INVESTIGATION OF THE TRACTOR PERFORMANCE WHEN BALLASTING ITS REAR HALF-FRAME
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ДОСЛІДЖЕННЯ ТЯГОВИХ ПОКАЗНИКІВ ТРАКТОРА ПРИ БАЛАСТУВАННІ ЙОГО ЗАДНЬОЇ ПІВРАМИ

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ABSTRACT

When designing and determining the potential technical capabilities and characteristics of the tractor, the so-called traction calculation is carried out. The aim of the work is to study regularities of changes in the traction and coupling characteristics of a tractor with an articulated frame depending on the size of the ballast, the presence of dual wheels and other parameters, using the KhTZ-242K tractor as an example. The numerical solution of the deduced mathematical model made it possible to obtain graphical dependences of the change in the main traction and coupling characteristics of the tractor KhTZ-242K depending on the mass of the tractor, which varied from 8600 kg to 10100 kg by changing the ballast mass. The maximum tractive power of the tractor on single wheels without a ballast is 121 kW, which is achieved at a speed of 12 km·h⁻¹, and on single wheels with a ballast 122 kW, at a speed of 15 km·h⁻¹. This, in its turn, greatly affects the productivity and fuel efficiency of the aggregate.

АНОТАЦІЯ

При конструюванні та визначенні потенційних технічних можливостей і характеристик трактора важливе значення має, так званий тяговий розрахунок. В результаті теоретичних досліджень отримано закономірності зміни тягово-зчіпних характеристик трактора з шарнірно зчленованою рамою в залежності від величини баласту та наявності здвоєних коліс. Проведене числове рішення отриманої математичної моделі дозволило отримати графічні залежності зміни основних тягово-зчіпних характеристик трактора ХТЗ-240К в залежності від маси трактора, яка змінювалася від 8600 кг до 10100 кг шляхом зміни маси баласту. Максимальна тягова потужність колісного трактора ХТЗ-240К на одинарних колесах без баласту становить 121 кВт, що досягається за швидкості 12 км·г⁻¹, а на одинарних колесах із баластом N_{kr} =122 кВт – при швидкості 15 км·г⁻¹.

INTRODUCTION

When designing and determining the potential technical capabilities and characteristics of the tractor, the so-called traction calculation is carried out *(Artiomov et al., 2021; Blundell and Harty, 2004)*. As investigations have shown, high traction and fuel economic indicators of the tractor can be reached only in the case of an optimal ratio of its main parameters, which are initially determined by calculation at the design stage *(Cviklovic et al., 2021; Artiomov et al., 2021; Bulgakov et al., 2016)*. Their interrelation, to a greater extent, characterizes the main operational qualities of tractors *(Adamchuk et al., 2016; Bulgakov et al., 2020; Bulgakov et al., 2022)*.

The analysis of some works showed that for the traction calculation of the tractor the following basic parameters are necessary: the total and the structural mass; the maximum operating power of the engine; the transmission gear ratios and the estimated gear speeds *(Adamchuk et al., 2016; Blundell and Harty, 2004; Bulgakov et al., 2019)*.

Vibrations of the elements of the machine and tractor aggregates lead to a deterioration in the fulfilment of the agrotechnical requirements because of nonstraightness of the trajectory of movement, over-compaction of the soil, etc. To reduce the vibrations of tractors and agricultural machines, the following measures are justified: the use of elastic elements in the transmission of the tractors; an active hitch with a pneumohydraulic elastic element; elastic fastening of the working bodies of the agricultural machines; installation of damping elements between the frames; the use of dual / triple wheels; ballasting, etc. With an increase in the weight of the tractor, the contact area of the wheel (caterpillar) with the field surface should also increase in such a way as to reduce the pressure on the soil or keep it at an acceptable level. For technological adaptation, reduction of vibrations of the elements of the wheeled tractors and increasing their productivity, it is recommended to install a ballast.

Ballasting of tractors is carried out mainly by installing removable ballast weights in front of the tractor frame on a special bracket and on the disks of the rear wheels, as well as an additional set of front and rear wheels, or only the rear ones. Sometimes on some models (Terrion 5280, Terrion 7360), weights are also installed on the front wheel disks *(Cviklovic et al., 2021; Artiomov et al., 2021; Ivanovs et al., 2020; Beloev et al., 2015)*.

Accordingly, when designing a tractor, the maximum value of the operating weight should be chosen in such a way that, when the tractor is operating on unpeeled stubble of cereal crops at the lowest operating speed and the rated traction load, the slippage of the movers does not exceed the allowed rate, which is set in the regulatory documents. This rate has the following permitted limits: 18% – for the tractors with a wheel configuration 4K2; 16% – for the tractors with a wheel configuration 4K4 and 5% – for the caterpillar tractors. Tractor ballasting in order to improve the traction characteristics was studied in *(Galych et al., 2021; Galych, 2018; Ivanovs at al., 2018)*. However, the existing design features of tractors require a more accurate determination of their traction properties, depending on the size of the ballast.

