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CALCULATION OF THE PERFORMANCE INDICATORS OF MACHINE AND TRACTOR AGGREGATES USING BIOFUEL

Summary

The existing methods for the calculation of the performance indicators of machine and tractor aggregates (MTA) do not take into account the effect of biodiesel fuel on their operation and therefore require corrections considering the key and associated factors. The aims of the investigation are clarification of mathematical dependencies in case another kind of fuel is used and development of a methodology for the calculation of the efficiency of the MTA. As a result of simulation of the MTA draft indicators, the effect of biodiesel fuel on the tractive characteristics of the MTA power unit has been investigated under different operating conditions.
Key words: *biodiesel fuel, machine and tractor aggregate, efficiency*

KALKULACJA WSKAŹNIKÓW WYDAJNOŚCI MASZYN ZAGREGOWANYCH Z CIĄGNIKIEM ZASILANYM BIOPALIWEM

Streszczenie

Istniejące metody kalkulacji wskaźników wydajności maszyn zagregowanych z ciągnikiem nie biorą pod uwagę wpływu biopaliwa na ich pracę i dlatego wymagają korekt po rozważeniu kluczowych i powiązanych czynników. Kierunki badań są wyjaśnieniem matematycznych zależności, jeżeli inny rodzaj paliwa jest używany i rozwój metodologii dla kalkulacji skuteczności maszyn zagregowanych z ciągnikiem. Jako wynik symulacji wskaźników maszyn zagregowanych z ciągnikiem w fazie projektu, wpływ biopaliwa na pociągowe cechy bloku energetycznego maszyn zagregowanych z ciągnikiem został zbadany na mocy innych warunków eksploatacyjnych.
Słowa kluczowe: *biopaliwo, maszyny zagregowane z ciągnikiem, wydajność*

1. Introduction

The existing mathematical models of the machine and tractor aggregates (MTA) do not give a full idea about their operation when biofuel is used, and not always do they provide a possibility for adequate estimation of the changes that occur in the process of operation of the aggregate when it changes the kind of fuel. For this reason it is necessary to clarify the existing dependencies [1]. Investigation of the work of machine and tractor aggregates was conducted by: B. Rodichev, A. Guskov, V. Anilovich et al. In work [2] a dynamic model of a machine and tractor aggregate is offered with variable parameters of the condition. Mathematical simulation of the MTA draft loading is discussed in [3]. A calculus and ways how to improve the performance indicators are presented in works [5-6]. Works [7-8] deal with the issues how to raise the efficiency of the MTA at the expense of a two-component biofuel, which is a mixture of the methyl ether of the canola oil (20%) and a diesel fuel (80%).

The issues of mathematical simulation of the interrelation of the parameters of a tractor engine with the operational parameters of a mobile machine are studied in works [9-10], the operation of an engine using biofuel is considered in [11]. In work [12] the tyre road hold of the driving wheels of the tractor is examined. In order to determine the basic operational parameters of the MTA, it is also important to take into account the basic familiar dependencies presented in [13-14], and for the calculation and simulation of the tractive characteristics of the tractor it is necessary to know its technical specifications [15-16].

The aims of the investigation are clarification of mathematical dependencies in case another kind of fuel is used and development of a methodology for the calculation of the efficiency of the MTA on the basis of its draft indicators.

2. Materials and methods

The methods applied in this work are based on mathematical simulation of the physical and physico-mechanical processes that take place in the MTA structural units and are considered successively one after the other. Let us consider a towed (*тяговый*) machine and tractor aggregate consisting of a tractor and an operator-guided agricultural machine. The tractor, in its turn, consists of such structural elements: the engine, the transmission, the undercarriage and the control mechanisms, the operational equipment. Interaction of these units and mechanisms is characterised by a multitude of various processes which successively affect each other. It is a mathematical description of these processes that is the task of the simulation of the MTA operation. The energy estimation of the MTA operation presupposes determination of the tractive power balance of the tractor. In order to determine the tractive power on a particular gear, a concept of the load factor k_{ed} of the engine can be used [17]:

$$k_{ed} = \frac{R_T + f \cdot G}{P_{ti}} \leq k_{ed\lim}, \tag{1}$$

where:
R_T – the draft resistance of the machine, *kN*;

f – the coefficient of travel resistance;

G – the operational weight of the tractor, *kN*;

P_{ti} – the tangential tractive force on the i-th gear, *kN*;

$k_{ed\,\lim}$ – the allowed load factor of the engine.

