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DYNAMIC POTENTIAL OF PASSABLENESS OF THE AGRICULTURAL TRACTION-TRANSPORT TECHNOLOGICAL MACHINE WITH A HYDRODRIVE OF WHEELS

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Summary. In article the technique and results of research of traction-dynamic characteristics of a harvester with a hydrodrive of wheels is stated. In the papers the methodology of study the agriculture machine with hydrostatical wheel as well as the estimation of its drawing properties. The stress distribution of drive wheels, wheel slip and changes of machine weight was presented. The computing project of agriculture machine was realized which contain: working pressure in driving set, wheel slip, drawing force, velocity and power of engine.

Key words: agriculture machine, hydrostatical drive, stress distribution, drawing and dynamical characteristics.

INTRODUCTION

Creation of any agricultural harvester is interfaced to the rational coordination of work of all internal subsystems taking into account real interaction of the machine with an environment. In this connection it is necessary to consider in a complex working process of a harvester, considering influence of loadings of its technological modules on indicators of operational properties.

Technological forward semihinged modules of harvesters include undermining passive or active working bodies, bodies scraps, crushing, rejection, removal of a tops of vegetable and others, in case of machines for cleaning of root crops: a potato, carrots, onions, a beet, or - harvesters at grain-harvesting machines and machines for cutting and crushing of the plants, separating and transporting mechanisms [1, 2, 3, 5, 6, 8, 10]. Power resistance from the working bodies co-operating with soil and plants [3, 4, 7, 9, 11], makes essential impact on traction dynamics of all machine. Besides, in the course of work the machine changes the weight, at bunker filling, that also influences properties of the machine.

Wide penetration a hydrovolume drive on modern agricultural harvesters is caused by its essential advantages, in comparison with machines with step mechanical transmissions. However traction-dynamic properties of such machines differ from machines with step mechanical transmissions and are less studied [12, 13, 14, 15].

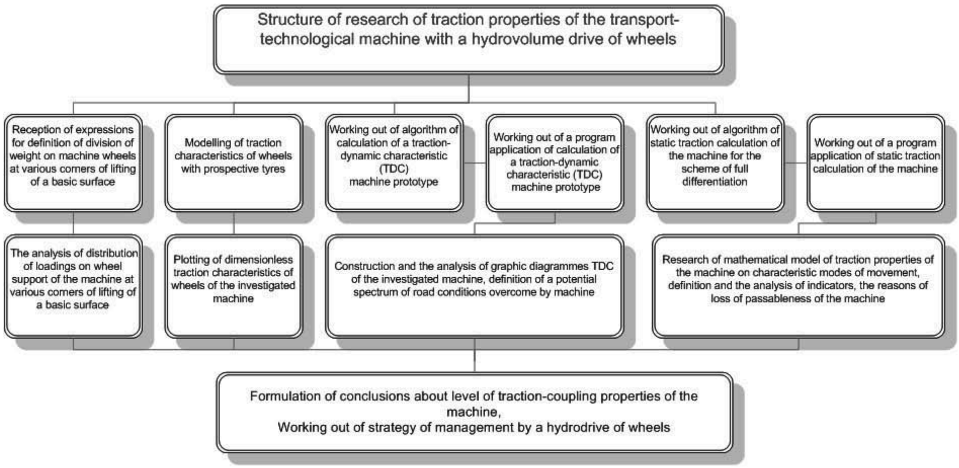

Fig. 1. Algorithm of research of traction-dynamic potential of the machine

In connection with considerable change of normal loadings on wheels of the harvester having the memory bunker, in the course of cleaning, moving across the field and on distant transport represents scientific and practical interest research of traction characteristics of a harvester with a hydrovolume drive of wheels. On the basic operating modes of such machines the drive of wheels from one pump under the scheme 4К4 with full differentiation is used.

The purpose of given article consists in definition of traction-dynamic characteristics of a harvester taking into account specificity of distribution of normal loadings on wheels of leading bridges and their valid slipping, as makes difference of this work from the known.

The block diagramme of algorithm of research of traction-dynamic properties of a harvester with a hydrovolume drive of wheels according to an object in view is resulted in figure 1.

DISTRIBUTION OF NORMAL LOADINGS TO MACHINE WHEELS

Tjagovo-coupling properties of a harvester are in many respects defined by normal loadings to a basic surface on wheels. Distribution of these loadings depends on a number of factors: the design-layout scheme of running system; masso-geometrical parametres: positions of the centre of weights, the moments of inertia concerning axes of cross-section and longitudinal-angular fluctuations; characteristics of rigidity and geometrical parametres of tyres; deformation characteristics and characteristics of microprofiles of a basic surface on trajectories of movement of wheels of different boards and harvester bridges; a mode of movement of the machine; influences of the technological module aggregated with machine at working and transport positions [7, 11, 13].

More often the harvester has statically definable running system with a shaking beam of the bridge of operated wheels. Expressions of reactions in support of wheels of the machine which settlement scheme is shown in figure 2 are for such a case received.

