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DETERMINATION OF THE LIFTING CAPACITY RESERVE FOR AGRICULTURAL TRACTOR'S LIFTING MOUNTED DEVICE

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Introduction

In the energy aspect the possibility of aggregating mounted working machines and tools with a lifting mounted device (LMD) of agricultural tractor is determined by the LMD lifting capacity.

The concept of the lifting capacity of a wheel tractor LMD is given in [1] and is determined by the lifting weight (or mass m) of the mounted implement (MI) at the maximum developed force value ($F_{\mathrm{r}}^{\max}$) on the rod of the hydraulic cylinder of the linkage mechanism (LM):

$$mg = \frac{F_{\mathrm{r}}^{\max}\eta_{\mathrm{LM}}}{I_S}, \tag{1}$$

where g is the acceleration of gravity; η_{LM} and I_S – efficiency and gear ratio of linkage mechanism.

It should be noted that in this expression: the friction losses in the hinges of the LM are determined from the test results, and in the process of changing the LM generalized coordinate is assumed constant; the influence of inertia forces is not taken into account; the gear ratio of LM is determined from the velocity plan as the ratio of the vertical component of the velocity of the center of gravity of MI to the piston speed of the hydraulic cylinder (HC). In accordance with the standard [2], the maximum weight of the load lifted by the LMD should be measured, increasing the load to the maximum weight that the LMD is capable of lifting throughout the LMD stroke at the maximum hydraulic pressure. As a result the determination of the lifting capacity of the agricultural tractor is relatively approximately and time consuming.

Objective: to develop a formalized description of the lifting capacity of the lifting mounted device, supplemented by the calculation of its reserve, calculated in the computer-aided design mode.

Formulation of the problem

LMD is an indispensable element of the machine-tractor unit, designed to communicate with the agricultural tractor with MI. LMD consists of a hydraulic drive (HD) and a linkage mechanism. A scheme of the LMD hydraulic drive, the hydraulic cylinder of which is loaded from the side of the linkage mechanism is shown in Fig. 1.

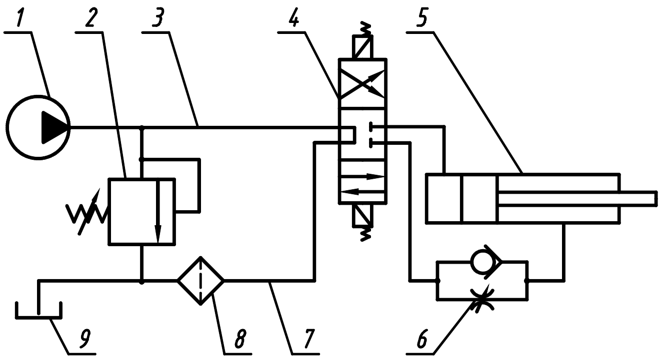

Fig. 1. Scheme of the hydraulic drive of lifting mounted device:
1 – gear pump; *2* – safety valve; *3* – pressure line; *4* – hydro-distributor; *5* – hydraulic cylinder;
6 – adjustable throttle; *7* – drain line; *8* – filter; *9* – tank

The magnitude of the load reduced to the piston of the HC F_{R} consists of the useful component F , as well as the reduced forces of friction $F_{\mathrm{fr}}^{\mathrm{R}}$ and inertia $F_{\mathrm{i}}^{\mathrm{R}}$:

$$F_{\mathrm{R}} = F + F_{\mathrm{fr}}^{\mathrm{R}} + F_{\mathrm{i}}^{\mathrm{R}}. \tag{2}$$

The useful load on the hydraulic cylinder is proportional to the gear ratio of the LM I_S [3]:

$$F = P_6 I_S. \tag{3}$$

The maximum driving force $(F_{\mathrm{r}}^{\max})$ developed on the HC piston to overcome the load given to the HC is determined by the expression:

$$F_{\mathrm{r}}^{\max} = p_{\mathrm{HC}}^{\max} F_{\mathrm{HC}}, \tag{4}$$

where F_{HC} – the area of the piston HC; $p_{\mathrm{HC}}^{\max}$ – maximum pressure in HC. The maximum pressure in HC is limited by the setting of the safety valve and the pressure losses in the hydraulic drive:

$$p_{\mathrm{HC}}^{\max} = p_{\mathrm{sv}} - (\Delta p_{\mathrm{thr}} + \Delta p_{\mathrm{prl}}), \tag{5}$$

where p_{sv} – pressure setting of the safety valve; Δp_{thr} – pressure loss at the throttle in the drain line; Δp_{prl} – pressure loss in the pressure line.