The aim of the work is to study regularities of changes in the traction and coupling characteristics of a tractor with an articulated frame depending on the size of the ballast, the presence of dual wheels and other parameters, using the KhTZ-242K tractor as an example.

MATERIALS AND METHODS

During the conducted experimental studies of the traction dynamics of the tractor, vibrations of the tractor half-frames along three axes were studied depending on the speed of the movement and load, taking into account ballasting of the rear half-frame of the tractor and two sets of the wheel systems (single and dual tires). The KhTZ-242K tractor was chosen as the object of research (Fig. 1).

Fig. 1 - Object of experimental research (tractor KhTZ-242K)

The tractor during the research was equipped with an inertial measuring device (IVP-1) *6*, which was installed on the first half-frame of the tractor; inertial measuring device (IVP-2) *7* which was installed on the second half-frame of the tractor; the wheel dynamics sensors *8*, which are located in the centre of rotation; electronic dynamometer *9*. Communication between the sensors, inertial measuring devices and the computing module takes place via CAN bus *5*. The measuring system was powered by a gel lead-acid AGM battery. The data of the research results are stored on the storage medium in the computing module *1*, which is controlled by console 2. A GPS *3* navigation receiver is connected to the module. Location of sensors on the tractor is shown in Fig. 2.

The dynamic sensors *1, 2, 7, 8* were installed on the axes of rotation of the wheels. The vibration of the first half-frame of the tractor was determined by IVP-1 *4*, respectively, for the second half-frame IVP-2 *6*. These sensors cannot be located in the centres of mass of the tractor half-frames *3* and *9*; therefore, the methodology for the compensation of the error in the location of the sensors, described in (*Galych et al., 2021*, *Galych, 2018)*, was used, which determined the actual accelerations in the centres of mass of the tractor half-frames. The navigation receiver antenna *5* is located on the tractor cab symmetrically to the longitudinal axis. The tractor is equipped with ballast *10* weighing 1500 kg, which is located on the second half-frame of the tractor. The traction force was measured by an electronic dynamometer *11*.

In experimental investigations there was used the measuring system VSDEMM for fixing the dynamics and energy of mobile machines (*Artiomov et al., 2021; Galych et al., 2021*). This measuring system relates to the technical means of operational control and diagnostics, and it is designed to determine the kinematic, dynamic, power and energy characteristics of the mobile machines and their elements during the road, field and bench tests (*Galych, 2018*).
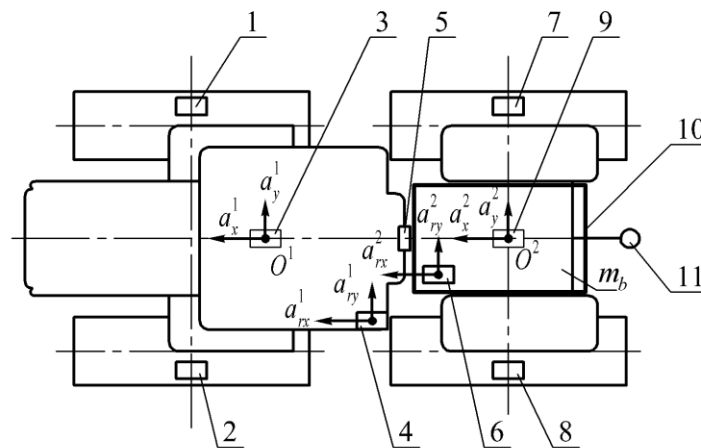

Fig. 2 - Layout of sensors on the tractor
1, 2, 7, 8 – the wheel dynamics sensors (front left, front right, rear left, rear right);
3 – centre of gravity of the first half-frame of the tractor; 4 – IVP-1; 5 – the GPS receiver antenna; 6 – IVP-2;
9 – centre of gravity of the second half-frame of the tractor; 10 – the –ballast; 11 – the traction force sensor

