plane.

It is considered advisable to evaluate the work of the shock-absorbing bodies and select a scheme according to the methodology adopted for tillage implements, according to the depth of stability of the working bodies.

The stability of the movement of the working bodies in depth is influenced by many factors. More objective ones include the type and condition of the soil, the roughness of the field surface and the type of previous development.

Another group of factors includes the organization of the soil development process and the design features of the tool: the geometric parameters of the working bodies, the ways they are connected to the frame, their location, the parameters of the suspension scheme and the mode of operation. operation.

Let's consider the second group of factors affecting the stability of the movement of a tillage tool. Among them, the parameters of the working bodies and their location in the frame, as well as the parameters of the tool circuit, are of greater importance. If the parameters of the working body and its placement in the frame have optimal values, then the stability in depth depends on the parameters of the tool circuit, the width of its rigid section, its mass, the location of the center of gravity, the location of the working bodies and the support wheel in relation to the attachment point.

The physical and mechanical properties of the soil should also be taken into account. They change the magnitude and direction of the accumulated ground resistance force acting on the working bodies in a longitudinally vertical plane. In this case, the longitudinal vibrations of the tractor on uneven ground are transmitted to the tool by suspension parts.

The stability of the movement of a single-section outboard motor block against erosion in a longitudinally vertical plane, depending on the forces acting on it during operation, has been studied (Figure). To do this, we accept the following conditions:

- the tool, together with the suspension, is an integral system connected by rigid joints;
- the speed of the tractor and the gun is constant;
- the exciting forces acting on the system at the moment in question are perpendicular to the longitudinally vertical plane. Since the working bodies are symmetrical, their location in the frame is assumed to be equal in size and opposite to the direction of movement. At this time, the instrument performs a plane-parallel movement;
- due to the exciting moment in the longitudinally vertical plane, the instrument enters a new state of equilibrium;
- the rotation of the tool body occurs at small angular intervals.

We will consider the stability of the gun in the longitudinal direction when the central hook of the tractor has a parallel suspension connected to the elastic part. The movement of the tiller cushion can be characterized as follows:

• vertical displacement of the center of gravity of the tillage tool – Z;

• the angular position of the tiller frame is ψ.

The tool rests on a circle with $C_{tek}$ stiffness. At this time, the exciting effect of h (t) on the terrain is transmitted to the movement of the $Z_{tak}$ wheel and from there on the gun frame affects the value of Z. The price of angular displacement is affected by the stiffness of the spring in the central hook of the tractor, $C_{tr}$

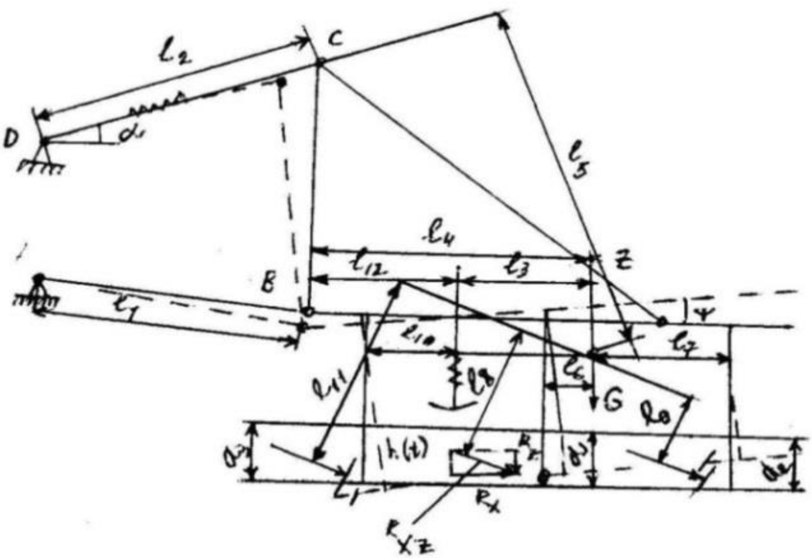

**Figure 1. Parameters and operating time of the tillerblock forces acting on it.**

Thus, it is necessary to know the generalized coordinates of the longitudinal –vertical plane.

The differential equation of the movement of the tool can be written using the LaGrange formula as follows:

$$\frac{d}{dt}\left(\frac{dT}{dk}\right)-\frac{dT}{dk}+\frac{dP}{dk}+\frac{dF}{dk}=Q_k, \tag{1}$$

where T – kinetic energy of the instrument, Nm;

P-potential energy of the instrument, Nm;

F-resistance function (dissipative function), Nm;

K-generalized coordinates, m;

$Q_k$-generalized force corresponding to the generalized coordinate, N.

