Original Article

COMPLEX PERFORMANCE INDICATORS OF MACHINE AND TRACTOR UNITS

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Abstract

Full compliance of the performance indicators of machine-tractor units with structural, agrotechnical, environmental and other requirements ensures compliance with the standard indicators of the quality of agricultural products produced and grown at the required level.

Keywords: Machine, tractor, unit, load, coefficient, engine, frequency, kinematic radius, specific fuel consumption, fuel consumption, slip coefficient, blocking, transmission, full gear.

INTRODUCTION

In tractor theory, when analyzing the performance of mainly agricultural MTAs, the coefficient of engine operational load by power is also used

$$k_N = N_e / N_н. \quad (1)$$

where kN- the coefficient of operational loading of the engine; Ne – effective engine power; Nн - rated (calculated) engine power.

From the analysis of this formula, it follows that if the tractor engine operates only on the external regulatory branch of the characteristic, then the rotation frequency of the nд engine shaft differs little from the nominal frequency nн, therefore, it can be assumed that kN=kм. Load factor value kN depends on the type of s.h. operation performed by table 1 below shows the values of this coefficient, as well as the probability p of performing various technological operations with the MTZ – 80X and New-Holland wheeled tractor during the year [1,2]

Table-1 Values of the coefficient kN and probability P

s.h.operation	$к_N$	P
Cultivation	0,95	0,27
Sowing and planting	0,90	0,04
Care of crops	0,90	0,15
Soil fertilization	0,80	0,09
Harvesting	0,80	0,22
Cargo transportation	0,65	0,23

The kinematic radius of the $r_к$ is the radius of such a fictitious wheel, which, when rotating with a given angular velocity $\omega_к$, moving without sliding on the ground surface, has the same translational velocity of its axis v, which has a real wheel. This radius determines the path traversed by the wheel in one revolution and is determined by the formula [1].

$$r_к = v / \omega_к = v_т \cdot (1- \delta) / \omega_к = r_д \cdot (1- \delta). \quad (2)$$

where $r_к$ – wheel radius; v-speed;; $\omega_к$ – angular velocity; $v_т$ - theoretical speed; δ- slipping coefficient; $r_д$- dynamic rolling radius.

The radius of the $r_к$ is a variable value, because it depends on the value of the slip coefficient (or yuz) δ. The value of the coefficient δ is calculated in the same way as for the case of rolling of the undeformable wheel and the support surface, according to the formula (3). However, it should be noted that the pneumatic wheel is characterized by sliding of its entire support platform ("spots contact"). Moreover, individual points of this platform located along its longitudinal axis of symmetry, due to the fact that they are located at different distances from the center of the wheel O and the tire has tangential elasticity, slide with different absolute velocity $v_a = v_{букс} = v_т – v$ is relatively expensive. Therefore, in theoretical studies, the lowest sliding speed is assumed, which is possessed by the point of the outer surface of the tire coming into contact with the support surface of the road, and this value $v_{букс}$ determines the value of the coefficient δ. In this case, the value of the theoretical speed is determined by the formula

$$v_т = r_д \cdot \omega_к. \quad (3)$$

From the analysis of expression (3), it follows that when the wheel is skidding (δ>0), the kinematic radius of the $r_к$ is less than the dynamic radius of the $r_д$, by the amount of Δr, and when the wheel is swinging (δ < 0), it is greater by the amount of Δr, where $\Delta r = = r_д \cdot \delta$.

Determine the static load on a tractor tire 13.6 R 38 YAR-318 if the width and height of its profile are respectively equal to b = 0.42 m and H = 0.43 m, and the air pressure in the tire is 150 kPa with a static radius of the tire $r_{ст}$ = 0.7 m [1]

-Determine the free radius of the tire 13,6 R 38 YAR-318,

$$r_o = D / 2 + H = 76 / 2 + 0,43 = 0,38 + 0,43 = 0,81 \text{ м},$$

where the landing diameter of the tire rim is

$$D = 30'' = 30 \cdot 0,0254 = 0,76 \text{ м}.$$

-According to the well-known formula, we determine the coefficient of normal tire stiffness 13,6 R 38 ЯР-318,

$$\lambda_н = 2 \cdot \pi \cdot p_ш \cdot (r_o \cdot r_c) \ 0,5 = 2 \cdot 3,14 \cdot 80 \cdot (0,81 \cdot 0,21) \ 0,5 = 259 \text{ кН / м2} = 259 \text{ кПа},$$

where is the radius of the tire rim section $r_c = b / 2 = 0,42 / 2 = 0,21$ м.