To carry out theoretical studies, a diagram of a tractor with an articulated frame, equipped with a ballast, was drawn up (Fig. 3).
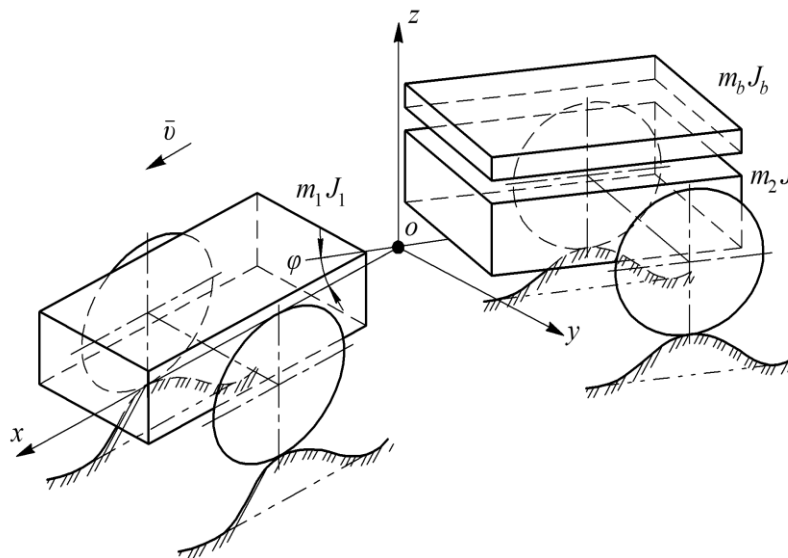

**Fig. 3 - Scheme of a tractor with an articulated frame, equipped with a
ballast on the rear half-frame**
m_b – weight of the ballast; J_b – the reduced moment of inertia of the ballast

When compiling a mathematical model of the effect of the ballast upon the traction-scene properties, the tractor traction balance equation was used, which, with steady motion on a horizontal surface, has the form *(Werner, 2012; Yahya et al., 2004; Bulgakov et al., 2020; Bulgakov et al., 2019)*:

$$P_k = P_{kr} + P_f \text{ , N,} \tag{1}$$

where P_k – the tangential traction force on the rim of the driving wheel;

P_{kr} – the traction force on the hook of the tractor;

P_f – the rolling resistance force.

The traction indicators of the tractor are determined for the following modes of operation: from the nominal – $M_{d.nom}$ up to the maximum torque $M_{d.max}$ for the range of the working gears (*Adamchuk et al., 2016*).

The tangential traction force is determined:

$$P_k = M_d \cdot U_{tr} \frac{\eta_{tr}}{r_{kd}} \text{ , N,} \tag{2}$$

where U_{tr} – the transmission ratio;

η_{tr} – coefficient of efficiency of the transmission;

r_{kd} – the dynamic radius of the wheel, m.

However, we use a model of the wheel that takes into account the dependence on pressure and speed; then the dependence of the rolling resistance force will take the form:

$$P_f = \left(\frac{P}{P_0} \right)^\alpha \left(\frac{P_Z}{P_{Z0}} \right)^\beta P_{Z0} \cdot \left(A + B|v| + Cv^2 \right) , \tag{3}$$

where P, P_0 – the actual and the nominal pressure in the tire;

P_Z, P_{Z0} – the actual and the nominal load onto the wheel;

α, β, A, B, C – the approximating coefficients.

The rolling resistance force is determined through the sum of the rolling resistances for each of the wheels of the tractor separately, taking into account the pressure in the tire, the load onto the wheel and the travel speed:

$$P_f = \sum_{i=1}^{2} \sum_{j=1}^{2} \left(\left(\frac{P_{ij}}{P_{0ij}} \right)^\alpha \left(\frac{P_{Zij}}{P_{Z0ij}} \right)^\beta P_{Z0ij} \cdot \left(A + B|v_{Cij}| + Cv_{Cij}^2 \right) \right) , \text{N} \tag{4}$$

Taking into account (1), the formula for determining the tangential traction force (2) and the rolling resistance force (3), the traction force on the tractor hook is determined from the expression:

$$P_{kr} = M_d \cdot U_{tr} \frac{\eta_{tr}}{r_{kd}} - \sum_{i=1}^{2} \sum_{j=1}^{2} \left(\left(\frac{P_{ij}}{P_{0ij}} \right)^\alpha \left(\frac{P_{Zij}}{P_{Z0ij}} \right)^\beta P_{Z0ij} \cdot \left(A + B|v_{Cij}| + Cv_{Cij}^2 \right) \right) , \text{N} \tag{5}$$

The theoretical speed of movement is determined (without considering the slipping of the driving wheels):

$$v_t = \omega_d \cdot \frac{r_{kd}}{U_{tr}} \text{ , m·s}^{-1}. \tag{6}$$

The actual speed of movement takes into consideration slipping of the driving wheels:

$$v_d = v_t (1 - \delta) \text{ , m·s}^{-1}, \tag{7}$$

where δ – slipping of driving wheels.

