OPTIMIZATION TECHNIQUE OF DRIVING DISC PARAMETERS
/
МЕТОДИКА ОПТИМИЗАЦИИ ПАРАМЕТРОВ ДИСКА-ДВИЖИТЕЛЯ

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ABSTRACT
Flat circular discs in powered operation mode create driving forces. These forces enable to decrease the wheel slippage of the energy saturated tractor of tillage unit and to reduce the specific energy consumption. The objective of this study was to develop a technique that enables to determine the driving disc optimal parameters for maximum efficiency criterion. The earlier developed mathematical model of soil-disc interaction was used for this purpose. Soil properties in the model are characterized by means of two empirical constants. The relative depth and the kinematic parameter determine the disc operation mode. It was shown that the driving disc can operate with high efficiency, if the disc operates at the optimal values of the parameters. The driving disc efficiency can achieve the value about fifty percent. The experimental results confirmed the adequacy of the technique. The discrepancy between the predicted and field experimental values of driving forces and applied moments was about 25%. The proposed technique can be modified to optimize the parameters of other powered rotary tools of tillage machines and units.

РЕЗЮМЕ
Плоские круглые диски в приводном режиме создают движущие силы. Эти силы позволяют уменьшить буксование колес энергонасыщенного трактора почвообрабатывающего агрегата и снизить удельные энергозатраты. Целью данного исследования явилась разработка методики определения оптимальных параметров диска-движителя по критерию максимального коэффициента полезного действия. Для этого использовалась математическая модель взаимодействия диска с почвой, построенная ранее. Свойства почвы в ней характеризовались двумя эмпирическими постоянными, а режим работы диска определялся его относительным заглублением и кинематическим параметром. Было показано, что КПД диска-движителя составляет около 50% при оптимальных режимах работы. Результаты экспериментов подтвердили адекватность предложенной методики. Расхождение между предсказанными и экспериментальными значениями движущих сил и приложенных моментов составило около 25%. Эта методика может быть модифицирована для оптимизации параметров приводных ротационных рабочих органов почвообрабатывающих машин и агрегатов.

INTRODUCTION
The analysis of proposed designs of primary tillage tools showed that, in terms of the quality indicators, the ordinary share plough is still the best. At the same time, ploughing is the most energy-intensive process in the plant growing. Its implementation takes about 30-40% of all energy consumption for field works. In this regard, conservation of energy resources in ploughing acquires the special importance, in particular, by reducing the power losses of the tractor drive wheels due to slippage (*Misuno, 2018*).

About 20–55% of all tractor power is lost due to the slippage of the tractor driving wheels and to the tire and soil deformation (*Eto et al., 2018; Janulevicius et al., 2019*). Wheeled tractors work with the greatest efficiency if the slippage of the driving wheel is in the optimal range, namely, in the range of 8–12% (*Zoz, 1972; Battiato et al., 2017*). In order for the slippage of the tractor drive wheel to correspond to this range it is necessary to reduce the traction resistance of tillage implements.

There are many ways to reduce the traction resistance of tilling implements, which can be divided into three main groups (*Akimov & Medvedev, 2004*). The first group is the improvement of passive tillage tools (coating the mouldboard surface with polymers, plastics, watering of the plough mouldboard, vibration of plough bottom, as well as using a plough bottom with variable geometry). The second group is the setting of additional active tillage tools (a vertical rotor instead of mouldboard wing, a rotary tiller section instead of skimmer, flat discs in front of each plough bottom, etc.). The third group is characterized by the creation of new type implements, which do not have share-mouldboard tillage bottoms. This group includes rotary ploughs of various designs, screw ploughs, blade ploughs, etc.

Energy saturation growth predetermines an increase in operating speed of tractors and an intensive increase in plough resistance. The increasing of the energy capabilities of the tractors must correspond to the technological plough capabilities at higher speeds. Therefore it is necessary to find ways to significantly reduce the soil resistance to plough movement. Ploughs in combination with driving discs (DDs) well satisfied this requirement (*Akimov & Medvedev, 2004*). A plough in combination with DDs (flat discs set in the plane of the field cutoffs of the plough bottoms and operating in the mode of movers) can operate at the forward speeds 2–3 times higher than the fastest rotary plough. At the same time it has the significant advantages in productivity and specific energy consumption with the good quality work compared to conventional share ploughs.

Ploughs with DDs enable to use efficiently the energy saturated wheeled tractors for ploughing. In this case the slippage of tractor drive wheels decreases significantly, productivity increases, fuel consumption per hectare decreases, the possibility of full engine loading and further energy saturation of tractors appears. The active discs are used in forced rotation mode not only in combination with plough, but in other various tillage units to create an additional driving force.