-We determine the static deformation of the tire

$$h_{ст} = r_o - r_{ст} = 0,81 - 0,70 = 0,11 \text{ м}.$$

-We determine the static weight load on the tire

$$Q_{ст} = h_{ст} \cdot \lambda_н = 0,11 \cdot 259 = 28,5 \text{ кН}.$$

The fuel consumption of $G_т$ in the area of the regulatory characteristic, i.e. in the interval ($n_н$, $n_х$), can be represented by a linear function varying from $G_{тн} = g_{ен} \cdot N_н/1000$ до $G_{тх} = (0,2...0,3) \cdot G_{тн}$, and on the unregulated branch (corrector branch) of the external speed characteristic - a nonlinear function [2].

$$G_т = G_{тн}[(1 - \beta)(g \cdot k_м^2 - 1) / (k_м^2 - k_м) + \beta^2] \cdot N_e / N_н, (2.14)$$

where Gт - hourly fuel consumption; Gтн - fuel consumption of the accumulator position; β - coefficient; g - specific fuel consumption; kм - coefficient of the torque adaptable; β = nд / nн.

The specific fuel consumption is carried out using the formula

$$ge = 1000\ Gт / Ne.$$

In this dimension formula Gт and Ne accordingly: г/(кВт-ч), кг/ч и кВт.

Blocked drive. The most widespread on full-drive tractors was the blocked drive. As already noted, when the drive is blocked, the front and rear drive axles are kinematic interconnected. Therefore, if one of the driving axes of the tractor reduces the angular speed of rotation, for example, by half, then the other axis will also decrease its angular speed by exactly half [3,4]

Consider the rectilinear movement of a tractor on a flat road in the presence of some difference in the theoretical speeds of the front and rear wheels.

The alignment of the translational speeds of both driving axles, i.e. ensuring equality of the actual speeds (v1 = v2 = v), can be provided only under the condition of a certain slipping or skidding of the wheels, since slipping reduces the translational speed of the wheel axis, and the skidding increases it.

The coefficient of kinematic mismatch of the rear and front driving wheels

For each tractor, this coefficient will be different, and it may vary depending on the working conditions. From the analysis of expression (4) it follows that the coefficient кн is always greater than one.

There is a certain dependence between the slipping of the running wheels and the slipping of the surfaces of the lagging wheels, which, based on the formula (6.1), is expressed by the ratio [1,2]

$$\delta1 = 1 - (vт2 / vт1) \cdot (1 - \delta2) = 1 - кн \cdot (1 - \delta2). (4)$$

The value of δ2 in this expression is always positive, because the running wheels always work with some slipping. In lagging wheels, the slipping of δ1 can be negative, zero and positive. If δ1 has a negative value, then the lagging wheels move with a skid, if δ1 = 0, then they roll without a skid and slipping, if δ1 has a positive value, then the lagging wheels work with slipping, but the amount of slipping is less than that of the running wheels.

The best traction characteristics of the tractor could be obtained if the equality of the circumferential speeds of the front and rear wheels, i.e., provided that the coefficient of kinematic discrepancy кн = 1. In this case, the front and rear wheels would work with the same slipping δ1 = δ2 and their coupling qualities would be used equally.

The presence of a kinematic mismatch of the wheels worsens the tractor's traction performance. If, as a result of a kinematic discrepancy, the front and rear wheels work with different slipping, then the coupling qualities of lagging wheels are used to a lesser extent than the coupling qualities of running wheels.

The greater the kinematic discrepancy, the more uneven the coupling qualities of the wheels of both axles are used. The most negative effect on tractor traction performance is the use of lagging wheels. In this case, only the running wheels remain the driving wheels, since the lagging wheels roll with the skid, therefore, they become driven [3,4].

The angular velocity of the engine shaft is set ωд = 200 rad / s, the transmission ratio u2 = 50 to the rear driving wheels, the dynamic radius of which is equal to rд2 = 0.5 m, and their slipping is equal to δ2 = 0.12. In addition, the value of the kinematic mismatch coefficient of the front and rear driving wheels of the New Holland tractor кн = 1.04 is known when their drive is blocked.