The hook power (the power on the hook) is determined from the expression:

$$N_{kr} = P_{kr} \cdot v_d \text{ , W,} \tag{8}$$

or in an expanded form:

$$N_{\mathrm{kr}} = \left(M_d \cdot U_{\mathrm{tr}} \frac{\eta_{\mathrm{tr}}}{r_{\mathrm{kd}}} - \sum_{i=1}^{2} \sum_{j=1}^{2} \left(\left(\frac{P_{ij}}{P_{0ij}} \right)^{\alpha} \left(\frac{P_{Zij}}{P_{Z0ij}} \right)^{\beta} P_{Z0ij} \right) \cdot \left(A + B \left| v_{Cij} \right| + C v_{Cij}^2 \right) \right) \times$$

$$\times \omega_d \cdot \frac{r_{kd}}{U_{tr}} (1 - \delta), \ \mathrm{W}.$$

(9)

Equation (9), obtained as a result of theoretical studies, in contrast to the known ones, makes it possible to simultaneously determine the influence of the type of movers, air pressure in the tires and ballasting upon the theoretical traction characteristic of the tractor, and the fuel consumption.

RESULTS

Using the results of the theoretical research and methodology (*Galych et al., 2021; Bulgakov et al., 2022)*, a numerical solution of the mathematical model (9) was carried out. Due to ballasting the mass of the tractor during its numerical modelling varied from 8600 kg to 10100 kg (tractor KhTZ-240K). As a result, the dependence of slipping of tractor's driving wheels upon the mass m_t of the tractor and tractive effort was obtained:

$$\delta = -459.9 + 0.1415 m_t - 0.00178 P_{\mathrm{kr}} - 1.459 \cdot 10^{-5} m_t^2 +$$
$$+ 1.391 \cdot 10^{-7} m_t P_{\mathrm{kr}} + 2.342 \cdot 10^{-7} P_{kr}^2 + 5.058 \cdot 10^{-10} m_t^3 +$$
$$+ 6.69 \cdot 10^{-12} m_t^2 P_{\mathrm{kr}} - 2.4466 \cdot 10^{-11} m_t P_{\mathrm{kr}}^2 + 3.811 \cdot 10^{-12} P_{\mathrm{kr}}^3.$$

(10)

A graphical display of dependence (10) of slipping of tractor's driving wheels upon the mass of the tractor and the tractive effort is shown in Fig. 4.
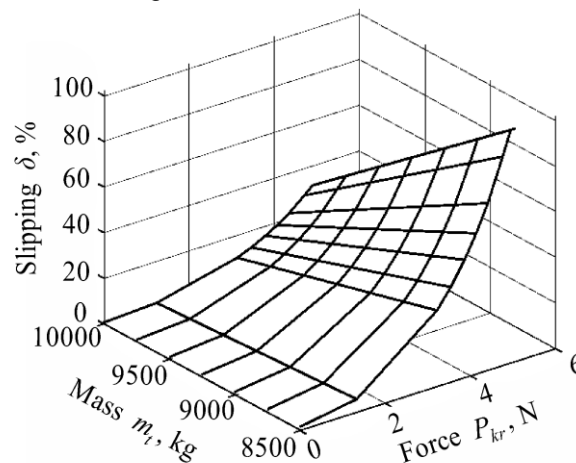

Fig. 4 - Dependence of slipping of the driving wheels δ of the tractors of the KhTZ-240K series upon mass m_t of the tractor and tractive effort P_{kr}

The analysis of the obtained dependence showed that an increase in the weight of the tractor by 1500 kg leads to a decrease in the wheel slip of the tractor from 5.9 to 1.8% at P_{kr} = 13000 N, from 33.5 to 10.3% at P_{kr} = 32000 N, and from 41.8 to 15.9% at P_{kr} = 35300 N (Fig. 4), etc. ballasting leads to the improvement of the traction class tractor.

Next, dependence of the rolling resistance force of the tractor wheel on the load onto the wheel G_k and the speed of movement were determined, which has the form:

$$P_{\mathrm{kf}} = 1255 + 739.7 v + 60.08 G_k + 171 v^2 + 35.42 v G_k - 0{,}0505 G_k^2 -$$
$$- 3.588 \cdot 10^{-14} v^3 + 8.186 v^2 G_k - 0.026 v G_k^2 + 0.00085 G_k^3.$$

(11)