A flat disc coulter is the simplest rotary tool, which is widely used in various tillage implements. Disc coulters are used in different modes in ploughs, seeders and planters to cut crop residues (*Morrison & Allen, 1987*). In addition, the active discs are used in a forced rotation mode in various tillage units to create an additional driving force. This enables to reduce the specific energy consumption for soil cultivation by decreasing the wheel slippage at the soil-tyre interface of energy saturated tractor (*Medvedev, 1972*). The use of driven discs also makes it possible to increase the directional stability of the tillage units (*Musin et al., 2011*). With the help of such discs, it is possible to significantly reduce the traction resistance of the row cultivator and to increase its directional stability (*Serguntsov, 2017*). In the forced rotation mode, the disc coulters of seeders better cut crop residues (*Sarauskis et al., 2013*).

Adequate mathematical model of the disc coulter-soil interaction makes it possible to create low-power-consuming tillage units by modeling technological processes with carrying out numerous calculations on a computer. For applications of this model, it is necessary to choose reasonably the criteria of optimality. The mathematical model of the interaction of a disc coulter with a soil was being developed firstly only for a freely rotating (passive) disc (*Nerli 1929–1930a, 1929–1930b; Sineokov, 1949; Skakun & Flaischer, 1981*). And then it was developed for the powered (active) disc (*Akimov & Konstantinov, 2017*).

The objective of this article is to develop a technique that enables to determine the driving disc (DD) optimal parameters for maximum efficiency criterion by using the previously developed mathematical model of soil-disc interaction. This technique enables the combination of ploughs with DDs to use the energy resources more efficiently.

MATERIALS AND METHODS

We suppose that the flat DD of radius r [m] moves with a constant forward velocity $\boldsymbol{v_f}$ [m/s] and simultaneously rotates with an angular velocity ω [rad/s], cutting the soil to a depth h [m]. The disc operation mode is determined by its kinematic parameter $\lambda = \omega r/v_f$ and the relative depth $\xi = h/r$. The experiments show (*Akimov & Konstantinov, 2017*) that the soil pressure p on the lateral surfaces of disc in the soil and the cutting resistance Q of a unit length of its blade are practically constant and do not depend on λ and ξ. The empirical constants p and Q can be determined from simple experiments.

Let us direct the Ox axis of the coordinate system moving together with the disc in the direction of the movement of the DD, and direct the Oz axis vertically downward (Fig. 1). An elementary soil friction force $\boldsymbol{dF_f}$ acts on an arbitrary elementary area $dS = dxdy$ of the lateral surface segment of the disc in the soil. This force is directed opposite to the absolute velocity vector of a point $M(x; z)$ of this area. Therefore, its projection onto the Ox axis is equal to:

$$dF_{fx} = dF_f \cdot \cos\varphi = fpdxdy \cdot \cos\varphi \tag{1}$$

where:

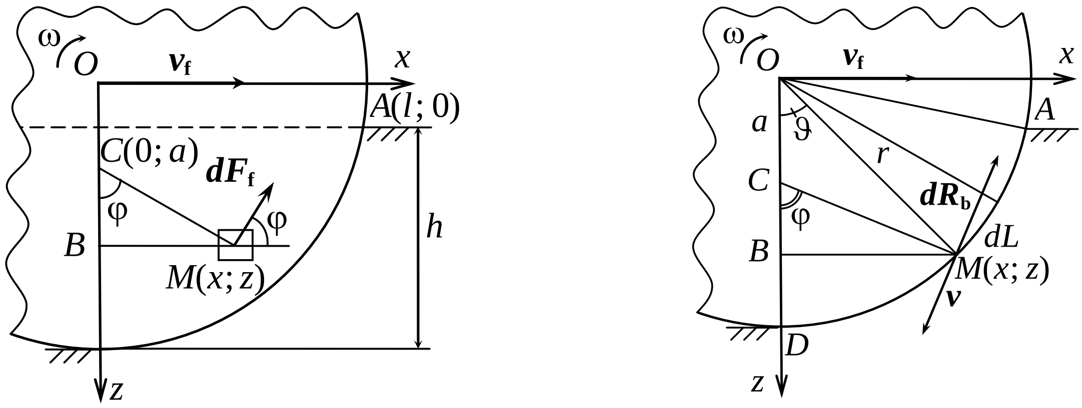
f is the soil-steel surface friction coefficient;

φ is the angle between CM segment and Oz axis (Fig. 1(a)).

If the point *M* is located below the point *C*, then the projection of the force ***dF****f* onto the *Ox* axis is positive, that is, the component of the force ***dF****f* relative to the *Ox* axis is the driving force.