The kinetic energy of the instrument is as follows.

$$T=\frac{m\cancel{Z}^2}{2}+\frac{J\psi^2}{2} \tag{2}$$

where m - the mass of the tool, kg;

J-moment of inertia of the instrument, kq m$^2$.

Potential energy is formed from the compression energy of the support wheels during tire deformation and the energy of the spring on the central hook. If we accept the linear stiffness of the tire and spring, we can write:

$$P = \frac{C_{tək}(h - Ƶ_{tək})^2}{2} + \frac{C_{tr}(\ell_s\psi - Ƶ_\alpha)^2}{2}, \tag{3}$$

where h - the height of the non-bathrobe of the relief under the support wheels of the instrument, m;

α-angle of rotation of the central hook, degrees;

ℓs – the shortest distance from the center of gravity of the tool to the central hook

($\ell_s$ – Ƶ α), (h - $Ƶ_k$) – prices for deformation of elastic elements, m; $Ƶ_{tək}$ = Z

$\ell_3\psi$;

$C_{tək}$, $C_T$ – stiffness of elastic elements, N/m$^2$.

$$F = \frac{d(Ƶ - \psi - \ell_6)^2}{2} + \frac{d(Ƶ - \psi - \ell_7)^2}{2} + \frac{d(Ƶ - \psi - \ell_{10})^2}{2}, \tag{4}$$

where $\ell_6$, $\ell_7$, $\ell_{10}$ – are the distances from the rows of working bodies to the center of gravity of the tool, m.

To calculate the generalized forces, it is necessary to know the working resistances of the instrument ($R_{XZ}$ and $R_Z$), as well as their change from the action of oscillations (Δ $R_{XZ}$ and $R_Z$).

$$\Delta R_{XƵ} = \nu\Delta\alpha; \quad \Delta R_Ƶ = \mu\Delta\alpha, \tag{5}$$

where Δα – change in soil processing depth, m;

ν, µ – coefficients that take into account the change in the resistance of the tool.

The change in the depth of soil processing during oscillations of the instrument frame is determined for each row of working bodies as follows:

$$\Delta\alpha_1 = -Ƶ + \ell_6\psi; \quad \Delta\alpha_2 = -Ƶ - \ell_7\psi; \quad \Delta\alpha_3 = -Ƶ + \ell_{10}\psi. \tag{6}$$

the generalized forces corresponding to their coordinates are as follows Z and ψ:

$$Q_Ƶ = \mu(-Ƶ + \ell_6\psi) + \mu(-Ƶ - \ell_7\psi) + (-Ƶ + \ell_{10}\psi); \tag{7}$$

$$Q_Ƶ = \nu\ell_8(-Ƶ + \ell_6\psi) + \nu\ell_9(-Ƶ - \ell_7\psi) + \nu\ell_{11}(-Ƶ + \ell_{10}\psi), \tag{8}$$

where $\ell_8$, $\ell_9$, $\ell_{11}$ – the shortest distances from the center of gravity of the tool to the line acting $R_{XZ}$ force.

Thus, all summation of Equation (1) becomes defined. (5), (6), (7), (8) we get their expressions by using them on the spot and doing transformations:

$$mƶ + 3dƶ + ƶ(C_{tək} + C_{tr}\alpha^2) + 3\muƶ + \psi[d(\ell_7 - \ell_6 + \ell_{10})] + \psi\mu$$
$$(\ell_7 - \ell_6 + \ell_{10}) - \psi(C_{tək}\ell_3 + C_{tr}\ell_5) = C_{tək}h(t); \tag{9}$$

$$\ell\psi + d\psi(\ell_7^2 + \ell_6^2 + \ell_{10}^2) + \psi(C_{tək}\ell_3^2 + C_{tr}\ell_5) + \psi\nu(\ell_6\ell_8 -$$
$$\ell_9\ell_7 + \ell_{11}\ell_{10}) + 3dƶ(\ell_7 - \ell_6 - \ell_{10}) - ƶ\,(C_{tək}\ell_3 + C_{10}\ell_5) -$$
$$-ƶ\nu(\ell_6\ell_8 + \ell_9\ell_7 + \ell_{11}\ell_{10}) = C_{tək}\ell_3 h(t) \tag{10}$$

The differential equations (9), (10) express the action of the soil-intensive tool against erosion in the main cultivation of the soil almost fully, taking into account its constructive parameters (mass, moment of inertia, dimensions), the location of the supporting wheels and working bodies, the alarming effects of the unevenness of the soil surface and tensile resistance.

Using these equations, it is possible to solve problems related to The Walking stability of the tool according to the depth of soil processing, taking into account the different working bodies, mass and location of the supporting wheels.

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