A graphical representation of the dependence of the rolling resistance of the tractor wheel on the load onto the wheel and the speed of movement is shown in Fig. 5.
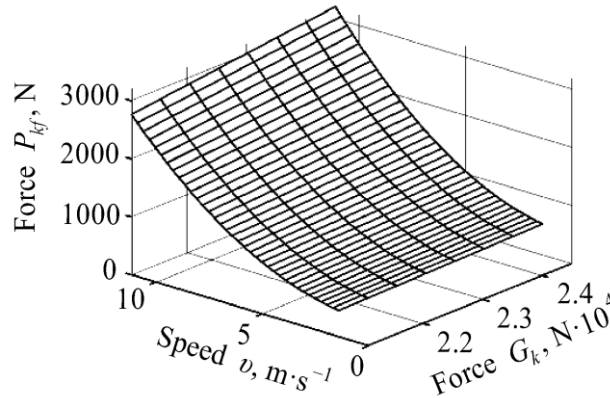
Fig. 5 - Dependence of the rolling resistance force G_k of the tractor wheel on the load onto the wheel and the speed of movement v

The analysis shows that the lowest rolling resistance force of the tractor wheel P_{kf} = 458 N is observed for the speed of movement v = 1.38 m·s⁻¹ and the load on the wheel G_k = 21070 N (Fig. 3). When the load onto the wheel is increased to 24500 N, P_{kf} increases to 530 N. At the speed of movement v = 2.8 m·s⁻¹ – P_{kf} = 568 N, and it increases to 657 N when the tractor is ballasted. The maximum value of the rolling resistance force of the tractor wheel $P_{kf\,max}$ = 2764 N at v = 11.1 m·s⁻¹ and G_k = 21070 N; and increases to $P_{kf\,max}$ = 3199 N (for a ballasted tractor).

With further solution of the obtained dependence (9), there was established a dependence of the power that is spent for rolling the tractor wheel from the load onto the wheel G_k and the speed of movement v, which has the form:

$$N_{\text{kf}} = 7842 + 8250v + 375.5G_k + 3206v^2 + 437.4vG_k - 0.3917G_k^2 +$$
$$+ 494.2v^3 + 153.5v^2G_k - 0.3242vG_k^2 + 0.006646G_k^3. \tag{12}$$

A graphical representation of the dependence of the power that is spent on rolling the tractor wheel is shown in Fig. 6.
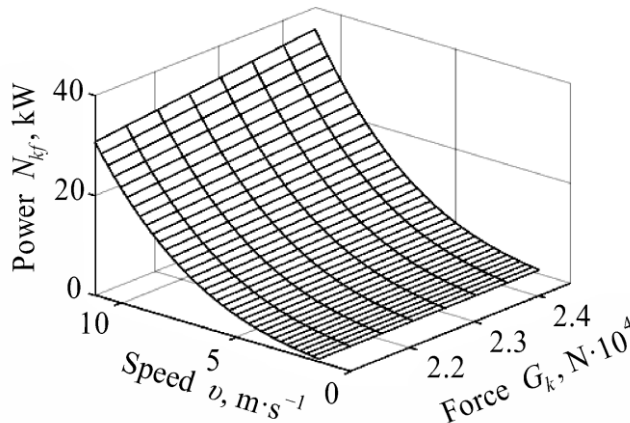

Fig. 6 - Dependence of the power which is spent for rolling the tractor wheel on the load onto the wheel G_k and the speed of movement v

After examination of Fig. 8 we can say that the minimum power that is spent for rolling the tractor wheel is N_{kf} = 0.63 kW for the speed of movement v = 1.38 m·s⁻¹ and the load on the wheel G_k = 21070N (Fig. 8), and it increases up to N_{kf} = 0.91 kW with an increase in load onto the wheel up to 24500 N. For the operating speed of the sowing aggregate v = 2.8 m·s⁻¹, the power is N_{kf} = 1.51 kW (for a tractor without a ballast) and N_{kf} = 1.82 kW (for a tractor with a ballast). At the maximum travel speed of 11.11 m·s⁻¹, for a tractor without a ballast, 30.7 kW is spent for the wheel rolling, and for a tractor with a ballast, 35.5 kW.

The dependence of the tractor rolling resistance force on mass m_t and the speed of movement v, has the form:

$$P_f = 5019 + 2959v + 204.3G_k + 683.8v^2 + 141.7vG_k - 0.2023G_k^2 -$$
$$-6.346 \cdot 10^{-15}v^3 + 32.74v^2G_k - 0.105vG_k^2 + 0.003432G_k^3. \tag{13}$$