The value of the total driving force created by the frictional forces of the soil on the lateral surfaces of the disc is equal to the doubled double integral of the elementary projection of the force ***dF****f* onto the *Ox* axis over the segment *S* immersed in the soil of one of the two lateral disc surfaces

$$F_{sx} = 2fp\iint\limits_S \cos\varphi \,\mathrm{d}x\mathrm{d}z. \tag{2}$$



(*a*) (*b*)

Fig. 1 - The scheme of interaction with the soil of an elementary part of
(a) the disc side surface and (b) the disc blade

From a right triangle MBC we have:

$$\cos\varphi = (z - a)/((z - a)^2 + x^2)^{0.5},\ \sin\varphi = x/((z - a)^2 + x^2)^{0.5}. \tag{3}$$

Substituting the first of these expressions into the integral (1) and representing the double integral as an iterated integral, we obtain the formula:

$$F_{sx} = 4fp\int_0^{l\cdot r} dx \int_{r-h}^{\sqrt{r^2-x^2}} \frac{(z-a)}{\sqrt{(z-a)^2+x^2}}\mathrm{d}z \tag{4}$$

where: $l = (2\xi - \xi)^{0.5}$.

Calculating the inner definite integral in the iterated one and changing the integration variable to $t = x/r$ in the outer integral, we obtain the final expression for the magnitude of the total driving force created by the friction forces on the lateral surfaces of the DD:

$$F_{sx} = 4fpr^2\int_0^l \left[\left(\mu^2 - 2\mu\sqrt{1-t^2} + 1\right)^{0.5} - \left((1-\xi-\mu)^2 + t^2\right)^{0.5}\right]dt \tag{5}$$

We consider as the positive direction of the moments of forces the direction coinciding with the direction of rotation of the DD.

It follows from Figure 1(b) that the moment of the elementary frictional force ***dF*****f** about point C (instantaneous centre of velocities) is equal to:

$$dm_C = -\,dF_f \cdot CM = -fpdxdy((z - a)^2 + x^2)^{0.5} \tag{6}$$

The value of the total moment about point C, created by the frictional forces of the soil on the lateral surfaces of the disc, is equal to the doubled double integral of the moment of the elementary force ***dF*****f** with respect to this point along the immersed in the soil segment *S* of one of the two lateral surfaces

$$m_C = -2fp\iint\limits_S \sqrt{x^2 + (z-a)^2}\,\mathrm{d}x\mathrm{d}z \tag{7}$$

Representing the double integral as an iterated one, we obtain:

$$m_C = -4fp \int_{r-h}^{r} dz \int_{0}^{\sqrt{r^2-z^2}} \sqrt{x^2+(z-a)^2}\,dx \tag{8}$$

Calculating the inner definite integral in the iterated one and changing the integration variable to $t = x/r$ in the outer integral, we obtain the final expression for the total moment about point C of elementary friction forces of soil on the lateral surfaces of the disc:

$$m_C = -2fpr^3 \int_{1-\xi}^{1} \left(\sqrt{1-t^2}\sqrt{\mu^2-2\mu t+1} + (\mu-t)^2 \ln \frac{\sqrt{1-t^2}+\sqrt{\mu^2-2\mu t+1}}{|\mu-t|} \right) dt \tag{9}$$

If we shall simplify the system of elementary friction forces to the point C, we shall get the system principal vector $\boldsymbol{F_s}$ acting at this point and the couple with the principal moment m_C. In order to simplify the simplified system to the new point O (the disc centre), it is necessary to move the principal vector of the system from the point C to the point O, and add a couple with a moment equal to the moment of the moving principal vector with respect to the point O (theorem of parallel translation of a force).

So, the moment about the point O of the system of elementary soil friction forces on the sidewalls of the disc is:

$$m_O = m_C - F_{sx} \cdot a \tag{10}$$

Let us consider an arbitrary elementary segment of the blade $dL = r \cdot d\vartheta$ with a point $M'(r\sin\vartheta;\ r\cos\vartheta)$ on it (Fig. 1 (b)). The elementary reaction of the soil resistance to penetration (wedging) by this elementary segment of the disc blade $\boldsymbol{dR_b}$ is directed against the vector of the absolute velocity of the point M', that is, perpendicular to the segment CM′ (Fig. 1 (b)).

Therefore, its projection onto the Ox axis is equal to:

$$dR_{bx} = Qrd\vartheta\cos\varphi \tag{11}$$

If the point M' is located below the point C, then the projection of the force $\boldsymbol{dR_b}$ onto the Ox axis is also positive, that is, the component of the force $\boldsymbol{dR_b}$ relative to the Ox axis is the driving force too.

From the equality (3) we find that:

$$\begin{aligned} \cos\varphi &= \frac{r\cos\vartheta - a}{\sqrt{(r\cos\vartheta - a)^2 + r^2\sin^2\vartheta}} = \frac{\lambda\cos\vartheta - 1}{\sqrt{(1+\lambda^2 - 2\lambda\cos\vartheta}}, \\ \sin\varphi &= \frac{r\sin\vartheta}{\sqrt{(r\cos\vartheta - a)^2 + r^2\sin^2\vartheta}} = \frac{\lambda\sin\vartheta}{\sqrt{(1+\lambda^2 - 2\lambda\cos\vartheta}}. \end{aligned} \tag{12}$$

By substituting the first of the two last equalities in the equality (11) and integrating the result we find the value of the total driving force created by the reaction forces of the soil on the blade of the disc which is equal to the curvilinear integral of the elementary projection of the force $\boldsymbol{dR_b}$ onto the Ox axis over the immersed in the soil segment L of the blade:

$$R_{bx} = Qr \int_{0}^{\vartheta_0} \frac{(\lambda\cos\vartheta - 1)d\vartheta}{\sqrt{1+\lambda^2 - 2\lambda\cos\vartheta}}, \tag{13}$$

where $\vartheta_0 = \arccos(1-\xi)$.