Determine the theoretical and actual translational speeds of the front and rear wheels [5,6]

-Determine the angular velocity of the rear driving wheels

$$\omega 2 = \omega д / u2 = 200 / 50 = 4 \text{ рад/с}.$$

where u2 - transmission gear ratio.

-We determine the theoretical speed of the rear driving wheels

$$vт2 = \omega 2 \cdot rд2 = 4 \cdot 0,5 = 2 \text{ м /с}.$$

- We determine the theoretical speed of the front driving wheels

$$vт1 = vт2 / кн = 2 / 1,04 = 1,923 \text{ м /с}.$$

- We determine the slipping of the front driving wheels

$$\delta 1 = 1 - кн \cdot (1 - \delta 2) = 1 – 1,04 \cdot (1 – 0,12) = 0,085.$$

- We determine the actual speeds of the front and rear wheels of the tractor

$$v1 = vт1(1 - \delta 1) = 1,923(1 – 0,085) = 1,76 \text{ м /с};$$

$$v2 = vт2(1 - \delta 2) = 2(1 – 0,12) = 1,76 \text{ м /с}.$$

Conclusion: As expected, the actual speeds of the front and rear speeds are the same.

The New Holland four-wheel drive tractor with a 4K4a wheel formula is equipped with the MA to eliminate power circulation. Determine the required ratio of gear ratios to the driving axles of the tractor, if it is known that the ratio of static radii, driving front and rear wheels is equal to rст1 / rст2 = 0,6.

We take, in accordance with the formula, the value of the kinematic discrepancy coefficient equal to кно = 1.05.

Using the formula, we determine the required ratio of gear ratios

$$u1 / u2 = кно \cdot rд1 / rд2 = 1,05 \cdot 0,6 = 0,63,$$

where it is assumed approximately when calculating that rд1 / rд2 = rст1 / rст2.

KPD, which characterizes the loss of rolling power

In accordance with the formula (7.7), this KPD of the tractor ηf will be determined by the following expression [7].

$$\eta f = Nост / Nк = (Nк - N f) / Nк = 1 – P f / Pк. \quad (7.10)$$

When calculating methods for determining ηf, it is necessary to know the values of Pf and Pк. In traction calculation, the rolling resistance Pf is calculated according to the approximate ratio Pf = f·Gэ, and the coefficient f included in this formula is selected in traction calculation in accordance with the type of tractor and the specified ground conditions according to reference data. When working in traction mode on the stubble of the ear for wheeled tractors take ηf = 0.12, and for tracked tractors ηf = 0.08. The KPD of slipping nδ, taking into account the loss of power due to a decrease in speed due to the slipping of the tractor engine is determined by the formula

$$\eta\delta = v / v_т = 1 - \delta, \quad (7.11)$$

where the KPD of slipping δ in the calculation method is determined by the dependencies given in the previous sections [8].

To estimate the traction efficiency of a New Holland 4K2 class 1.4 wheeled tractor with an operating weight of $m_э = 4.4$ t, operating in nominal traction mode on the stubble of the ears, if it is known that the power from the engine enters the tractor's driving wheels through a cylindrical series of gears with three gearing poles and one conical pair of gears.

-By the formula we determine the mechanical KPD of the tractor power circuit

$$\eta_м = \eta_{тр} = \eta_1^{n1} \cdot \eta_2^{n2} \cdot \eta_3^{n3} = 0{,}993 \cdot 0{,}981 \cdot 0{,}990 = 0{,}95.$$

-Accepting f = 0,12, we determine the rolling resistance force of the tractor when moving it along the stubble

$$P_f = f \cdot G_э = f \cdot g \cdot m_э = 0{,}12 \cdot 9{,}81 \cdot 4{,}4 = 5{,}17 \text{ кН}.$$

-By the formula we determine the KPD of the rolling resistance of the tractor

$$\eta_f = 1 – P_f / P_к = 1 – 5{,}17 / 14 = 1 – 0{,}36 = 0{,}64,$$

where for a 4K2 wheeled tractor in nominal traction mode the value is assumed $P_к = 14$ кН.