A graphical representation of the dependence of the tractor's rolling resistance force on mass m_t and the speed of movement is shown in Fig. 9. Analysing the obtained dependence (9), it can be seen that the minimum value of the tractor rolling resistance force is P_f = 1832 N at v = 5 km·h⁻¹ and m_t = 8600 kg. At the operating speed v = 10 km·h⁻¹, the rolling resistance force of the tractor is P_f = 2272 N (at m_t = 8600 kg), and an increase in the tractor weight by 1500 kg increases the rolling resistance to 2630 N. The maximum value of the tractor rolling resistance force is $P_{f\,max}$ = 11056 N (at m_t = 8600 kg) and $P_{f\,max}$ = 12798 N (at m_t = 10100 kg) (Fig. 7).
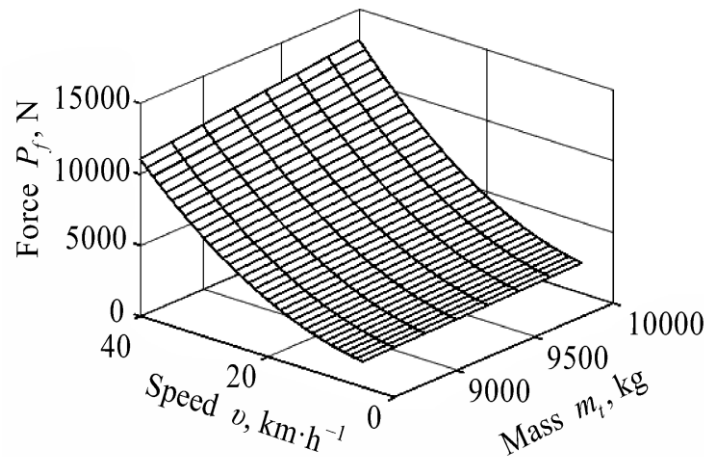

Fig. 7 - Dependence of the tractor rolling resistance force on mass m_t and the speed of movement v

Next, the dependence of the power that is spent for rolling the tractor was determined, which has the form:

$$N_f = 3.137 \cdot 10^4 + 3.3 \cdot 10^4 v + 1502G_k + 1.283 \cdot 10^4 v^2 + 1750vG_k -$$
$$-1.567G_k^2 + 1977v^3 + 614.1v^2G_k - 1.297vG_k^2 + 0.02658G_k^3. \tag{14}$$

A graphical display of the dependence of the power that is spent for rolling the tractor on the mass m_t and the speed of movement v is shown in Fig.8.
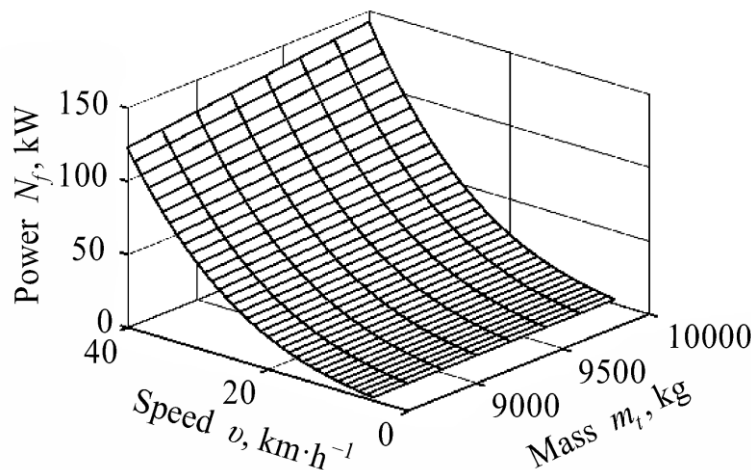

Fig. 8 - Dependence of the power which is spent for rolling the tractor on the load onto the wheel G_k and the speed of movement v

The analysis shows that the power that is spent for rolling the tractor at a speed of v = 10 km·h⁻¹ and m_t = 8600 kg is equal to N_f = 2.2 kW. It increases to N_f = 2.6 kW with an increase in the tractor weight by