It follows from Figure 1 (b) that the moment of the elementary soil reaction $\boldsymbol{dR_b}$ about point O (the disc centre) is equal to:

$$dM_O = -\,Qrd\vartheta r\cos(\varphi - \vartheta) = -Qr^2(\cos\varphi\cos\vartheta - \sin\varphi\sin\vartheta) \tag{14}$$

Substituting the equalities (12) into the last equality and finding the curvilinear integral of dM_O over the immersed in the soil segment L of the blade we obtain the principal moment M_O of the soil reactions to the DD blade about the disc centre:

$$M_O = Qr^2 \int_{0}^{\vartheta_0} \frac{(\cos\vartheta - \lambda)d\vartheta}{\sqrt{1+\lambda^2 - 2\lambda\cos\vartheta}}. \tag{15}$$

In order to reduce the number of independent parameters, we introduce the dimensionless force characteristics of the DD:

$$F_{sx}^{*} = \frac{F_{sx}}{4fpr^{2}},\ R_{bx}^{*} = \frac{R_{bx}}{Qr},\ m_{O}^{*} = \frac{m_{O}}{4fpr^{3}},\ M_{O}^{*} = \frac{M_{O}}{Qr^{2}}. \tag{16}$$

To find the total driving force of DD we add the driving force of the disc surfaces and the driving force of the blade

$$R_{x} = QrR_{bx}^{*} + 4fpr^{2}F_{sx}^{*} = Qr(R_{bx}^{*} + nF_{sx}^{*}), \tag{17}$$

where $n = 4fpr/Q$ is the dimensionless coefficient ($n \geq 0$).

In a steady motion mode the moment applied to the powered disc from the PTO is equilibrated with the principal moment M_p of all soil reactions, which is equal to:

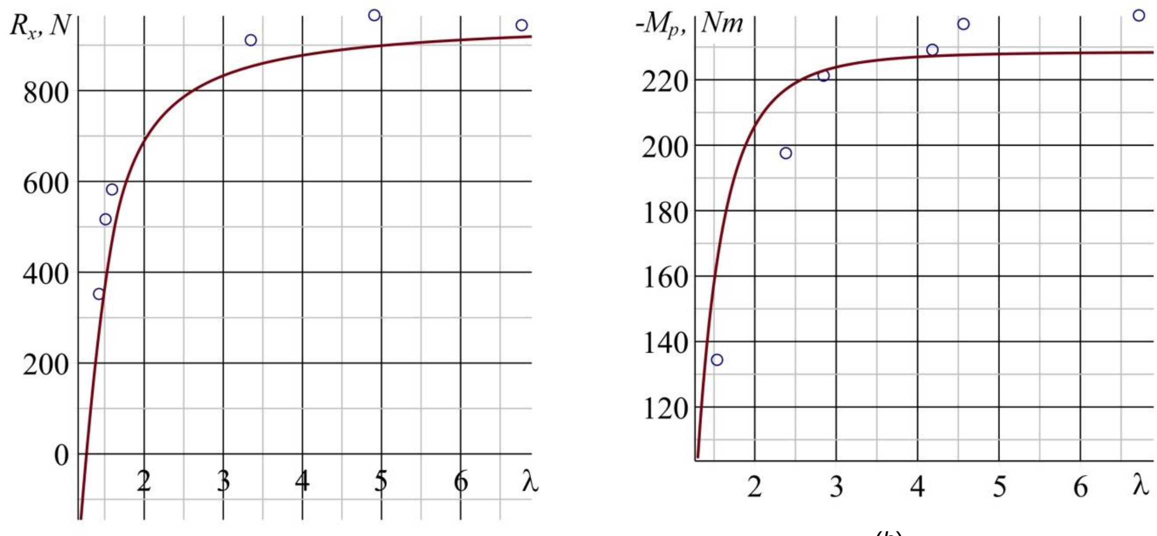
$$M_{p} = Qr^{2}M_{O}^{*} + 4fpr^{3}m_{O}^{*} = F_{sx}^{*} = Qr^{2}(M_{O}^{*} + nm_{O}^{*}). \tag{18}$$

The field unit was created for experimental investigations of the DD operation mode influence on energy characteristics of tillage unit. This unit allowed to vary the λ parameter in a wide range.

