THE ANALYSIS OF OPERATIVE PARTS OF A MINI ROTARY CULTIVATOR.

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Abstract

This study seeks to establish and analyze the mathematical models of the plowing force and trajectory curve for tiller blades and cultivation blades of a rotary cultivator. Both blades are widely used in Word. To ensure good efficiency and uniform resistance of the plow to soil, the blade must be protected from intertwisting. The theoretical and calculated cutting angles are 85.88° and 85.10°, the edge-curve angles are 55.00° and 56.18°. The rear of well designed plowing blade does not cause any friction with the soil. The curve of plowing shows that when the relief angle of the tiller blades is 25°, the turning and the throwing have good performance. When the relief angle of the cultivation blade is 10°, the impact force and the crushing force of the blade into the soil will be minimized during plowing.

Introduction:

Mini cultivator (fig. 1) that work the soil generate some forces between the tiller blades and the soil, such as the compression of the soil, the resistance of the soil to its being cut, the adhesive forces on the soil and abrasive forces with the soil, among others. Several researchers have been interested in establishing the relationships among several design variables and the power requirement of rotary tillers.

Fig.1:- Mini cultivator
Ghosh model- A dimensional analysis of the torque requirement under different operating conditions was performed by developing a dimensional formula using the angular velocity of the rotor, the depth of working, the velocity of travel, the soil bulk density, the soil cohesion, the acceleration due to gravity and the soil particle size as independent variables. For a given soil, the moisture content and compaction conditions in the field, the forward speed of travel, are related in a dimensional equation:

$$T_r = K(n)^b(h)^c(\gamma)^d$$

Where $T_r$ - torque, in kg-cm
$K$-constant
$h$-depth of working, in cm
$\gamma$-forward speed of travel, in cm/s
$n$-rotary speed, in rpm
$b,c,d$-powers of unknown value

Gupta model- Based on the predicted behavior of saturated soil under impact loading and pure shear, a mathematical model to predict the power requirement of a rotary tiller has been developed. The torque required by a rotary tiller is based on the following operations. The impact cutting of the saturated soil by the rotary tiller initially starts with compression. When the cutting speed is high, compression is very quickly followed by impact shear. Force required to cut the soil in $NF_{CS} = \sigma_{is}A_1$ (3):

Where $\sigma_{is}$ represents the impact shear stress to cut the soil. $A_1$ is the total wetted area of the cutting surface of the tool, in mm$^2$, and $\sigma_{is}$ is given by the equation.

$$\sigma_{is} = \left[ \frac{u_s \eta_{is}}{2R \cos^{-1} \left[ \frac{R - h}{R} \right]} \right]^{1/\eta_{is}} + \sigma_{yis}$$

Where $\sigma_{yis}$ - yield stress impact shear
$\eta_{is}$-exponent=1.512
$\eta_{is}$-coefficient of viscosity $(2.8 \cdot 10^4, N - S/m^2)$
$u_s$-cutting velocity, in m/s
(2) Torque required to throw the cut soil slice The cut soil slice is thrown away by the centrifugal action of the rotating blades: Centrifugal force in $N F_e = \frac{m_s u_s^2}{R}$ (5)

With reference to figure 1:

$$m_s = \frac{hPwp_s}{g}$$

Where $m_s$-mass of the soil slice in kg, P- tilling pitch in mm and $\rho_s$-specific weight of the saturated soil in N/mm$^2$

(3) Torque required to overcome soil-metal friction Normal force $N$ that acts on the tool is equal to weight of cut soil slice given by

$$N = hPwp_s$$

Force $F_1$ required to overcome soil-metal friction is given by

$$F_1 = hPwp_s \mu_k$$
where $\mu_k$ - kinetic coefficient of soil-metal friction. (4) Torque required to overcome soil-soil sliding friction.