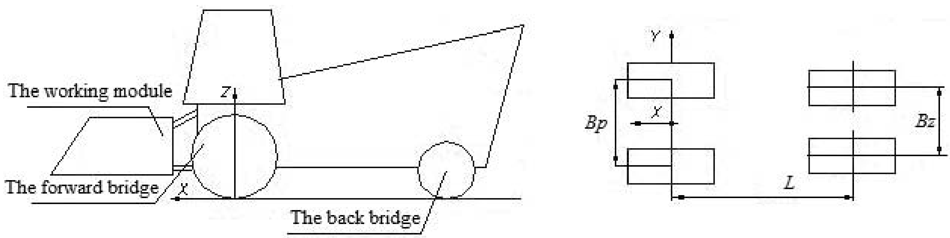

Fig. 2. The scheme of a transport technological harvester

The example of a copy of the screen of the monitor with calculation listing in a program application of symbolical mathematics with initial data of one of possible variants of a harvester and settlement expressions depending on a corner of lifting of basic surface α is resulted in figure 3.

By results of calculation schedules of dependences of distribution of normal loadings on wheels of a harvester for various conditions of the memory bunker and position of the technological module (figures 4-6) are constructed.

In figures 4-6 among schedules the curve only for one of wheels of the back bridge of operated wheels, as normal loadings on a wheel of other board same is shown.

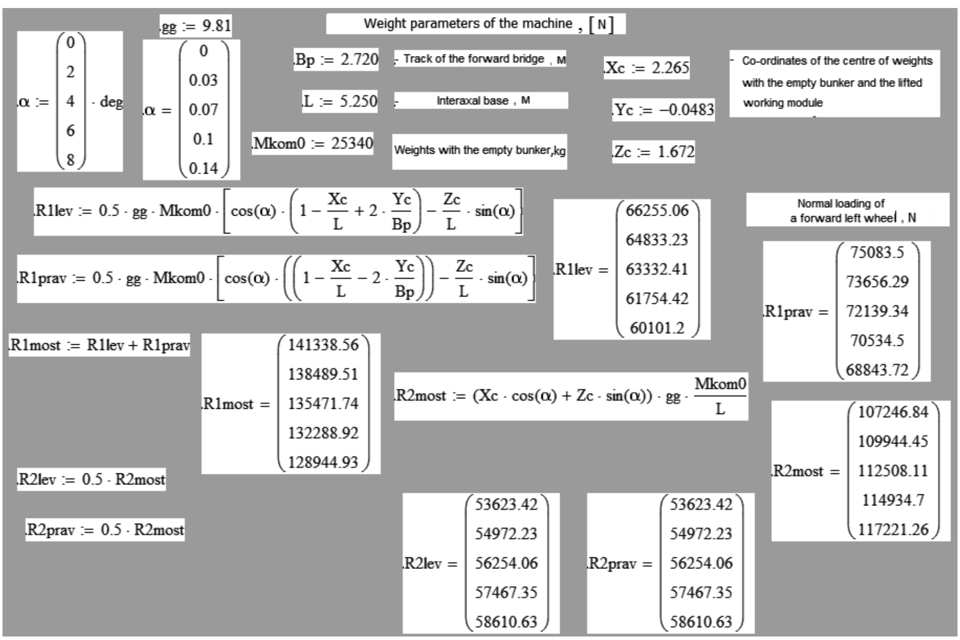

Fig. 3. Listing of calculation of normal loadings on harvester wheels

Apparently from figures 4-6, wheels of running system are exposed at work to considerable change of loadings, that essentially affects traction efforts developed by them and traction properties of the machine as a whole.

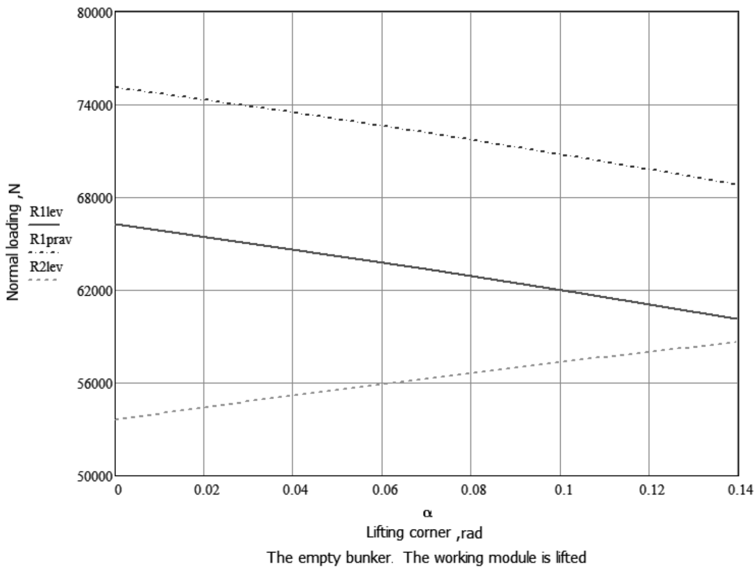

Fig. 4. Normal loadings on harvester wheels at the empty bunker and the lifted forward working module

As follows from figures 4-6, with growth of a corner of lifting of a surface of movement of a wheel of the back bridge of operated wheels are loaded in addition also their normal loading approaches with similar loadings of forward wheels, both at the empty bunker, and at full, but in case of the lifted forward technological module.