During the test, the torsion torque on the shaft of driving discs, the horizontal and vertical soil reaction forces on the DDs, and the rotation frequency of DDs were synchronously measured and recorded. The torsion torque transmitted to the shaft with discs was measured with the help of rotational dynamograph, the forces acting on the discs were measured with the help of strain gauge dynamometers, and the rotation frequency was measured with the help of the inductive sensor. The oscillograph was used for recording the measured values.

The tests were performed on the wheat field after harvesting. At the soil depth ranges of 0.10–0.20 m the average soil cone index was 2.95 MPa. At the soil depth ranges of 0–0.10 m the soil moisture content was 13.6%, and at the soil depth ranges of 0.10–0.20 m the soil moisture content was 16.1%.

A series of tests were conducted with discs of radius r = 0.25 m at constant depth h = 0.1 m, and their operation mode was varied at the expense of peripheral rotational velocity. As a result of the treatment of oscillograms the experimental values of the driving forces of the driving discs and moments applied to discs were determined. The experimental points with theoretical curves for f = 0.5, Q = 440 N/m and p = 34.4 KPa drawn by formulas (11), (12) are shown in Figure 2.



(*a*) (*b*)

Fig. 2 - Comparison of the theoretical and the experimental driving disc force characteristics

(a) for the driving force and (b) for the modulus of principal moment of soil reactions

As one can see from Figure 2, the maximum relative error for the driving force is equal to 24.6%, and for the modulus of principal moment of soil reactions it is equal to 25.4%. Such errors are acceptable for field measurements, so the mathematical model adequately describes the driving disc-soil interaction.

In accordance to the classical definition, the efficiency of a tool is the ratio of the useful tool power (work or energy) to the total one supplied to it.

The useful power developed by the DD is $P_1 = R_x v_f = (R_{bx}^* + nF_{sx}^*)Qrv_f$, and the total power supplied to the disc is $P_2 = -M_O\omega = -(M_O^* + nm_O^*)Qr^2\omega$.

By the definition, the efficiency of the drive disc is $\eta = P_1/P_2$, that is:

$$\eta = -\frac{R_{bx}^* + nF_{sx}^*}{(M_O^* + nm_O^*)\lambda} > 0. \tag{19}$$

We notice that, in active operation mode $R_{bx}^* + nF_{sx}^* > 0$ and $(M_O^* + nm_O^*)\lambda < 0$, that is the soil reaction forces push the DD and slow down its rotation.

In the formulas (5)–(10) and (13)–(15), the definite integrals parametrically depend on the quantities λ and ξ. Therefore, these formulas and formulas (16), (19) make it possible to analyse the dependence of the coefficient η on the parameters. These integrals are expressed in terms of Legendre elliptic integrals in normal form (*Akimov & Konstantinov, 2017*), but they are not expressed in terms of elementary functions. Therefore, the integrals were calculated numerically in the Maple package.

RESULTS

We also introduce the two dimensionless power characteristics of the active disc:

$$P_1^* = \frac{P_1}{Qrv_f} = R_{bx}^* + nF_{sx}^*,\; P_2^* = \frac{P_2}{Qrv_f} = -(M_O^* + nm_O^*)\cdot\lambda, \tag{20}$$

where P_1^* is the dimensionless useful power developed by the DD, and P_2^* is the dimensionless total power supplied to the disc.

The graphs of the resultant side friction force projection onto the *Ox* axis versus λ parameter are shown in Figure 3(a), and such graphs of the projection of the resultant soil reaction to the blade – in Figure 3(b).