1500 kg (Fig. 10). The minimum power that is spent for rolling the power tool without a ballast is $N_{f\ min}$ = 1.8 kW. Ballasting raises this value to N_f = 2.1 kW. The maximum value of the power that is spent for rolling the power tool without a ballast is N_f = 110 kW, with a ballast − N_f = 127.9 kW. There was determined the dependence of the tractive power of the KhTZ-242K tractor on the mass of the ballast and the traction force on the hook (Fig. 9).
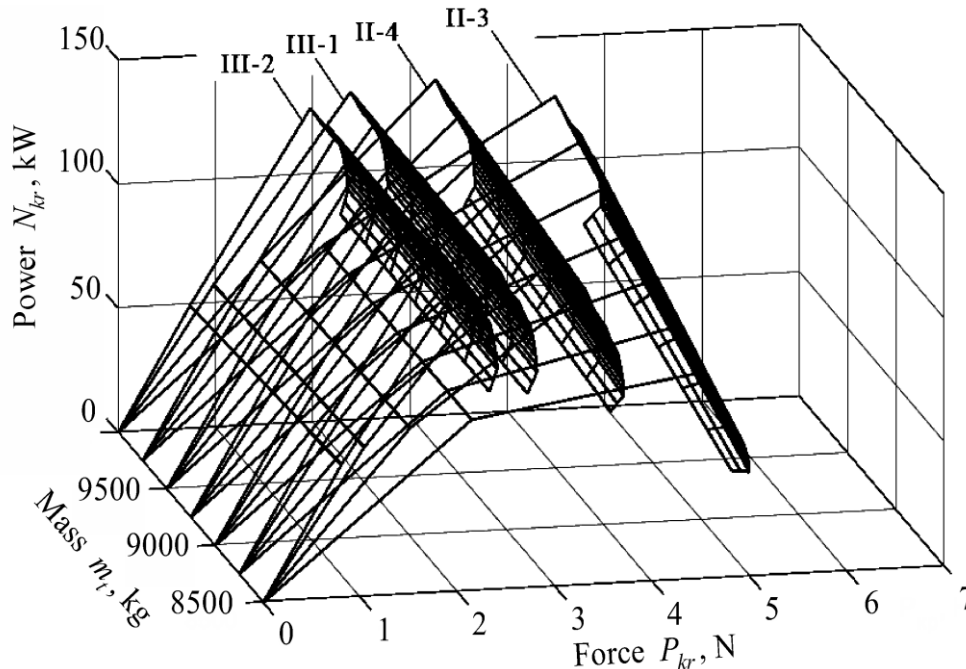

Fig. 9 - Dependence of the tractive power of the KhTZ-242K tractor on the mass of the ballast and the traction force on the hook

The analysis of Fig. 11 shows that the traction power N_{kr} = 113 kW is developed by the tractor without a ballast in the III-2 gear with the traction force on the hook P = 20698 N, and the installation of the ballast leads to an increase in the traction power N_{kr} = 115 kW with a decrease in the traction force on the hook to 19755 N. Accordingly, for transmission III −1 − N_{kr} = 118 kW, P = 24770 N for a tractor without a ballast and N_{kr} = 121 kW, P = 23927 N with a ballast. The maximum tractive power of the tractor is achieved in the II-4 gear and is N_{kr} =120 kW at P=33179 N, and with the ballast installed N_{kr} = 122 kW at P = 32677 N. In the II-3 gear the installation of the ballast increases the traction power by 31% from 82 kW to 108 kW.

Thus, the maximum tractive power of the tractor on single wheels without a ballast 121 kW is achieved at a speed of 12 km·h⁻¹; on single wheels with a ballast N_{kr} = 122 kW – at speed v = 15 km·h⁻¹. The lowest value of slippage was observed for the KhTZ-242K tractors on dual wheels with a ballast installed. However, at the traction forces over 60 kW, slippage of the gearbox clutch was observed during testing. The highest value of slippage is observed for tractors in standard setting on single wheels without a ballast. To confirm the obtained theoretical results, experimental studies were conducted, and the traction power of the KhTZ-242K tractor was determined (Fig. 10).

The analysis of the graphical results of experimental studies showed that for a KhTZ-242K wheeled tractor on single wheels without a ballast, the maximum traction power N_{kr} =121 kW is achieved at a translational speed $v = 12$ km·h⁻¹, the traction efficiency η_t =0.68, and traction force on the hook P = 30.2 kN.