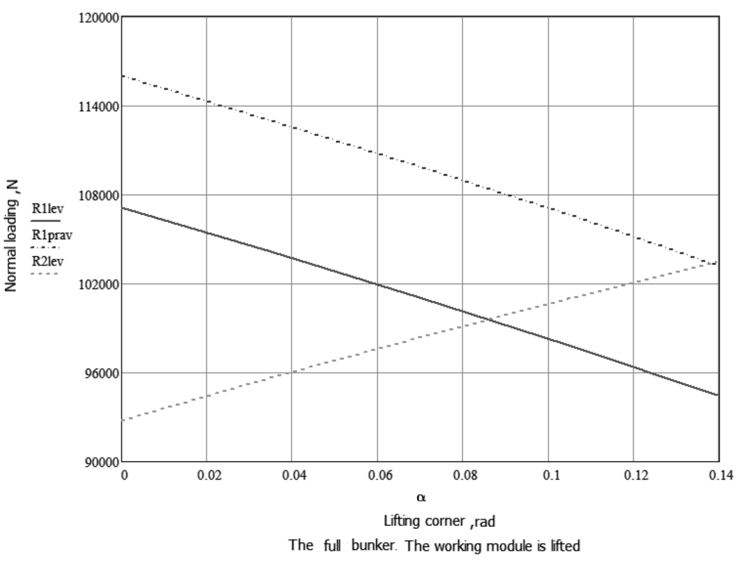

Fig. 5. Normal loadings on harvester wheels at the full bunker and the lifted forward module

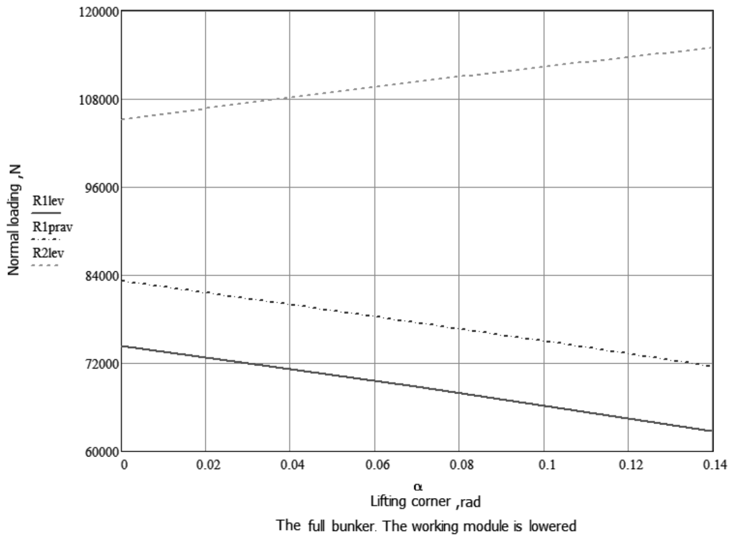

Fig. 6. Normal loadings on harvester wheels at the full bunker and the lowered forward module

At lowering of the forward module and its interaction with soil loading essentially increases by back wheels. With growth of a corner of lifting of a surface weeding this redistribution of loadings it is aggravated. Various loadings on wheels of the left and right board are caused by discrepancy of the centre of weights of the machine with is longitudinal a vertical plane of symmetry of running system of the machine that is connected with feature of placing of the process equipment on a considered variant of a harvester.

TRACTION-DYNAMIC POTENTIAL OF THE HARVESTER

Passableness of the harvest wheel machine is a difficult technical quality which is shown in its work to destination in interaction with environment through properties of passableness which characterise the ability of the machine caused by set of systems of its design and their interrelations, to carry out reliable safe overcoming of the set limited working space of the movement which are in a certain condition of structure (figure 7). In this figure the most widespread measuring instruments of properties of passableness are resulted.

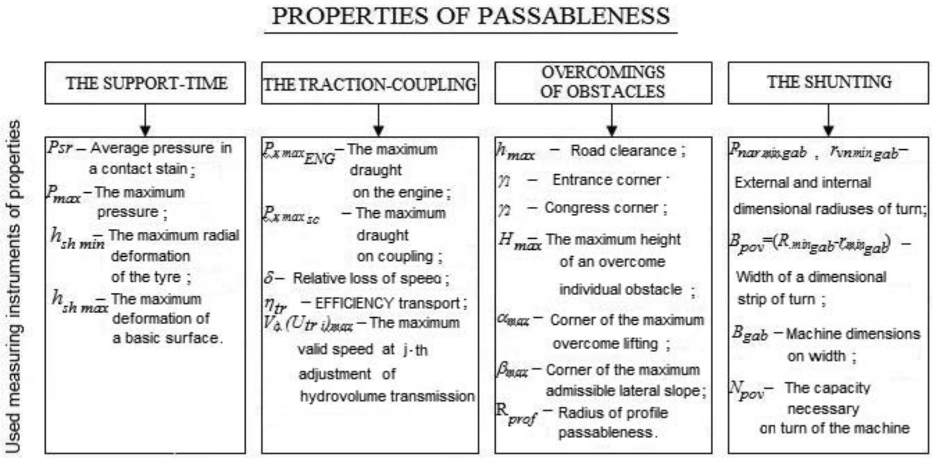

Fig. 7. Structure and most often used measuring instruments of properties of passableness

At the lifted forward working module of the machine its settlement scheme is resulted in figure 8.