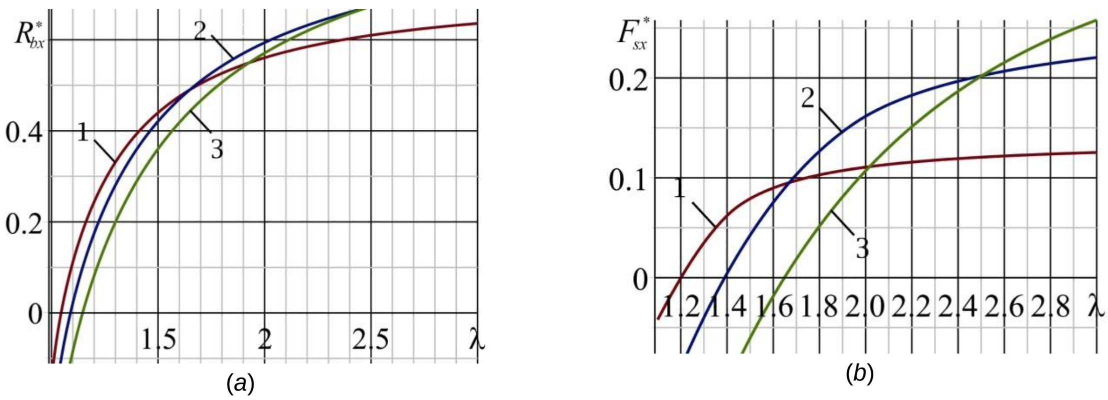
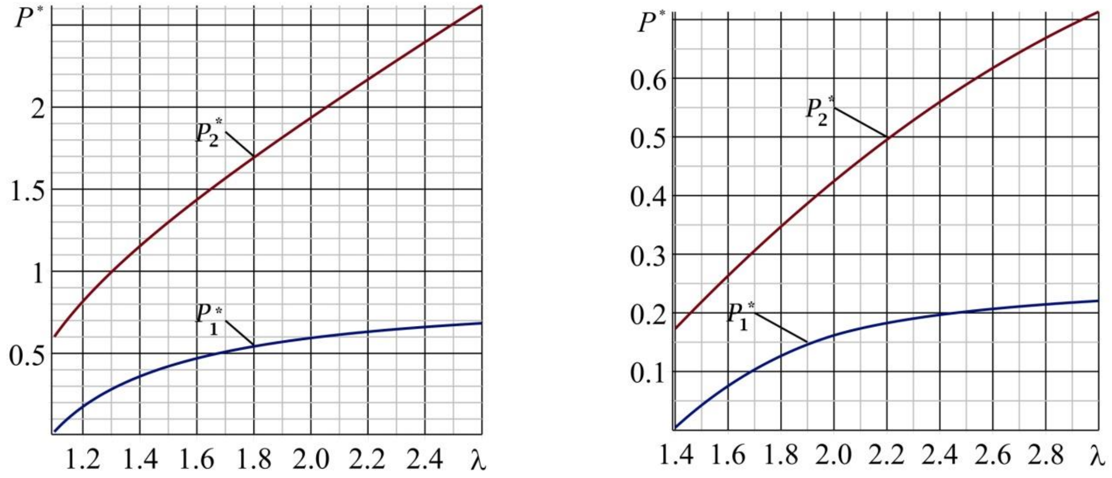


Fig. 3 - The graphs of the dimensionless projections of the soil resultant reactions versus λ
(*a*) R_{bx}^* (*on the disc blade*) *and* (*b*) F_{sx}^* (*on the disc sides*)

The graphs in Figure 3 were plotted for the following three ξ values: 0.3 (curve 1), 0.5 (curve 2), and 0.7 (curve 3).

For any given ξ there exists a value $\lambda_1^* = \lambda_1^*(\xi)$ such that at $\lambda = \lambda_1^*$, $R_{bx}^* = 0$, and there exists a value $\lambda_2^* = \lambda_2^*(\xi)$ such that at $\lambda = \lambda_2^*$, $F_{sx}^* = 0$ (Fig.3). Both projections are positive for values $\lambda > max(\lambda_1^*, \lambda_2^*)$, hence the sum $R_{bx}^* + nF_{sx}^*$ is positive too, and the powered disc can perform the moving function in the corresponding operation modes. The projections decrease with increase in ξ for small λ parameter values and increase for the larger values of this parameter. The increase rates of the projections decrease with λ increase, therefore operation modes with small λ parameter values are more effective for the moving function.

Theoretically, the dimensionless coefficient n is in the range from 0 to ∞. For n = 0, there are no friction forces on the lateral surface of the disc and one can easily obtain from the formulas (20) that $P_1^* = R_{bx}^*$, $P_2^* = -M_O^*\lambda$. The graphs of these power characteristics versus λ parameter are shown in Figure 5(a). For $\lambda \approx 1.1$ the useful power $P_1^* \approx 0$, and it increases with the increase in λ. For λ values greater than 1.5 with the λ increase the supplied power P_2^* increases almost linearly, but the power P_1^* tends to some limit (Fig. 4(a)).



(*a*) (*b*)

Fig. 4 - The graphs of the dimensionless powers P_1^* and P_2^* versus λ

(a) for n = 0 (the disc blade) and (b) for n = ∞ (the disc sides)

In the case n = 0 the expression for the efficiency of the DD is also simplified:

$$\eta_0 = -\frac{R_{bx}^*}{M_O^*\lambda}. \tag{21}$$

This formula can be used to calculate the efficiency of the DD when the disc operates in light soils, for which friction forces on the lateral surfaces of the disc can be neglected in comparison with the cutting resistance forces on the disc blade (n << 1).

Value $n = \infty$ corresponds to the case when there are no soil resistance to cutting, so Q = 0, $Q{\cdot}n = 4fpr$, and $P_1 = 4fpr^2 v_f F_{sx}^*$, $P_2 = 4fpr^3 m_O^*\omega$. In this case, the dimensionless powers are equal to $P_1^* = P_1/(4fpr^2v_f) = F_{sx}^*$, $P_2^* = P_2/(4fpr^2v_f) = -m_O^*\lambda$. The graphs of these power characteristics versus λ parameter are shown in Figure 4(b).

One can see from Figure 4(b) that again with the λ increase the supplied power P_2^* increases almost linearly, but the power P_1^* tends to some limit.

