Modeling the process of wheel drive slipping with anti-skid devices

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Abstract. This article investigates the increase of traction-coupling properties of propellers, patency of machine-tractor units and decrease of soil compaction. For a propulsor equipped with anti-skid devices, the slippage process is formed due to factors of soil compression by the soil-tread and shear of the soil by tire hooks. With the decrease in the number of detachable hooks, the period when the first hook disengaged and the second has not yet entered into interaction with the ground, increases. At the moment, traction and coupling properties of the wheel are formed only due to the tire hooks. When the detachable hook engages with the soil, the traction capacity of the wheel is made up of the forces of shear of soil “bricks” sandwiched between tire hooks and the forces of soil deformation by detachable hooks. As a result of integrating the dependence of the shear stress and deformation of the soil, a formula was obtained to determine the tangential traction force of the tractor. If the skidding of the wheel assembly depends on the pulling force, then the drag force from the anti-skid device. Using the known dependencies of the slippage of wheeled propellers, dependences are obtained to determine the slippage of the propulsion unit with the anti-skid device. With the increase in the number of anti-skid devices on the wheel, slippage is reduced, however, according to the research, the time required for the assembly and disassembly of the device reduces the interchangeability of the wheel assembly. We calculated the number of anti-skid devices at which the maximum exchange capacity will be reached. The penetration depths at which the maximum efficiency of the wheel assembly running system will be reached are obtained and the results are tabulated.

1. Introduction
Characteristics of running systems of tractors significantly affect the technical level of the unit. The advantages of wheeled propellers are unequivocally established in comparison with caterpillar. They are small specific metal consumption and the possibility of application on highways. Wheeled tractors provide the best working conditions for the driver and maintenance personnel. Recently, with the development of farms and individual households, the importance, especially of low-power tractors, has become even greater. [1,2].

The basis for the theory of the rolling of the wheel and its contact with soil and soil physics was made by M.G. Becker [3], I.P. Ksenevich [4], M.M. Makhmutov [5,6,7,8], N.K. Mazitov [9] and others.

Their works state that the traction-coupling properties of propellers are improved due to:
1) use of the entire weight of the tractor as a coupling;
2) increase of tire contact with soil;
3) development of new design solutions;
4) development of new technological solutions.

These measures for a tractor with two driving wheels can be implemented by turning all four wheels into driving wheels, doubling the wheels, using loaders and hydraulic hooks, traction and semi-track. However, a significant disadvantage of the semi-caterpillar track is the short service life of the parts, the wear of the drive wheels and a slight increase in the drag force for movement.

2. Determination of the tangential pulling force

As a result of integrating the dependence of shear stress and soil deformation, V.V. Guskov [10] justified the following formula for determining the tangential traction force of a tractor:

$$P_c = \frac{f_{cs} K_c G_c}{\delta_m L_k} \left[ \ln ch \frac{\delta_m L_k}{K_c} - f_{sp} \left( \frac{1}{ch} \frac{\delta_m L_k}{K_c} - 1 \right) \right] + 2\tau_{xp} \frac{h_{sp} \cdot L_k}{t_m} ;$$

$$f_{sp} = 2.55 \left( \frac{f_{rs} - f_{cs}}{f_{cs}} \right)^{0.825}$$

where $f_{rs}$, $f_{cs}$ – respectively, coefficients of friction of rest, sliding and deformation; $f_{sp}$ – the reduced coefficient of friction; $\tau_{xp}$ – soil shear stress, N/m; $h_{sp}$, $t_m$ – correspondingly the height and pitch of the tire hook, m; $\delta_m$ – coefficient of wheel slippage, $K_c$ – coefficient of soil deformation.

V.V. Guskov’s formula takes into account the influence of wheel parameters and tire hooks on the traction and coupling properties of the propulsion unit. However, it does not take into account the effect of additional anti-skid devices on the traction characteristics of the running system.

When the propulsion unit is equipped with an anti-skid device due to the additional interaction of the hook with the soil, the traction power of the wheel assembly will increase by the following amount:

$$P_c = \left( \delta_k - \delta_m + \frac{S_{Tz} \cdot Z_c \cdot \delta_m}{2 \pi \cdot R_k} \right) 2\pi R_k \cdot K_c \cdot l_h \cdot h_p$$

$$Z_c = \frac{R_k + h_k - h_i - h_r}{R_k + h_r} (\delta_k - \delta_m)$$

where $\delta_k$, $\delta_m$ – wheel slip coefficients respectively with and without anti-sliding devices; $S_{Tz}$ – the theoretical way of moving the wheel, respectively, in the joint interaction of tire and removable hooks with soil, m; $Z_c$ – number of removable hooks on the drive wheel, pcs; $R_k$ – radius of the wheel, m; $K_c$ – coefficient of bulk crushing of soil, N/m^3; $l_h$ – width of removable hooks, m; $h_p$ – reduced height of removable hooks, m.