The maximum traction power of 122 kW for a tractor on single wheels with a ballast (Fig. 10, b) is achieved at a speed of 15 km·h⁻¹, a traction efficiency of 0.68, and a traction force on the hook of 29 kN. The maximum allowed value of slipping of the tractor wheels is achieved with a traction force on the hook of 45 kN. The tractor KhTZ-242K on dual wheels without a ballast develops maximum tractive power of 121 kW at a speed of 14.5 km·h⁻¹, traction efficiency of 0.68 and traction force on the hook of 32 kN. However, 15% slippage of the tractor wheels is achieved with a traction force on the hook of 42 kN.
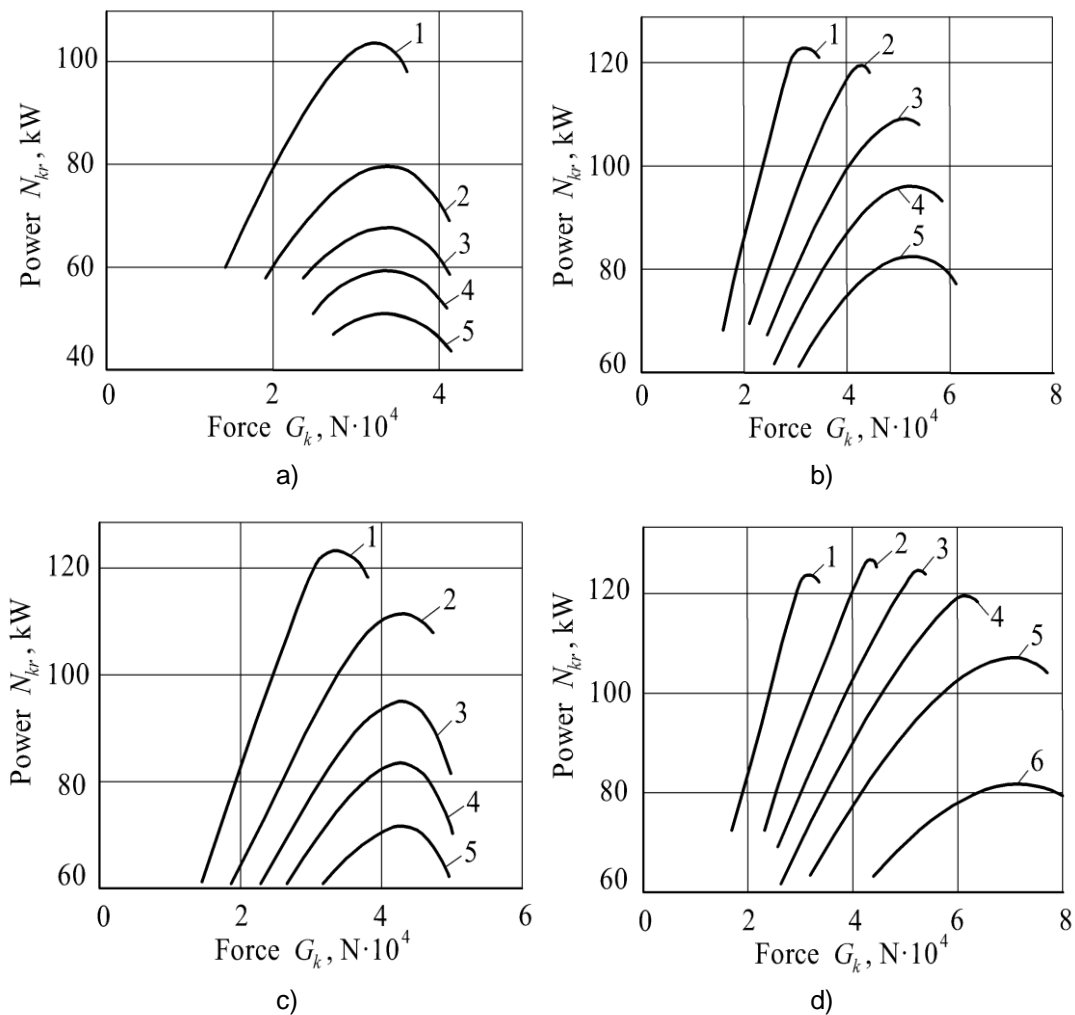
Fig. 10 - Traction power of the KhTZ-242K tractor
a – single wheels without a ballast; b – single wheels with a ballast; c – twin wheels without a ballast;
g – dual wheels with a ballast; on gears: 1 – II-4; 2 – II-3; 3 – II-2; 4 – II-1; 5 – I-4; 6 – I-3

The use of twin wheels and a ballast allows the tractor to develop a maximum traction power of 126 kW at a speed of 11...15 km·h^{-1}, a traction efficiency of 0.72 and a hook pull of 35...45 kN. The maximum value of the wheel slip according to agrotechnical requirements of 15% is achieved with a traction force on the hook of 60 kN. However, it is not recommended to operate the tractor with pulling forces over 60 kN, as the reliability of its units is reduced. At tractive forces over 45 kN, the tractors of the KhTZ-240K series without a ballast with single or dual wheel systems are prone to galloping.

CONCLUSIONS

When conducting theoretical research, regularities were obtained for changing the traction and coupling characteristics of a tractor with an articulated frame, depending on the size of the ballast and the presence of dual wheels.

The numerical solution of the deduced mathematical model made it possible to obtain graphical dependences of the change in the main traction and coupling characteristics of the tractor depending on the mass of the tractor, which varied from 8600 kg to 10100 kg by changing the ballast mass.

The maximum tractive power of the tractor on single wheels without a ballast is 121 kW, which is achieved at a speed of 12 km·h^{-1}, and on single wheels with a ballast N_{kr} = 122 kW, at a speed of v = 15 km·h^{-1}. Which, in its turn, greatly affects the productivity and fuel efficiency of the aggregate.

As a result of the analysis of the experimental data of the traction tests, it is recommended to equip the KhTZ-240K tractor with a ballast, weighing 1500 kg, to reduce slipping. For light soils and the spring work the twin wheel systems are recommended.

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