If we substitute $n = \infty$ into the formula (5), we obtain one more simple expression for the driven disc efficiency:

$$\eta_1 = -\frac{F_{sx}^*}{m_O^*\lambda}. \tag{22}$$

The latest formula can be used to calculate the efficiency of the DD when the disc operates in hard soils, for which cutting resistance forces on the disc blade can be neglected in comparison with the friction forces on the disc lateral surfaces (n >> 1).

Calculations show that in the general case the efficiency satisfies the following inequality: $min(\eta_0, \eta_1) \le \eta \le max(\eta_0, \eta_1)$. For the calculation of DD efficiency by the formulas (19), (21) or (22), several formulas (16) are used, as well as the formulas (5)–(10), (13), and (19).

The results of calculations on a personal computer are shown in graphical form in Figure 5. The DD efficiency dependences on λ have the graphs which are shown in Figure 5(a).

They were plotted for the value of ξ = 0.4 and for three values of n = 0 (curve 1), n = 3 (curve 2), and $n = \infty$ (curve 3). The presence of the DD efficiency maximum is explained by the earlier discussed changes of the P_1^* and P_2^* powers with the λ increase.

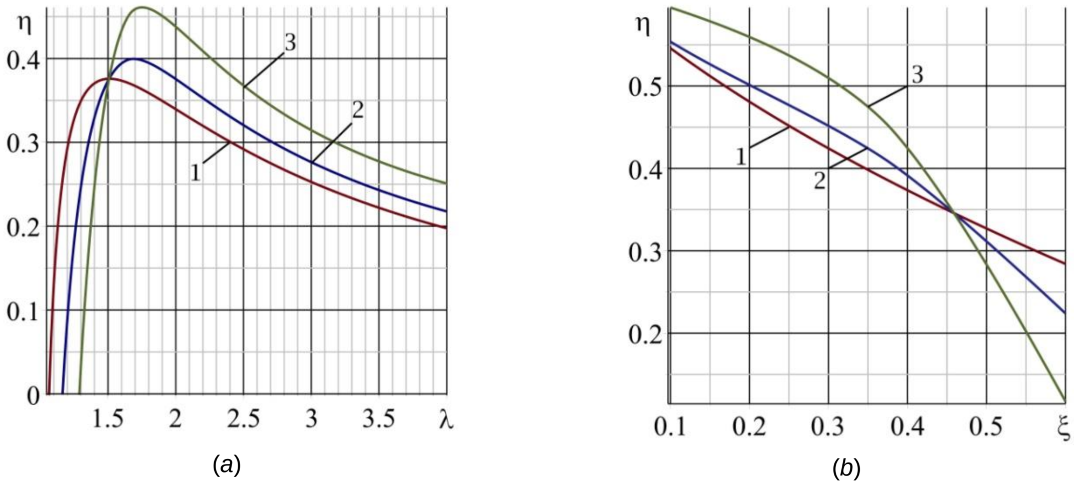


Fig. 5 - Dependence of the driving disc efficiency η
(a) on the kinematic coefficient λ; (b) on the relative depth ξ

The driving force of the disc knife is vanished at the λ values close to one. One can see from the graphs that at these λ values the DD efficiency is too close to zero. Already at relatively small λ values the η coefficient sharply increases to its maximum value 0.38 – 0.51 at some optimal value λ^* from the range $1.5 \le \lambda \le 1.75$ (depending on soil properties). Then it gradually and almost uniformly decreases to $\eta \approx 0.2$ at the point $\lambda = 5$.

The result is in good agreement with the field experimental data (*Akimov, 1976*), according to which the maximum value of the efficiency of a tillage unit with the driving disc with radius of 25 cm and a cutting depth of 9 – 10 cm (ξ = 0.36 – 0.4) corresponded to $\lambda \approx 2$.

The Figure 5(b) shows the graphs of the DD efficiency dependence on ξ at λ = 1.6 for the same values of n. It follows from them that the efficiency of the disc decreases monotonically with an increase in the relative depth ξ, and almost linearly at n = 0 and 3, and at $n = \infty$ the graph of η versus λ is close to a parabola. With an increase in the depth of the location of the knife part interacting with the soil, the projections of elementary cutting resistance forces and elementary friction forces on the direction opposite to the direction of disc movement increase. A decrease in the relative depth of the disc (i.e., an increase in its diameter) at the given operating depth leads to an increase in the DD efficiency. However, the increase in diameter is limited by the requirement of low material consumption. Therefore, a reasonable compromise is required when we choose the disc diameter.

Comparative tests of a ploughing unit with the DDs and a conventional ploughing unit (without driving discs) directly confirmed the effectiveness of using DDs to reduce slippage of an energy saturated tractor. The tests have shown that the slippage of the driving wheels of the tractor with DDs does not exceed 22%, and for the control unit it reaches 35%. In addition, the acceptable slippage was observed for the control unit already at a speed of 4 km/h, while for the experimental unit only at a speed of 7.5 km/h.