In order to calculate the draft resistance R_T of the machine, depending on the kind of agricultural work, we will select a structure of the aggregate and choose this or that formula.

For instance, calculation of the draft resistance of a ploughing aggregate MTZ-80+PLN-3-35, used for ploughing, is carried out according to the familiar V.P.Goryachkin's formula [15]:

$$R_T = f_0 \cdot Q + a_{pc} \cdot b_{pc} \cdot n_{pc} \cdot \left(K_p + \varepsilon_{pc} \cdot V^2\right), \qquad (2)$$

where:

f_0 – the coefficient of the first term of the rational V.P.Goryachkin's formula;

Q – the weight of the tool (plough) in the composition of the aggregate, *N;*

K_p – the specific resistance of the plough, $N \cdot m^{-2}$;

a_{pc}, b_{pc}, – the depth of ploughing and the working width of the plough body respectively, m;

n_{pc} – the number of the plough bodies with a respective number of coulters;

ε_{pc} – a coefficient depending on the parameters of the working surface of the plough body and the soil properties, $N \cdot s^2 \cdot m^{-4}$;

V – the working travel speed of the aggregate, *m s*$^{-1}$

To determine the travel speed v of the aggregate, we use the familiar formula [12]:

$$V = \frac{\pi \cdot n}{30} \cdot \frac{r_r}{U_t} \cdot \left(1-\delta\right), \qquad (3)$$

where:

n – the rotation frequency of the engine crankshaft, $\min^{-1}$;

r_r – the rolling radius of the driving wheel, *m*;

U_t – the gear reduction rate;

δ – the skidding coefficient.

Let us determine the rolling radius r_r of the driving wheel according to the formula:

$$r_r = 25.4 \cdot 10^{-3} \cdot \left(0.5 \cdot d_{dw} + 0.8 \cdot b_{bw}\right), \qquad (4)$$

where:

d_{dw}, b_{bw} – the diameter at rim seat and the tire section width of the driving wheel in inches.

The gear reduction rate U_t of the tractor is calculated as a product of the gear reduction rates of its units and mechanisms. Knowing the kinematic scheme of the tractor [14], we build a formula to determine the total gear reduction rate of the tractor:

$$U_t = U_{mgb} \cdot U_{rg} \cdot U_{mg} \cdot U_{fg}, \qquad (5)$$

where:

$U_{mgb}, U_{rg}, U_{mg}, U_{fg}$ – the respective gear reduction rates of: the gearbox, reduction unit (*редуктора*), the main and the final gear.

By substituting formulae (4, 5) into formula (3), we obtain:

$$V = \frac{0{,}423 \cdot 10^{-3} \cdot \left(d_{dw} + 1{,}66 \cdot b_{bw}\right) \cdot \pi \cdot n}{U_{mgb} \cdot U_{rg} \cdot U_{mg} \cdot U_{fg}} \cdot (1-\delta). \qquad (6)$$

In order to determine the skidding coefficient δ, we use the empirical dependency [17]:

$$\delta = a \cdot \left(f + \frac{\varphi_h}{\lambda_{вк}}\right) + b \cdot \left(\frac{\varphi_h}{\lambda_{вк}}\right)^4, \qquad (7)$$

where:

a, *b* – the empirical coefficients depending on the medium soil hardness and pressure onto the ground;

φ_h – the coefficient of the used adhesive force (*коэффициент использования сцепного веса*);

λ_{dw} – the load factor of the driving wheels.

The empirical coefficients *a*, *b* are found according to formula [17]:

$$a = 0.25 \cdot \sqrt[3]{\frac{q_a}{\sigma}}, \qquad (8)$$

$$b = 2{,}5 \cdot \sqrt[3]{\frac{q_a}{\sigma}}, \qquad (9)$$

where:

q_a – the medium pressure of the wheeled driver of the tractor upon the soil, $kg \cdot cm^{-2}$;

σ – the medium hardness of the soil, $kg \cdot m^{-2}$.