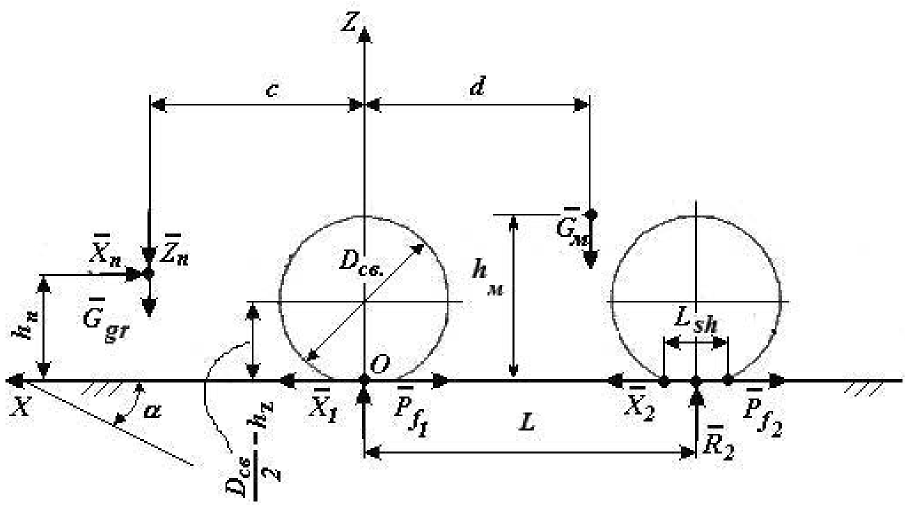

Fig. 8. the Settlement scheme of a harvester in transport position
for a complex estimation of passableness

Ggr - weight lifted in transport position the forward technological module;

G_M - harvester weight;

L_{sh} - length of a stain-contact of a wheel with a ground

The equations of static balance of a harvester at the established speed of movement look like:

$$\left.\begin{aligned}&\sum X = 0: -X_n + X_1 - R_1 \cdot f_1 + X_2 - R_2 \cdot f_2 = 0,\\&\sum Z = 0: -Z_n + G_{gr} + R_1 - G_M + R_2 = 0,\\&\sum M_O = 0: \left(Z_n + G_{gr}\right) \cdot C - X_n \cdot h_{\Pi = n} - G_M \cdot d + R_2(a+b) = 0.\end{aligned}\right\} \tag{1}$$

At the lifted forward technological module and maintenance of absence of a kinematic mismatch of district speeds of wheels of leading bridges we can write down:

$\varphi_1 = \varphi_2 = \varphi$ - operating ratio of chain weight of leading bridges of the machine;

$\delta_1 = \delta_2 = \delta$ - slipping of driving wheels of the machine;

$$\left.\begin{aligned}&X_1 = R_1 \cdot \phi;\ X_2 = R_2 \cdot \phi;\ X_n = 0;\ Z_n = 0,\ then\ P_{f_1} = R_1 \cdot f_1;\ P_{f2} = R_2 \cdot f_2,\\&\phi = \frac{R_1 \cdot f_1 + R_2 \cdot f_2}{R_1 + R_2};\quad R_2 = \frac{G_M \cdot d - G_{gr} \cdot C}{L},\\&R_1 = G_M + G_{gr} - R_2;\quad \varphi = \varphi_{\max}\left(1 - e^{-\kappa_{sh} \cdot \delta}\right),\end{aligned}\right\} \tag{2}$$

whence

$$\delta = -\frac{1}{\kappa_{sh}} \cdot \ln\left(\varphi_{\max} - \varphi\right). \tag{3}$$

The indicator of potential of passableness looks like [4]:

$$P = \sqrt[3]{P_{sc} \cdot P_{op} \cdot P_N},$$

where: P_{sc} - a passableness indicator on coupling,

P_{op} - an indicator of basic passableness,

P_N - A passableness indicator on the engine capacity N_n,

$$P_{sc} = 1 - \frac{P_\kappa}{P_{\kappa\max}};\ P_{op} = 1 - \frac{P_{sr}}{\left[P_{sr}\right]_{dop}};\ P_N = 1 - \frac{N}{N_n}.\quad P_{op} = 1 - \frac{P_{sr}}{\left[P_{sr}\right]_{dop}};\ P_N = 1 - \frac{N}{N_n}. \tag{4}$$

$$\left.\begin{aligned}&P_\kappa = \left(R_1 + R_2\right) \cdot \phi;\ P_{\kappa\max} = \left(R_1 + R_2\right) \cdot \varphi_{sc};\quad P_{sc} = 1 - \frac{\varphi}{\varphi_{sc}},\\&h_{z_i} = \frac{R_i}{2} \cdot \frac{1}{\pi \cdot p_{w_i} \cdot \sqrt{B_{sh_i} \cdot D_{cb_i}}};\quad L_{sh_i} = \sqrt{\frac{D_{cb}^2}{4} - h_{z_i}^2},\\&S_{k_i} = B_{sh_i} \cdot L_{sh_i};\quad P_{sr_i} = \frac{R_i}{2 - S_{k_i}};\ P_{op} = 1 - \frac{P_{sr_{i\max}}}{\left[P_{sr}\right]_{dop}}.\end{aligned}\right\} \tag{5}$$