During cutting with the rotary tiller, the cut soil slice is separated from the uncut soil by pure shearing or soil-soil sliding. Equation 9 describes the behavior of the saturated soil under pure shear, according to the experiments conducted in a coaxial viscometer.

$$\gamma_0 = \frac{-dy}{dr} = \left[ \frac{1}{\mu_{ps} (\tau - \tau_y)} \right]^{1/n_{ps}}$$

(9)

The shearing rate $\gamma_0$ is a function of the speed of rotation of the blade and the depth of working:

$$\gamma_0 = \frac{V_p}{h}$$

Substituting the value $\gamma_0$ of into Eq. 9, yields,

$$\tau = \eta_{ps} \left( \frac{V_p}{h} \right)^{n_{ps}} + \tau_y$$

(10)

Where

- $\gamma_0$ - shear rate, in Pa
- $\tau$ - shearing stress under pure shear, in Pa
- $\tau_y$ - yield stress under pure shear, in Pa
- $V_p$ - peripheral speed of the rotor, in m/s
- $h$ - depth of working, in cm
- $n_{ps}$ - exponent=0.3
- $\eta_{ps}$ - coefficient of viscosity=49.8 N.s/m²

The rate of attrition of the tiller blade depends on the workload of the tractor. According to the analysis by Yamada, four major factors are correlated with the attrition of the tiller blades - (1) the conditions of the soil; (2) the quality of the tiller blades; (3) their mechanical conditions, and (4) the operating conditions. The farming conditions are related to the mechanics of the soil. A survey of the soil on farms, conducted by the Kogi Agriculture and Forestry Technology Institute (Japan) showed that a higher percentage of sand and silt causes faster attrition of the tiller blades.

Fig 3:- The shapes’ scoop-surface and outing function of tiller blades
The attrition of blades depends on the quality of the material from which they are made, their solidity, the emulsion with which they are sprayed, the heat treatment undergone, their depth, toughness and shape, the cutting angle, the edge of the blades, their curvatures, and the angle of scoop. The attrition of a blade is correlated with mechanical output forces, the arrangement of the tiller blade, the repeated crushing of the soil, and other factors. The operating conditions also depend on the tilling area, the compression of the soil, the impact loads, the resistance due to abrasion and sharpening.

The geometry of tiller blades is considered to be the most important factor in their design because both the shape of the blade tip and the lengthwise of the tiller blades facilitate cutting. The width of the blade tip exceeds the lengthwise of the blade. The contact between the blade and the soil moves slowly from the handle near the center of the shaft to the length of the blade. The tip of the blade at the boundary between the lengthwise of the blade and the blade tip cuts the intact grass. The grass can also be thrown away or torn off by the outward rotation. This type of blades performs well in the soil in Europe and is extensively used in Georgia. Figure 2 shows both the blade tip and the lengthwise of the blade, as segments ED and Dn, respectively in Fig. 3. The lengthwise of the blade must meet two conditions. One is the absence of intertwisting and the other is a low drag force during cultivation. The cutting conditions are $T \geq F$ (Fig. 2). $T = Ncott; F = Ntang; cott > tang; \tan(90^\circ - \tau) > tgg; 90^\circ - \tau > 90^\circ - \varphi; \tau < 90^\circ - \varphi$.

Where $\varphi$ is the friction angle of the rootstock with respect to the blade edge. The lengthwise of the blade is part of an Archimedean curve whose parametric equation is,

$$R = R_0(1 + K\theta)$$

where

$R_0$ - initial blade helical line radius, in cm

$R$ - rotation radius of the blade, in cm

$R_{max}$ - maximum tilling radius, 24-26 cm

$S_x$ - thickness of soil in transverse section, in cm

$h_{max}$ - maximum depth of working, in cm

$\theta_{max}$ - maximum central angle, 35-45°

$T$ - cutting force, in N

$F$ - friction force, in N

Take $i$ as the angle between $AA'$ and line $Om$;

$$i = \tan^{-1} \frac{1 + K\theta}{K}$$

The angle $i$ increases with the angle $\theta$, but the angle $\tau$ decrease, as determined by graphical analysis. The low $\tau$ positively affects the cutting work. Equations (14), (15), and (16) present another closed solution (Sakai et al., 1976; Sakai, 1978) which can be obtained using polar coordinates on the spiral line (Figure 4). These equations have the same meaning as Eq. (13).