We recommend determining the medium pressure q_a for the driving wheels according to the formula:

$$q_a = \frac{\lambda_{dw} \cdot m_{tr}}{2 \cdot F_{cp}}, \qquad (10)$$

where:

m_{tr} – the operational mass of the tractor, *kg*;

F_{cp} – the area of the contact plot of the wheel with the soil (cm^2), which is defined as:

$$F_{cp} = F_c \cdot K_1, \qquad (11)$$

where:

F_c – the contour area of the contact, cm^2;

K_1 – a coefficient depending on the tire diameter of the wheel.

Area F_c is found out according to a simplified formula on the basis of [13] and considering the size of the tires in inches:

$$F_c = 25.4 \cdot 10^{-3} \cdot b_{dw} \cdot L, \qquad (12)$$

where:

L – the length of the support surface, determined by A.V. Guskov's methodology [13], *cm*:

$$L = \frac{D_r}{2} \cdot \alpha_0 + \sqrt{D_r \cdot h}, \qquad (13)$$

where:

D_r – the reduced (*приведённый*) diameter of the driving wheel, *m*;

α_0 – the angle of the covered (*охвата*) support surface of the wheel, *rad*;

h – the depth of the track, *m*.

Diameter D_r is determined according to formula [13]:

$$D_r = D_0 \cdot \left(1 + \frac{2 \cdot m_{ltrw}}{\sigma \cdot D_0}\right), \tag{14}$$

where:

D_0 – the nominal diameter of the tire, *m*;

m_{ltrw} – the mass of the tractor per one driving wheel, *kg*.

Depending on the rolling conditions of the deforming wheels along the deformed surface, we select D_0 for an external diameter of the wheel in a free state. Thus, by analogy with formula (4), we determine:

$$D_0 = 25.4 \cdot 10^{-3} \cdot \left(d_{dw} + 1.6 \cdot b_{dw}\right). \tag{15}$$

Depending on the distribution of the operational weight on the axle of the tractor, we determine:

$$m_{ltrw} = \frac{m_{tr} \cdot \lambda_{dw}}{2}. \tag{16}$$

Angle α_0 is determined according to formula [13]:

$$\alpha_0 = arctg \sqrt{\frac{D_r \cdot h - h^2}{\frac{D_r}{2} - h}}. \tag{17}$$

The depth of the track [13]:

$$h = \frac{m_{ltrw}^2}{645.16 \cdot 10^{-6} \cdot K^2 \cdot b_{dw}^2 \cdot D_r^3}, \tag{18}$$

where:

K – the coefficient of the volumetric deformation of the soil, $kg \cdot m^{-3}$.

3. Results and discussion

By substituting formulae (11-18) into formula (10), we obtain an equation for the medium pressure of the wheeled driver (*колесного движителя*) of the tractor onto the soil:

$$\begin{aligned}
q_{\text{ср}} = {} & m_{\text{тр}} \cdot \lambda_{\text{вк}} \cdot 10^{-4} \Big/ \Big[645.16 \cdot 10^{-6} \cdot b_{\text{к}} \cdot K_1 \cdot \left(d_{\text{к}} + 1.6 \cdot b_{\text{к}}\right) \times \\
& \times \left(1 + \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{25.4 \cdot 10^{-3} \cdot \sigma \cdot \left(d_{\text{к}} + 1{,}6 \cdot b_{\text{к}}\right)}\right) \times \\
& \times arctg \Bigg\{ \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{1290.32 \cdot 10^{-6} \cdot K \cdot b_{\text{к}} \cdot \left(d_{\text{к}} + 1.6 \cdot b_{\text{к}}\right) \cdot \left(1 + \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{25.4 \cdot 10^{-3} \cdot \sigma \cdot \left(d_{\text{к}} + 1{,}6 \cdot b_{\text{к}}\right)}\right)} \times \\
& \times \Bigg[\Bigg(1 - \frac{m_{\text{тр}}^2 \cdot \lambda_{\text{вк}}^2}{0.423 \cdot 10^{-7} \cdot K^2 \cdot b_{\text{к}}^2 \cdot \left(d_{\text{к}} + 1{,}6 \cdot b_{\text{к}}\right)^3 \cdot \left(1 + \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{25.4 \cdot 10^{-3} \cdot \sigma \cdot \left(d_{\text{к}} + 1.6 \cdot b_{\text{к}}\right)}\right)^3} \Bigg) \times \\
& \times 1 \Big/ \Bigg(12.7 \cdot 10^{-3} \times \left(d_{\text{к}} + 1.6 \cdot b_{\text{к}}\right) \cdot \left(1 + \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{25.4 \cdot 10^{-3} \cdot \sigma \cdot \left(d_{\text{к}} + 1.6 \cdot b_{\text{к}}\right)}\right) - \\
& - \frac{m_{\text{тр}}^2 \cdot \lambda_{\text{вк}}^2}{0.423 \cdot 10^{-7} \cdot K^2 \cdot b_{\text{к}}^2 \cdot \left(d_{\text{к}} + 1.6 \cdot b_{\text{к}}\right)^3 \cdot \left(1 + \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{25.4 \cdot 10^{-3} \cdot \sigma \cdot \left(d_{\text{к}} + 1.6 \cdot b_{\text{к}}\right)}\right)^3} \Bigg) \Bigg]^{0,5} \Bigg\} + \\
& + \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{25{,}4 \cdot 10^{-3} \cdot K \cdot \left(d_{\text{к}} + 1{,}6 \cdot b_{\text{к}}\right) \cdot \left(1 + \frac{m_{\text{тр}} \cdot \lambda_{\text{вк}}}{25{,}4 \cdot 10^{-3} \cdot \sigma \cdot \left(d_{\text{к}} + 1{,}6 \cdot b_{\text{к}}\right)}\right)} \Big]
\end{aligned} \tag{19}$$

Coefficient φ_h is found according to formula [18]:

$$\varphi_h = \frac{P_h}{G}, \tag{20}$$

where:

P_h – the draft force on the hook of the tractor (*kN*), determined by the familiar dependency [14]:

$$P_h = P_t - P_f, \tag{21}$$

where:

P_f – the resistance to self-propulsion of the tractor, *kN*.

To determine the tangential tractive force P_t, we use the following dependency [18]:

$$P_t = \frac{U_t \cdot \eta_t}{r_r} \cdot M_e, \tag{22}$$

where:

η_t – the coefficient of efficiency of the tractor transmission

M_e – the efficient engine torque, *N·m*.

On the basis of the kinematic scheme of the transmission, as well as using the familiar dependency [14, 16], we determine:

$$\eta_t = \eta_{cgt}^{K_c} \cdot \eta_{bgd}^{K_b} \cdot \eta_{it}, \tag{23}$$

where:

η_{cgt}, η_{bgd}, η_{it} – the respective coefficients of efficiency: of the cylindrical and the conical gears, as well as in idle running of the transmission;

K_c, K_b – the quantity of cylindrical and conical gears in the transmission.

Substituting formula (22) of dependency (4-5, 23), we obtain:

$$P_t = \frac{M_e \cdot U_{mgb} \cdot U_{rg} \cdot U_{mg} \cdot U_{fg} \cdot \eta_{cgt}^{K_c} \cdot \eta_{bgd}^{K_b} \cdot \eta_{it} \cdot 78{,}74}{(d_{dw} + 1.66 \cdot b_{bw})}. \quad (24)$$

Parameter M_e in formula (24) considers the connection of the draft parameters of the tractor with the parameters of the engine efficiency.

The interrelation of the efficient engine parameters are described by the familiar dependency [14]:

$$M_e = \frac{9550 \cdot N_e}{n}, \quad (25)$$

where:

N_e – the efficient capacity of the engine, *kW*.

Consequently, formulae (24-25) reflect the connection of the efficient parameters of the engine with the draft parameters of the tractor, which affect the draft force P_h (11) and, in their turn, the skidding coefficient δ (7, 20) and, as a result, the MTA energy parameters.

Let us simulate changes of the tangential force P_t depending on the efficient capacity N_e of the engine by using the data presented in article [16], where for engine *4Ch11.0/12.5* (the manufacturer's mark D-240), there are investigated changes of N_e and the rotation frequencies n of the engine crankshaft in an experimental way, using a diesel and a biodiesel fuel B100 for all the working conditions.