Taking into account the accepted designations capacity N spent for movement, is equal:

$$N = \left(X_n + R_1 \cdot f_1 \cdot \cos\alpha + R_2 \cdot f_2 \cdot \cos\alpha + G_M \cdot \sin\alpha\right) \cdot V, \tag{6}$$

where: $V = \dfrac{\pi \cdot n_d}{30 \cdot U_{TR_i}} \cdot r_{ri}^o$,

$$\delta = 1 - \frac{V_d}{V}; \quad V = \frac{V_d}{1-\delta}; \quad V_d = V \cdot (1-\delta). \tag{7}$$

The capacity lost on slipping of wheels, is equal: $N_d = N \cdot \delta$,

$$\left.\begin{aligned} & N_n = \frac{\pi \cdot n_{d_{nom}}}{30} \cdot M_{d_{nom}}, \\ & \text{At } X_n = 0; \quad N = N_f + N_\alpha + N_\delta, \\ & \left(N_f + N_\alpha\right) = N \cdot (1-\delta); \quad P_N = 1 - \frac{N}{N_n}, \end{aligned}\right\} \tag{8}$$

Transport EFFICIENCY of the machine: $\eta_{trn} = \dfrac{N_{f_{gr}}}{N}$. (9)

where capacity on overcoming of resistance to movement:

$$N_f = G_{gr}(f \cdot \cos\alpha + \sin\alpha) \cdot V_d, \tag{10}$$

As a result of calculation of all intermediate sizes value P pays off.

For an estimation of traction-dynamic potential of the wheel machine widely use as a measuring instrument dynamic factor D [1]:

$$D = P_k - P_w / G, \tag{11}$$

where: P_k - a total tangent force of wheels of running system; P_w - force of resistance of air; G - machine weight.

Thanks to high general EFFICIENCY modern aksial-piston hydromachines even more often find application in hydrovolume drives of wheel running systems of technological multipurpose machines. For a design stage carrying out of research of traction-coupling properties of such machines is necessary, for the purpose of a choice of rational parametres of a drive of wheels, algorithms of management by hydromachines in characteristic working conditions of running system.

In case of work of hydromotors of a drive of wheels from one pump under the scheme of full differentiation the total traction effort of driving wheels of the machine essentially depends on coupling conditions of separate wheels and resistance to their movement. And this effort is limited by a wheel which is in the most adverse conditions: with the least vertical loading, coupling and the greatest resistance to movement. Such conditions lead to sharp growth of its slipping, decrease in total draught of all wheels and, frequently, to loss of passableness of the machine.

For maintenance of demanded draught the control system of a hydrodrive realises various algorithms. Necessity of check of their efficiency at a design stage demands working out of adequate mathematical model of working process of a hydrovolume drive. Thus it is necessary to consider real traction characteristics of the wheels equipped with pneumatic tyres then the mentioned model will allow to trace continuously true values of pressure and expenses of a working liquid in system, slippings of each of the wheels, the valid speed of the machine, all components of expenses of capacity on movement of the machine and other target sizes depending on parametres of the machine, conditions of movement, position of controls by hydromachines of a hydrovolume drive and the machine engine (figure 1).

By a known technique traction-dynamic characteristics in the beginning have been calculated on the computer (TDC) a harvester with a hydrovolume drive [1] at work with the wheel formula 4K2 and 4K4. Results of calculations are resulted in figures 9,10.