$$r = r_0\sin^\frac{1}{\alpha}(\alpha_0 + K\theta)$$

$$\alpha' = \alpha_t + \left[\frac{\tan^{-1} \frac{n\pi\sqrt{h} + 2R - h}{30\gamma - n\pi(R - h)}}{30\gamma - n\pi(R - h)} + 90\right]$$

$$r = r_0e^{-cota'/\theta}$$

Where

$r_0$ - maximum radius at the tip point of the edge curve, in cm

$\alpha_0$ - edge-curve angle at the tip point where $r$ is maximum

$r$ - calculated radius of the spiral, in cm

$\gamma$ - forward speed of travel, in cm/sec

$\alpha'$ - 4° or greater than $\alpha_t$

First, use Eq. (17) to calculate angle $\beta$ (Fig. 3). Angle $\beta_1$ can be easily found by using Eq. (18).

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Fig. 3: Planing design

Then, use Eq. (19) to calculate the forward speed of travel. The acceleration can be determined by Eq. (20) according to table 1.

**Table 1:** Operational model for different types of tiller blades

| Blade type                  | v(cm/sec) | n (rpm) | γ (°) | a (cm/sec^2) |
|-----------------------------|-----------|---------|-------|--------------|
| Cultivation blade           | 20-30     | 300-400 | 5-10  | 0            |
| Cultivation broken bit blade| 30-40     | 250-300 | 10-15 | 2.3          |
| Cultivation wasteland blade | 40-60     | 150-250 | 15-20 | 0            |
| Tiller blade                | 40-60     | 150-300 | 20-30 | 2.3          |
| Cutting blade               | 20-30     | 250-500 | 20-30 | 2.3          |

Finally, Eq. (21) was used to determine time t for cultivation and the parameters such as A^2, B, r, and D can be determined from Eqs. (22), (23), (24), (25), and (26).

\[
\beta = \cos^{-1}\left( \frac{30V}{R} \sqrt{\frac{h(2R-h)}{(30V)^2 - 60\pi V(R-h) + (R\pi n)^2}} \right)
\]

(17)

\[
\beta_1 = \beta \cdot V
\]

(18)

\[
\gamma = \frac{n\pi R}{30V} \left( n\pi c_1 - 30V \right) \cdot \tan \left( 180^\circ - \gamma - \tan^{-1}\left( \frac{n\pi \sqrt{h(2R-h)}}{30V - n\pi c_1} + n\pi \sqrt{(2R-h)} \right) \right) \frac{30\sqrt{(30V - n\pi c_1)^2 + n^2\pi^2 h(2R-h) \cdot tan \left( 180^\circ - \gamma - tan^{-1}\left( \frac{n\pi \sqrt{h(2R-h)}}{30V - n\pi c_1} \right) \right)}}{2n\pi R}
\]

(19)

\[
a = \gamma \frac{cm}{sec^2}
\]

(20)

\[
t = \frac{- (V + \gamma) + \sqrt{(V + \gamma)^2 + 2a\sqrt{R^2 - C^2}}}{a}
\]

(21)

\[
A^2 = \left\{ \sqrt{R^2 - C^2} - (V + \gamma)t - \frac{a}{2} t^2 \right\}
\]

(22)

\[
B = (V + \gamma)t
\]

(23)

\[
r = \sqrt{C^2 + A^2}
\]

(24)

\[
\theta = \sin^{-1}\left[ \frac{C}{r} \right] - \frac{n\pi t}{30}
\]

(25)

\[
D = \frac{a}{2} t^2
\]

(26)
Conclusion:-
This study establishes and analyzes mathematical models of the plowing force and orientation curve of tiller blades and cultivation blades. To ensure good efficiency and uniform resistance of the plow to soil, the blade must be protected from intertwisting. The theoretical and calculated cutting angles are 85.88° and 85.10° and, the edgecurve angles are 55.00° and 56.18°, with the data in the literature. The rear of a well designed plowing blade does not cause any friction with the soil. The curve of plowing shows that when the relief angle of the tiller blades is 25°, the turning and the throwing have good performance. When the relief angle of the plowing blade is 10°, the impact force and the crashing force of the blade into the soil will be minimized during plowing.

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