Accordingly, with an increase in traction force, the slippage will decrease and, depending on the parameters and number of hooks on the drive wheel, the coefficient of wheel slip will be determined $\delta_k$:

$$\delta_k = \frac{Z_c \cdot \arccos \frac{R_k + h_k - h_i - h_r}{360} \cdot (\delta_k - \delta_m)}{R_k + h_r}$$

where $\delta_k$ – wheel slip ratio at the moment of interaction of removable and tire hooks; $Z_c$ – number of removable hooks on the drive wheel, pcs; $R_k$ – radius of the wheel, m; $h_r$ – height of removable hooks, m; $h_k$, $h_c$ – tire deflection and track depth, m.

Using the known dependencies of skidding of wheeled propellers with additional anti-skid devices, we will apply them for the operation of the wheeled unit and obtain:

$$\delta_k = 1 - \left( \frac{S_{Tz} - \lambda_1}{S_{Tz} + \lambda_1} + \frac{S_{Tz} - \lambda_2}{S_{Tz} + \lambda_2} \right),$$

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where \( S_{r_1}, S_{r_2} \) – the theoretical way of moving the wheel, respectively, in the interaction of tire hooks and their joint interaction with additional devices with soil, m; \( \lambda_1, \lambda_2 \) – the values of the shear and compression of the soil in the horizontal direction, m.

The sum of the theoretical displacements of the wheel when slipping between the hooks and when interacting with the ground is equal to the step size of the removable hooks, \( t_c \):

\[
t_c = \frac{2\pi \cdot R_k}{Z_k} = S_{r_1} + S_{r_2},
\]

\[
\lambda_1 = k \cdot S_{r_1},
\]

\[
\lambda_2 = \frac{P}{K_c \cdot D_s \cdot h_p},
\]

\[
S_{r_1} = 0.035 R_k \cdot \arccos \left( \frac{R_k}{R_k + h_p} \right).
\]

where \( Z_k \) – the number of anti-slip devices on the wheel, pcs; \( h_p \) – the reduced height of the rack in the soil, m.

The resulted height of introduction of a rack into the soil will be determined:

\[
h_p = \frac{\pi \cdot (R_k + h_p)^2 \cdot \arccos \left( \frac{R_k - h_k - h_p}{R_k + h_p} \right)}{180} - R_k \sqrt{(R_k + h_p)^2 - (R_k - h_k - h_p)^2} \cdot h_p.
\]

Substituting the presented expressions in (9) we obtain the following expression for determining the slippage:

\[
\delta_s = \delta_{sw} - \frac{S_{r_2} \cdot Z_k \cdot \delta_{sw} + P \cdot Z_k}{\pi \cdot D_k \cdot K_c \cdot D_s \cdot h_p},
\]

where \( D_k \) – radius of the wheel, m; \( \delta_{sw} \) – coefficient of wheel slip from the tire; \( h_p \) – depth of penetration of the device rack into the soil, m; \( h_k \) – tire deflection and gauge depth, m.

The obtained expression takes into account the effect of the parameters of the anti-slip device and the soil properties on the amount of slippage of the wheeled propeller operating in the aisles.

3. Determination of the resistance to movement from the anti-slip device.

If the skidding of the wheel assembly depends on the traction force, then the drag force from the anti-slip device \( P_s \) is determined by the penetration force \( P_a \). Using the known dependences of the effect of the device parameters on the resistance to movement, we obtain the following formula:

\[
P_a \cdot (0.5D_k + h_k) \cdot \sin \left( \arccos \left( 1 + 8 \cdot \frac{B}{A} \right)^{1/2} \right)
\]

\[
P_s = \frac{P_a}{0.5D_k - h_k},
\]

where \( P_a \) – penetration force, kN; \( D_k \) – diameter of the tractor wheel, m; \( h_k \) – depth of penetration of the device rack into the soil, m; \( h_k \) – tire deflection and gauge depth, m.

The calculated coefficients A and B are determined by the following formulas:
\[ A = \left( -K_s \cdot D_s \right) \left( \frac{\pi D_s}{4} \cdot 0.5 D_s^2 + h_s \cdot \frac{\pi D_s}{4} \cdot 0.5 D_s \cdot h_s \cdot 0.5 D_s \cdot \frac{\pi D_s}{4} \cdot h_k \cdot h_s + \frac{\pi D_s}{4} \cdot h_k \right), \]

\[ B = \left( K_s \cdot D_s \right) \left( \frac{\pi D_s}{4} \cdot 0.5 D_s^2 + h_k \cdot \frac{\pi D_s}{4} \cdot 0.5 \cdot D_s \cdot \frac{\pi D_s}{4} \cdot h_k \cdot h_s + \frac{\pi D_s}{4} \cdot h_k, \right), \]

where \( K_s \cdot 10^6 \) – coefficient of volume crushing of soil N/m^3; \( D_s \) – diameter of the device's rack, m; \( D_k \) – wheel diameter, m; \( h_k \) – depth of track, m; \( h_s \) – deflection of the tire, m; \( h_k \) – depth of introduction of the device rack into the soil, m.