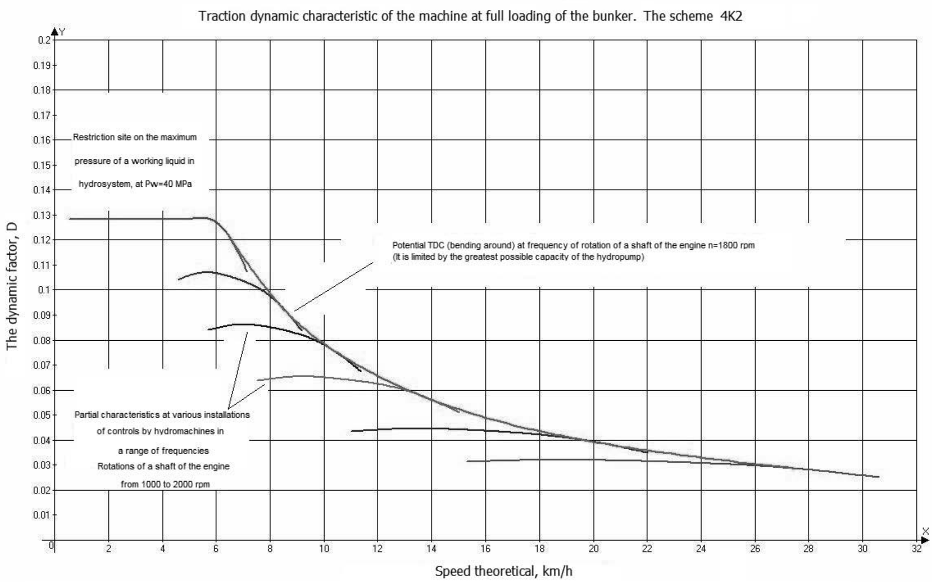

Fig. 9. TDC a harvester at work under the scheme 4K2

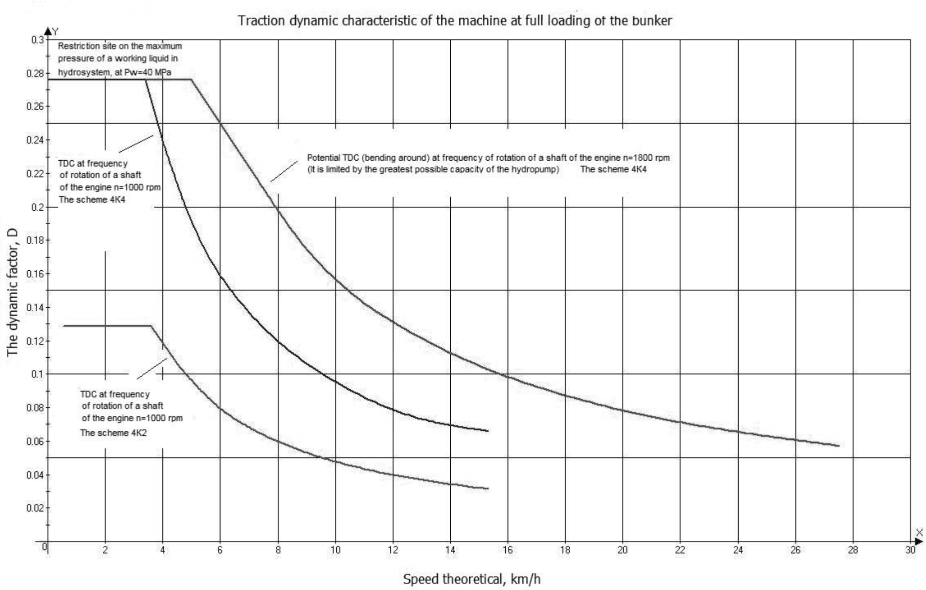

Fig. 10. TDC a harvester at work under the scheme 4K4

As follows from the characteristic, the dynamic factor for the scheme 4K2 at full loading of the bunker does not exceed value 0,13, and at work under the scheme 4K4 - 0,28 and is limited by admissible pressure of a working liquid in pressure head highways of a hydrodrive of hydromotors of a drive of wheels.

On TDC have defined the maximum corners of overcome lifting α_{ij}, being set by factor of resistance to movement of the machine and solving the trigonometrical equation. For concrete j-th point TDC:

$$D_{j\,max} = f_i \cdot cos\ \alpha_{ij} + sin\ \alpha_{ij}, \tag{12}$$

where: f_i - factor of resistance to machine movement on the given site of a surface of movement which is equal to the sum of factor of resistance movement machines and specific, on a unit of weight of the machine, technological resistance $P_{tech.}$ The forward module:

$$f_i = f_{\kappa i} + P_{tech.}/G_{k\,\Sigma}. \tag{13}$$

Unknown values of corners α_{ij} at set $D_{j\ max}$ and f_i can be defined a numerical method, for example in a program application of symbolical mathematics, or to calculate under the formula:

$$\alpha_{ij}. = arcsin\ (D_{j\,max} {*} cos\ (arctg\, f_i)) - arctg\, f_i\,. \tag{14}$$

In figures 11 and 12 schedules of dependences of the maximum corners of lifting of a surface of movement overcome by a harvester are resulted, at work under the scheme 4K4 with the hung out forward module.