-We determine the KPD of tractor slipping by the formula

$$\eta\delta = v / v_т = 1 - \delta = 1 – 0{,}18 = 0{,}82$$

-We determine by the formula the desired traction KPD of the tractor

$$\eta_т = \eta_м \cdot \eta_f \cdot \eta_\delta = 0{,}95 \cdot 0{,}64 \cdot 0{,}82 = 0{,}498.$$

To estimate the traction KPD of a class 1.4 with 4K4a wheeled tractor with an operating weight of $m_э = 4.4$ t, operating in nominal traction mode on the stubble of the ears, if it is known that the power from the engine is supplied to the tractor's driving wheels through transmission drives to the front and rear wheels, respectively, having mechanical KPD

$\eta_{м1} = 0{,}94$ и $\eta_{м2} = 0{,}95$.

-We determine the parameters necessary to calculate the normal reactions Y1 and Y2 according to the formulas set out above:

• operational weight of the tractor $G_э = g \cdot m_э = 9{,}81 \cdot 4{,}4 = 43{,}16$ кН;

• static wheel load factors $\lambda_{ст1} = 0{,}35$; $\lambda_{ст2} = 0{,}65$;

• we take the design parameters to be equal $h_{кр} = 0{,}4$ м и $L = 2{,}46$ м

-Using the formulas, we calculate the coefficients of the dynamic load of the wheels in the considered driving mode

$\lambda_1 = \lambda_{ст1} - P_{кр} \cdot h_{кр} / (L \cdot G_э) = 0{,}35 - 14 \cdot 0{,}4 / (2{,}46 \cdot 43{,}16) = 0{,}35 - 0{,}116 = 0{,}30;$

$\lambda_2 = \lambda_{ст2} + P_{кр} \cdot h_{кр} / (L \cdot G_э) = 0{,}65 + 14 \cdot 0{,}4 / (2{,}46 \cdot 43{,}16) = 0{,}65 + 0{,}116 = 0{,}81.$

-According to the formulas, we calculate the normal loads acting on the front and rear wheels

$$Y_1 = \lambda_1 \cdot G_э = 0{,}30 \cdot 43{,}16 = 12{,}94 \text{ кН};$$

$$Y2 = λ2 \cdot Gэ = 0{,}81 \cdot 43{,}16 = 34{,}95 \text{ кН}.$$

-According to the formula , we determine the ratio of normal loads on tractor axles

$$β = Y1 / Y2 = 12{,}94 / 34{,}95 = 0{,}370.$$

-We determine the partial rolling resistance forces of the wheels

$$Pf1 = f \cdot Y1 = 0{,}12 \cdot 12{,}94 = 1{,}55 \text{ кН};$$

$$Pf2 = f \cdot Y2 = 0{,}12 \cdot 34{,}95 = 4{,}19 \text{ кН}.$$

-We determine the total tangential traction force of the tractor

$$Pк = Pf + Pкр = (Pf1 + Pf2) + Pкр = (1{,}55 + 4{,}19) + 14 = 5{,}74 + 14 = 19{,}74 \text{ кН}.$$

-We use the formulas to determine the partial tangential traction forces of the tractor

$$Pк1 = β \cdot Pк / (1 + β) = 0{,}370 \cdot 19{,}74 / (1 + 0{,}370) = 5{,}33 \text{ кН}.$$

$$Pк2 = Pк / (1 + β) = 19{,}74 / (1 + 0{,}370) = 14{,}40 \text{ кН};$$

-We use the formulas to determine the partial KPD of the rolling resistance of the tractor wheels

$$ηf1 = 1 – P\,f1 / Pк1 = 1 – 0{,}96 / 5{,}33 = 0{,}81;$$

$$ηf\,2 = 1 – P\,f2 / Pк2 = 1 – 2{,}57 / 14{,}40 = 0{,}82.$$

-With the coefficient of slipping of the rear wheels $δ2 = 0.16$ and the coefficient of kinematic discrepancy of the front and rear wheels $кн = 1.05$, we determine by the formula the coefficient of slipping of the front wheels

$$δ1 = 1 – кн \cdot (1 – δ2) = 1 – 1{,}05 \cdot (1 – 0{,}16) = 0{,}118.$$

-We determine the KPD of the front and rear wheels slipping

$$ηδ1 = 1 - δ1 = 1 – 0{,}118 = 0{,}88;$$

$$ηδ2 = 1 – δ2 = 1 – 0{,}16 = 0{,}84.$$

- By the formula we find the coefficient $γ$

$$γ = β \cdot ηтр2 / (кн \cdot ηтр1) = 0{,}370 \cdot 0{,}95 / (1{,}05 \cdot 0{,}94) = 0{,}35.$$