The obtained expressions allow us to determine the strength of the resistance to movement, depending on the parameters of the anti-skid device, the wheel, and the physical and mechanical properties of the soil.

The maximum efficiency of the running system of the wheeled vehicle, taking into account the conicity of the anti-skid device, is determined by the formula:

\[ \eta_x = \left( 1 - \frac{P_{sl}}{P_{fr} + P_{vl}} \right) \cdot (1 - \delta_x) \rightarrow \text{max}, \quad \text{(12)} \]

where \( P_{sl} \) – the forces of resistance to movement, respectively, from the wheels and the anti-sliding device, kN; \( P_{sp} \) – hook load, kN.

The strength of the anti-skid device in the soil depends on the diameter of the strut, its taper, the hardness of the soil, the depth of penetration. Substituting in the experimental model (12) the initial data for a hook load of 12.5 kN and a drag force of 3.1 kN, we obtain the following values of the efficiency of the running system as a function of the taper of the anti-slip device (Table 1).

| №  | \( C_x \) | \( P_{c}, \text{N} \) | \( P_{w}, \text{N} \) | \( P_{vy}, \text{N} \) | \( \delta_x, \% \) | \( \eta_{bc}, \% \) |
|----|--------|----------------|----------------|----------------|-------------|-------------|
| 1   | 0,5    | 1625,0         | 812,0          | 229,5          | 9,0         | 69,451      |
| 2   | 0,6    | 1706,7         | 852,6          | 241,0          | 8,9         | 69,454      |
| 3   | 0,7    | 1797,8         | 898,0          | 253,9          | 8,8         | 69,456      |
| 4   | 0,8    | 1898,5         | 948,2          | 268,0          | 8,7         | 69,457      |
| 5   | 0,9    | 2008,5         | 1003,2         | 283,6          | 8,6         | 69,458      |
| 6   | 1,0    | 2128,0         | 1063,0         | 300,5          | 8,5         | 69,458      |
| 7   | 1,1    | 2256,9         | 1127,6         | 318,8          | 8,4         | 69,457      |
| 8   | 1,2    | 2395,3         | 1197,6         | 338,4          | 8,3         | 69,454      |
| 9   | 1,3    | 2543,1         | 1271,6         | 359,4          | 8,2         | 69,451      |
| 10  | 1,4    | 2700,3         | 1350,0         | 381,7          | 8,1         | 69,445      |
| 11  | 1,5    | 2867,0         | 1434,0         | 405,4          | 8,0         | 69,438      |

Analysis of the plot (Figure 1) shows that with an increase in the coefficient of volume fracture of the soil in the range \( 6 \cdot 10^6 < K_s < 8 \cdot 10^6 \) N/m^3, the optimal value of the taper decreases by 0.4-0.5 (Figure 1).
Figure 1. Changing the efficiency of the running system depending on the taper of the anti-slip device: 1 – $K_\tau = 6 \cdot 10^6 \text{N/m}^3$; 2 – $K_\tau = 8 \cdot 10^6 \text{N/m}^3$; 3 – $K_\tau = 10 \cdot 10^6 \text{N/m}^3$.

With the increase in the penetration depth of the anti-skid device, the force of introduction and resistance to movement of the wheel assembly increases, but on the other hand, slippage decreases. The value of the penetration depth at which the maximum efficiency of the undercarriage system will be reached, the wheel assembly is given in Table 2.

Table 2. Coefficient of efficiency of the running system depending on the depth of penetration of the anti-skid device.