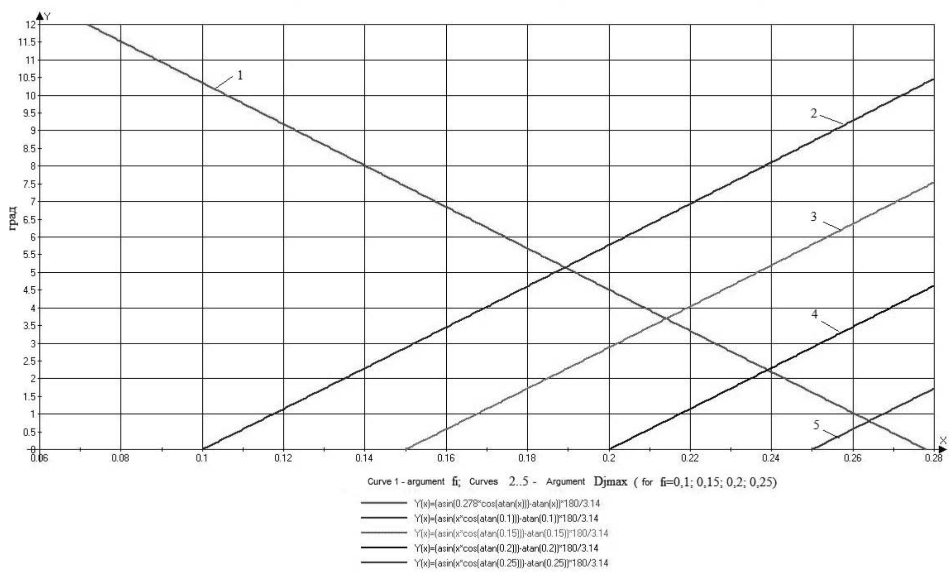

Fig. 11. The diagram for definition of corners of overcome liftings

And the curve 1 reflects dependence $\alpha_{ij}\,(f_i)$ at the maximum value of the dynamic factor for the specified condition of the machine in a spectrum of road conditions, resistance to movement of wheels of the machine on which is not exceeded by values 0,28 that corresponds really meeting. In heavy road conditions the machine is capable to overcome lifting no more than 3,2 degrees. Curves 2. 5 - $\alpha_{ij}(D_j)$ - in figure 11 are received for surfaces of movement with values f_i, accordingly the equal: 0,1; 0,15; 0,2; 0,25, depending on the dynamic factor which the harvester can have, that is to the maximum value - 0,278.

TDC a harvester allows to define values of theoretical speeds of movement, traction efforts so and spent capacity, pressure in pressure head highways of motors, the spectrum on weight of overcome road conditions, and represents result of traction-dynamic calculation. The loading traction characteristic at a small range of resistance of giving of the technological module is not informative.

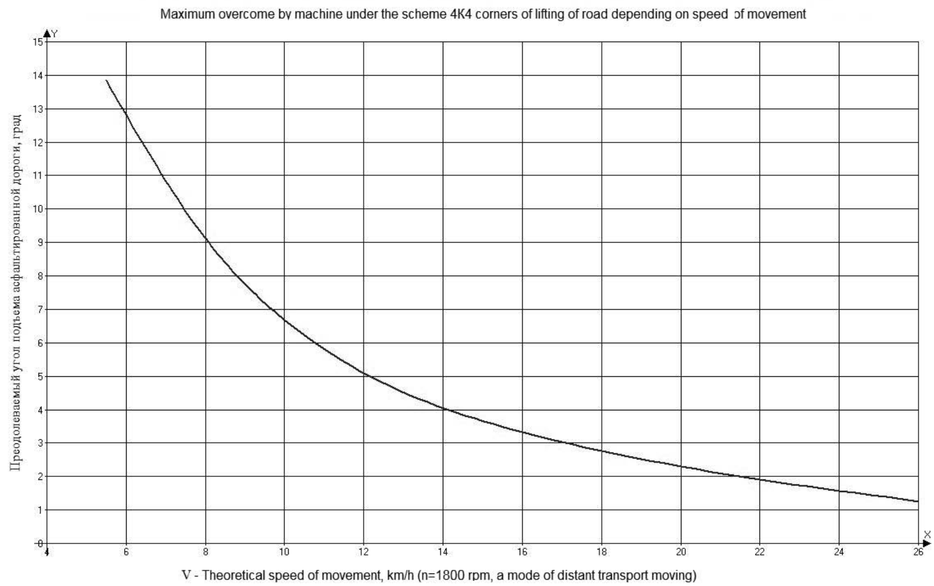

Fig. 12. Influence of theoretical speed on corners of overcome liftings

STATIC TRACTION CALCULATION OF A HARVESTER WITH A HYDROVOLUME DRIVE OF WHEELS

Research of distribution of traction forces on wheels, slippings of wheels and high-speed losses of capacity of the machine for the scheme of full differential communication of pumps with motors and distributions of vertical loadings on wheels on various surfaces of movement makes a subject of static traction calculation of the machine, which has been executed on algorithm specially developed by authors to consider true values of pressure in hydrosystem of a drive and slipping of each of driving wheels.

Taking into account calculated earlier for a studied variant of division of weight of a harvester on wheels and accepted maximum resolved on a drive of wheels of capacity of the engine in 340 kw it is executed on the resulted algorithm (figure 1) the analysis of traction indicators of a harvester. Calculations in the developed program application in a package of symbolical mathematics were spent depending on a corner of lifting of a surface of movement for following five basic regular modes of movement across the field, and also on the asphalted road on distant transport.

Parametres of modes the following: 1st technological mode - a field, n=1400 rpm, the empty bunker, 4К4, the full differentiation, the lowered forward module, volume of hydropumps - maximum, volume of hydromotors - maximum; 2nd technological mode - the same, but with the full bunker; 3rd mode - short technological transport moving across the field, n=1400 rpm, the empty bunker, 4К4, the full differentiation, the lifted forward module, volume of hydropumps - maximum, volume of hydromotors - maximum; 4th mode - the same, as in previous, but with the full bunker; 5th mode - asphalt, n=1800 rpm, the empty bunker, 4К4, the full differentiation, the lifted forward module, distant transport moving, volume of hydropumps - maximum, volume of hydromotors - minimum.