- Using the formulas , we determine the coefficients s1 and s2

$$s1 = γ / (1 + γ) = 0{,}35 / (1 + 0{,}35) = 0{,}25;$$

$$s2 = 1 / (1 + γ) = 1 / (1 + 0{,}35) = 0{,}74.$$

-We use formulas to determine the partial traction KPD

$$ηт1 = ηм1 \cdot ηf1 \cdot ηδ1 = 0{,}94 \cdot 0{,}81 \cdot 0{,}88 = 0{,}67;$$

$$\eta т2 = \eta м2 \cdot \eta f2 \cdot \eta \delta 2 = 0{,}95 \cdot 0{,}82 \cdot 0{,}84 = 0{,}65.$$

-Using the formula , we determine the desired value of the tractor traction KPD[9].

$$\eta т = s1 \cdot \eta т1 + s2 \cdot \eta т2 = 0{,}26 \cdot 0{,}67 + 0{,}73 \cdot 0{,}65 = 0{,}17 + 0{,}47 = 0{,}64.$$

Experimental verification of the nominal traction force

Experimental verification of the nominal traction force of an existing tractor is carried out during traction tests of an existing tractor. To obtain experimental traction characteristics, tractor traction tests are carried out on various gears, the essence of which is as follows [10,11,12].

A field section that is homogeneous in type and condition of processing is selected or roads with smooth horizontal terrain. The tractor is loaded with a special brake trolley, another tractor or a trailed implement (most often a plow). Several experiments are carried out on each transmission of the tractor: first at idle, and then at 25, 50, 75, 85, 100 or more percent of the total traction load on this transmission until the engine stops from overload or until the tractor stops due to slipping of the propellers. In the range from 85% to full overload of the tractor, 5 ... 7 experiments are carried out. With each experiment, the traction force, the duration of the experiment, the speed of rotation of the driving wheels, fuel consumption during the experiment are measured [13,14,15].

The main backgrounds on which tests are carried out for wheeled tractors are a track with an asphalt (concrete) coating, stubble of ears, a field prepared for sowing; for tracked tractors – a clay track (clay rolled road), stubble of ears, a field prepared for sowing. The main background for industrial tractors is a clay track. The main parameters of the physical and mechanical properties of these backgrounds are given in Table.2.

Table-2 Backgrounds for tractor traction testing

Kinds Backgrounds	Type of tractor	Background humidity, %	Background hardness	
			T, МПа	$C_{уд}$
Asphalt	Wheeled	-	-	-
Stubble of ears	Crawler	8,0…15	4…6	5…12
Clay track	Wheeled and Crawler	8,0…22	1,0…1,5	1…3
The field under. under sowing	Wheeled and Crawler	8…22	0,1…0,7	0,5…1,5

According to the results of traction tests, an experimental check of the nominal traction force of the tractor is carried out, the essence of which is as follows [16,17,18].

For agricultural tractors whose slipping at the maximum traction KPD (at values $\eta \geq 0{,}95\eta Tmax$) is less than its limit value δДОП (18, 16 and 5%, respectively, for wheeled 4K2 and 3K2, 4K4 and tracked tractors), the nominal traction force PK и Pн, determined by calculation, if it is in the zone of the maximum values of the traction KPD of the experimental traction characteristic between the traction forces of PОПТ и PКР2, corresponding to the maximum KPD and the ultimate slipping [19,20].

If the PКPн, determined theoretically, does not fall into the zone of maximum KPD values, then the tractor traction force value corresponding to the nearest boundary of the zone limited by the maximum traction KPD value and the maximum slip is taken as nominal.

According to the traction characteristic, the highest value of the traction force PКPmax, is determined, at which stable operation of the engine is still possible, and then the nominal traction force is calculated by the formula

$$P_{КРн} = B \cdot P_{КРmax},$$

Where B is the coefficient for agricultural tractors - 0.5 and 0.6, respectively, for tracked and wheeled tractors;

Conclusion:

The use of machine-tractor units in agricultural work, i.e. during operation, with due observance of agrotechnical requirements, in addition to reducing the cost of production, increases labor productivity, reduces the cost of aggregates.

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