| № | $h_v$, m | $P_v$, N | $P_{c_v}$, n | $P_{c_s}$, n | $\delta_\kappa$, % | $\eta_{c_v}$, % |
|---|----------|-----------|--------------|--------------|----------------|---------------|
| 1 | 0,05 | 2496,0 | 1248,0 | 455,5 | 14,6 | 64,722 |
| 2 | 0,06 | 2504,6 | 1252,6 | 502,3 | 14,2 | 64,844 |
| 3 | 0,07 | 2523,1 | 1262,0 | 548,4 | 13,8 | 64,936 |
| 4 | 0,08 | 2551,4 | 1276,2 | 594,6 | 13,5 | 65,000 |
| 5 | 0,09 | 2589,8 | 1295,2 | 642,1 | 13,2 | 65,036 |
| 6 | 0,10 | 2638,0 | 1319,0 | 691,3 | 13,0 | 65,046 |
| 7 | 0,11 | 2696,2 | 1347,6 | 743,0 | 12,8 | 65,030 |
| 8 | 0,12 | 2764,2 | 1381,0 | 797,7 | 12,5 | 64,988 |
| 9 | 0,13 | 2842,2 | 1419,2 | 855,8 | 12,3 | 64,919 |
| 10 | 0,14 | 2930,2 | 1462,2 | 917,8 | 12,0 | 64,825 |
| 11 | 0,15 | 3028,0 | 1510,0 | 983,9 | 11,8 | 64,704 |

Analysis of the graph (Figure 2) showed that with an increase in the coefficient of the volume fracture of the soil in the range $8 \cdot 10^6 < K_\tau < 10 \cdot 10^6 \text{N/m}^3$, the optimal depth of penetration of the anti-skid device is reduced by 0.04 m.

Figure 2. Effect of the penetration depth of the anti-skid device on the efficiency of the running system: 1 – $K_\tau = 6 \cdot 10^6 \text{N/m}^3$; 2 – $K_\tau = 8 \cdot 10^6 \text{N/m}^3$; 3 – $K_\tau = 10 \cdot 10^6 \text{N/m}^3$. 
With the increase in the number of anti-skid devices on the wheel, slippage is reduced, however, the time it takes to install and disassemble the device reduces the performance of the wheel assembly. Consequently, there is such a value of the quantity at which the maximum replacement capacity will be reached:

$$W_{\text{cr}} = 0,36 \cdot (T_o - Z_c T_c) \cdot B_p \cdot V_t \cdot (1 - \delta_k) \rightarrow \max,$$

(13)

where $V_t$ – theoretical speed of the wheel assembly, m/s; $T_o$ – the main operating time of the wheel assembly, h; $B_p$ – working width of the combine harvester, m; $T_c$ – time required to mount the anti-slip device, h. For a wheeled unit at $T_o = 7$ h, $T_c = 0.11$ h, $B_p = 1.4$ m, $V_t = 2.88$ km/h, we obtain the optimum values of the shift capacity according to formula (13) and add it to Table 3.

Table 3. The effect of the amount of the anti-skid device on the drive wheel on the interchangeable performance of the wheel assembly.

| $Z_c$, pcs. | $K_v$=6·10$^6$ N/m$^3$ | $K_v$=8·10$^6$ N/m$^3$ | $K_v$=10·10$^6$ N/m$^3$ |
|------------|-------------------|-------------------|-------------------|
| $\delta_k$, % | $W_{\text{cr}}$, g/c | $\delta_k$, % | $W_{\text{cr}}$, g/c | $\delta_k$, % | $W_{\text{cr}}$, g/c |
| 0          | 24,8              | 2,12              | 21,2              | 2,23              | 19,3              | 2,28              |
| 1          | 24,5              | 2,16              | 20,9              | 2,24              | 19,1              | 2,29              |
| 2          | 23,6              | 2,19              | 20,2              | 2,25              | 18,4              | 2,30              |
| 3          | 22,1              | 2,21              | 18,9              | 2,26              | 17,2              | 2,29              |
| 4          | 20,1              | 2,23              | 17,2              | 2,27              | 15,6              | 2,28              |
| 5          | 17,4              | 2,24              | 14,9              | 2,26              | 13,6              | 2,27              |
| 6          | 14,2              | 2,25              | 12,2              | 2,25              | 11,1              | 2,26              |

Analysis of the optimization showed (Figure 3) that with an increase in the coefficient of the volume fracture of the soil in the range $6 \cdot 10^6 < K_v < 10 \cdot 10^6$ N/m$^3$, the optimum amount of the anti-skid device would be 2 pcs.

![Figure 3](image-url)

**Figure 3.** Change in the changing performance as a function of the amount of the anti-skid device on the drive wheel: 1 – $K_v$=6·10$^6$ N/m$^3$; 2 – $K_v$=8·10$^6$ N/m$^3$; 3 – $K_v$=10·10$^6$ N/m$^3$.

**Conclusion.**

With the increase in the penetration depth of the anti-skid device, the force of introduction and resistance to movement of the wheel assembly increases, but on the other hand, slippage decreases. The value of the penetration depth at which the maximum efficiency of the wheeled vehicle running gear will be reached is shown in Table 1.
Analysis of the graph showed that with an increase in the coefficient of volume fracture of the soil in the range $8 \cdot 10^6 < K < 10 \cdot 10^6$ N/m$^3$, the optimum depth of penetration of the anti-skid device is reduced by 0.04 m.

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