In connection with absence of results of traction tests of tyres of wheels of the machine their traction characteristics have been constructed by calculation by the developed technique for two surfaces of movement: a field and the asphalted road.

Results of static traction calculation of the machine are presented in table 1. As follows from table 1, on coupling of one of driving wheels with a ground the machine reaches restrictions only on the first mode at coal of lifting of a surface of the field, exceeding 6 degrees. On modes 2, 4 and 5 restrictions come at various corners of lifting of a surface of movement or on pressure in a pressure head highway of a hydrodrive, or because of the big machine of the capacity exceeding capacity of the established engine demanded for movement.

At machine work under the scheme of full differentiation each wheel has the values of slipping, angular and theoretical speed. In this connection restriction of the maximum corner of overcome lifting on coupling owing to insufficiency of total draught wheels in each concrete case will be defined by one of wheels system which the first will reach absolute value of slipping, as it is visible from table 1.

Table 1. Results of traction calculation of a harvester

Lif-ting cor-ner, degr.	Pressure in hydro-system of a drive of wheels, kPa	Relative loss of speed of a wheel the forward left wheel	Relative loss of speed of a wheel the forward right wheel	Relative loss of speed of a wheel, back left/ rights wheels	Total draught of wheels, кN	Speed real-ity, km/h	Power engine on a drive of wheels, kW
1 mode							
0	17999,224	0,262	0,21	0,186	53,36	5,387	158,228
2	20332,403	0,319	0,24	0,198	60,277	5,19	178,633
4	22644,750	0,407	0,28	0,21	67,132	4,898	197,929
6	24933,449	0,598	0,33	0,223	73,917	4,222	207,732
8	27195,712	1,0	0,41	0,237	80,624	0	231,436
2 mode							
0	29399,051	0,197	0,182	0,188	87,156	5,532	250,186
2	33533,686	0,228	0,206	0,201	99,413	5,398	285,372
4	37631,406	0,267	0,235	0,214	111,561	5,238	320,244

6	41687,219	0,321	0,273	0,227	123,585	5,034	354,759
8	45696,184	0,405	0,325	0,241	135,470	4,737	388,875
3 mode							
0	17689,627	0,153	0,144	0,209	52,442	5,579	150,539
2	20485,029	0,169	0,157	0,229	60,729	5,461	174,328
4	23255,472	0,187	0,171	0,249	68,943	5,335	197,904
6	25997,583	0,207	0,187	0,27	77,072	5,198	221,24
8	28708,019	0,232	0,206	0,292	85,107	5,047	244,306
4 mode							
0	29082,473	0,154	0,148	0,202	86,217	5,598	247,492
2	33678,228	0,171	0,163	0,219	99,842	5,485	286,602
4	38232,951	0,191	0,18	0,236	113,345	5,365	325,363
6	42741,093	0,214	0,199	0,254	126,709	5,233	363,727
8	47197,162	0,242	0,223	0,272	139,920	5,087	401,649
5 mode							
0	17319,848	0,047	0,046	0,066	19,070	22,831	189,504
2	24864,972	0,053	0,05	0,075	27,377	22,667	272,059
4	32379,802	0,059	0,056	0,083	35,652	22,496	354,282
6	39855,182	0,065	0,061	0,092	43,882	22,315	436,074
8	47282,005	0,073	0,067	0,100	52,060	22,123	517,334

CONCLUSIONS

Developed and realised as information technology the technique of a design estimation of tjagovo-dynamic potential of a harvester with a hydrovolume drive of wheels allows to accept correctly its rational parametres and to investigate traction properties of a wide class of similar machines taking into account the valid slipping of driving wheels on a movement surface.

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METODYKA OCENY WŁAŚCIWOŚCI UCIĄGOWYCH MASZYNY ROLNICZEJ Z HYDROSTATYCZNYM NAPĘDEM KÓŁ JEZDNYCH

Streszczenie. W publikacji przedstawiono metodykę badań i oceny właściwości uciągowych oraz uciągowo-dynamicznych charakterystyk maszyny rolniczej z hydrostatycznym napędem kół jezdnych. Przy opracowaniu metodyki uwzględniono specyfikę rozkładu normalnych obciążeń na koła mostów napędowych, ich rzeczywistego poślizgu i możliwości weryfikacji algorytmów sterowania napędem kół przy zmiennej masie maszyny. Przedstawiono projekt obliczeniowy maszyny rolniczej obejmujący: ciśnienie robocze w hydrostatycznym układzie napędowym, indywidualny poślizg kół napędowych, siłę uciągu, prędkość roboczą i moc silnika przeznaczoną na napęd kół.

Słowa kluczowe: maszyna rolnicza, napęd hydrostatyczny, obciążenia normalne, charakterystyki uciągowe i uciągowo-dynamiczne.