The technical specifications of a tractor of the drawbar category 1.4 with engine *4Ch11.0/12.5* are given in work [20].

Using indicators N_e and n as external characteristics of the engine [12], after formulae (14-15) we obtain a dependency $N_e = f(P_{ti})$ on various transmissions. We perform simulation of the change of such a dependency by means of electronic tables Microsoft Excel. On the basis of the results of calculation with the help of the grapho-mathematical editor *OriginPro 8* we build graphs of the obtained dependencies for four working transmissions (see Fig. 1).

Analysis of the obtained graphs, for example, for transmission *VIImg* show that for the diesel fuel the maximum efficient capacity (N_e = 55.2 *kW*) is achieved at value $P_t = 15.31$ *kN*, for the biodiesel fuel – the value of maximum capacity is N_e = 53.67 kW at $P_t = 15.63$ *kN*, which is a witness of insignificant improvement of the tractive characteristics of the power unit within the MTA under nominal conditions of the engine and under conditions close to the nominal. At the maximum value of the tangential tractive force of the tractor for the same transmission: when the diesel fuel is used ($P_t = 18.32$ *kN*), the efficient capacity is N_e = 40.89 *kW*; when the biodiesel fuel is used ($P_t = 18.01$ kN) – N_e = 37.11 *kW*.

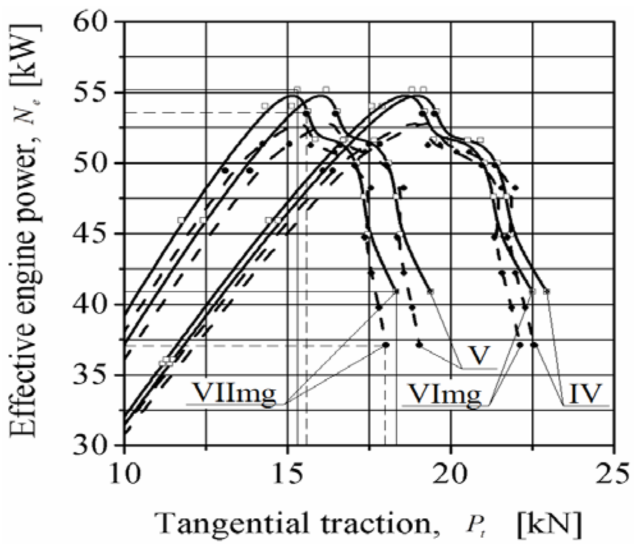

Fig. 1. Dependence of the efficient capacity N_e of the engine on the tangential tractive force P_t of the tractor: —□——□——□— - for the diesel fuel; —• — —• — —• — - for the biodiesel fuel; VImg, VIImg – for transmissions with a reduction gear; IV, V – for transmissions without a reduction gear

As a result, a dependency of the efficient capacity of the engine on the tangential tractive force is obtained for four basic transmissions, using a diesel and a biodiesel fuel. This dependency is produced on the basis of experimental data [16], and it characterises the effect of the kind of fuel on the draft characteristics of the tractor. However, when doing calculations, one should consider also extraneous factors which arise in interaction of the wheeled drivers (*движителей*) with the soil and may level out or increase the effect of the kind of fuel on the MTA tractive characteristics. This leads to an error in calculations and, consequently, to incorrect conclusions. To avoid this and to make the calculations more exact, in the present methodology there are formulae (7-9, 19) used which take into account the impact of the skidding coefficient δ upon the draft characteristics.

4. Conclusions

1. A mathematical model is offered which considers not only the basic working processes that take place in the structural elements of the MTA but also the physical and physico-mechanical properties of the soil. Such a scientific approach provides a possibility for a more objective assessment of the effect of biofuel on the performance and technological indicators of the machine and tractor aggregate.
2. The results of the simulation of the MTA tractive indicators show that under nominal conditions and conditions close to the nominal the use of the biodiesel fuel can raise the tractive characteristics of the energy unit of the MTA by 2.1 %; but when the engine is working under overload conditions, the use of the biodiesel fuel reduces the tractive properties of the energy unit of the MTA by 1.7%.

5. References

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