CONCLUSIONS

The technique has been developed to determine the optimal parameters of the driving disc for maximum efficiency criterion. The experimental results confirmed the adequacy of the technique. The discrepancy between the predicted and field experimental values of driving forces and applied moments was about 25%.

The numerical simulation showed, that:

- the position of the point of the DD maximum efficiency weakly depends on the parameter n, therefore the optimal modes are quite close even for different types of soils;

- the values of the efficiency of the discs-movers at the maximum points are quite high, so the powered discs can effectively perform the function of movers;

- an increase in the diameter of the discs with their constant deepening leads to an increase in the efficiency of the DD;

- for the constant depth of the disc equal to h = 10 cm, with an increase in its diameter from 50 to 75 cm and a simultaneous decrease in the kinematic parameter from 2 to 1.4–1.5, the efficiency of the DD increases from 37 to 48%.

The developed technique for optimizing the parameters of the DD can be modified to optimize the parameters of other powered rotary tools of tillage machines and units.

REFERENCES

[1] Akimov A. P. (1976), Influence of the operation mode of discs-movers on the energy characteristics of tillage unit (Влияние режима работы дисков-движителей на энергетические показатели почвообрабатывающего агрегата), *Transactions of the Izhevsk Agricultural Institute, Questions of Theory and Exploitation of Tractors and Cars (Труды Ижевского сельскохозяйственного института, Вопросы теории и эксплуатации тракторов и автомобилей)*, 33–37.

[2] Akimov A. P., & Konstantinov Yu. V. (2017), *Optimization of parameters and operation modes of discs of tillage machines and implements (Оптимизация параметров и режимов функционирования дисков почвообрабатывающих машин и агрегатов)*. Chuvash State Agricultural Academy pbl, ISBN 978-5-906305-24-4, 108 p.

[3] Akimov A. P., Medvedev V. I. (2004), *Rotary operating tool-movers (Ротационные рабочие органы-движители),* Moscow State Open University pbl., 233 p.

[4] Battiato A., & Diserens E. (2017). Tractor traction performance simulation on differently textured soils and validation: A basic study to make traction and energy requirements accessible to the practice. *Soil & Tillage Research,* 166, 18–32.

[5] Eto R., Sakata K., & Yamakawa J. (2018). Driving force distribution based on tyre energy for independent wheel-drive vehicle on rough ground. *Journal of Terramechanics*, 76, 29–38.

[6] Janulevicius A., Juostas A., & Pupinis G. (2019). Estimation of tractor wheel slippage with different tire pressures for 4wd and 2wd driving systems. *Proc. Int. Conf. Engineering for rural development, Jelgava, Latvia*, 18, 88-93.

[7] Medvedev V. I. (1972), *Power engineering of machine units with tools-movers (Энергетика машинных агрегатов с рабочими органами-движителями),* Chuvash Book Publishing House, 180 p.

[8] Misuno O. I. (2018). Reducing power losses due to slipping of a wheeled tractor on ploughing. *Proc. Int. Conf. Technical support of innovative technologies in agriculture*, *Minsk*, 229–231.

[9] Morrison J., Allen J. (1987). Planter and drill requirements for soils with surface residues. *Proc. No-till Conf. Conservation Tillage: Today and Tomorrow Southern Region, College Station, Texas, USA*, 44–58.

[10] Musin R. M., Mingalimov R. R., Gashenko A. A. (2011). Influence of driven discs on the movement stability of the cultivator unit in the horizontal plane and its comparative assessment with the serial unit. *Tidings of the Samara State Agricultural Academy*, 3, 79–82.

[11] Nerli N. (1929–1930a), On the dynamic advantage of the rotating coulter (Sul vantaggio dinamico del coltro rotante), Pubblicazioni della R. *Scuola d'ingegneria di Pisa*, 3 (127).

[12] Nerli N. (1929–30b). On the dynamic problem of the plough disc (Sul problema dinamico dell' aratro a disco), Annali delle Università Toscane. *Sezione delle Scienze Mediche, Fisiche, Matematiche e Naturali, Nuova Serie*, 48 (14), 51–78.

[13] Sarauskis E., Masilionyte L., Romaneckas K., Kriauciuniene Z., Jasinskas A. (2013). The effect of the disc coulters forms and speed ratios on cutting of crop residues in no-tillage system. *Bulg. J. Agric. Sci.*, 19 (3), 620–24.

[14] Serguntsov A. S. (2017). Traction resistance reduction of row cultivator. *Proc. Conf. Scientific support of the agroindustrial complex*, *Krasnodar*, *Russia*, 460–461.

[15] Sineokov G. N. (1949). *Disc tools of tillage machines (Дисковые рабочие органы почвообрабатывающих машин)*. Mashgiz, 86 p.

[16] Skakun T. S., Flaischer N. M. (1981). To the dynamics of the disc coulter. *Transactions of the VIM: Parameters of promising tillage tools and machines* 90, 138–145.

[17] Zoz F. (1972). Predicting tractor field performance. *Trans. ASAE*, 15, 249–255.