A mathematical model of dynamic analysis [4] was formed in the form of a system of differential equations:

$$\begin{cases} \dot{p}_1 = \dfrac{E_{\mathrm{R}}}{V_0 + F_{\mathrm{HC}}(S - S_0)} Q - \dfrac{F_{\mathrm{HC}} E_{\mathrm{R}}}{V_0 + F_{\mathrm{HC}}(S - S_0)} \dot{S}; \\ p_2 = p_1 - \left(a_1 \ddot{S} + a_2 \dot{S} + a_3 \dot{S}^2\right); \\ m_{\mathrm{R}} \ddot{S} + \dfrac{1}{2} m'_{\mathrm{R}} \dot{S}^2 = p_2 F_{\mathrm{HC}} - \left[F + F_{\mathrm{fr}}^{\mathrm{R}}\right], \end{cases} \tag{6}$$

where E_{R} – the reduced modulus of the bulk elasticity of the working fluid; V_0 – the initial volume of working fluid in the pressure line; S, S_0 – current and initial values of the generalized coordinate; a_1, a_2, a_3 – coefficients proportional to various types of pressure losses; m_R, m'_R – reduced mass and its derivative with respect to the generalized coordinate; Q – volumetric flow rate of the working fluid.

As a result of solving the system (6) by a numerical method (for example, Runge–Kutt of the 4th order), the law of motion of the HC piston is determined – $S(t) = f(S_0, \dot{S}, \ddot{S}, t)$, and also changes of pressure at pump – p_1 and in the HC cavity, defined from the pressure line – p_2.

The linkage mechanism is the main component of the LMD, which determines the nature of the interaction of the tractor with the MI. This is a lever mechanism hinged to the frame of the tractor. Links of the LM through the connecting triangle associated it with MI, taken as the output link. Tractor frame, links of the LM and MI altogether form a closed kinematic chain. The calculation of the output parameters of an LM is carried out on the basis of its flat analogue, obtained from the 3d geometric model by projecting the centers of the hinges of the mechanism – onto its longitudinal plane of symmetry (Fig. 2).

As a result of the structural analysis of a flat kinematic chain, we have a single-motion eight-link mechanism [5], the change in the generalized coordinate of which (S) is uniquely related to the position of its output link (L_6).

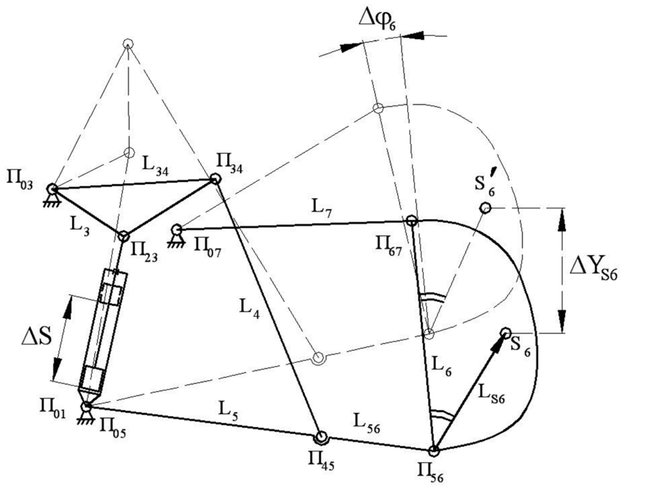

Fig. 2. Flat analogue of the linkage mechanism of an agricultural tractor (scheme of lifting mounted implement from working to transport position)

Geometrical analysis of a closed kinematic chain is performed by the method of closed vector contours and is considered in detail in [5]. As a result of the geometric analysis the coordinates of the movable LM hinges and the characteristic points of the closed kinematic chain are determined. In particular, the coordinates of the axis of suspension LM – are determined by the expressions:

$$X_{56}(S) = X_{05} + L_{56} \cos\varphi_5(S); \quad Y_{56}(S) = Y_{05} + L_{56} \sin\varphi_5(S), \tag{7}$$

where X_{05}, Y_{05} – the coordinates of the stationary hinge $П_{05}$ on the frame of the tractor; φ_i – the angle formed by the corresponding link in the right Cartesian coordinate system.

The coordinates of the characteristic point – the center of gravity of the MI (for example – working tool) are determined in accordance with the expressions:

$$X_{S6}(S) = X_{56}(S) + L_{S6}\cos[\varphi_6(S) + \varphi_{S6}]; \tag{8}$$

$$Y_{S6}(S) = Y_{56}(S) + L_{S6}\sin[\varphi_6(S) + \varphi_{S6}], \tag{9}$$

where L_{S6} and φ_{S6} are the characteristics of the vector drawn from the axis of suspension to the center of gravity of the working implement.

The gear ratio LM is an analogue of the vertical velocity of the center of gravity MI [3], depending only on the internal parameters of the LM and MI:

$$I_{S6} = \varphi_3' U_{53}\left[L_{56}\cos\varphi_5 + U_{65} L_{S6}\cos(\varphi_6 + \varphi_{S6})\right], \tag{10}$$

where φ_3' – analogue of angular velocity of link L_3; U_{53} and U_{65} – the transfer ratios, which connect the angular velocities of the links L_{56} and L_3 and also L_{56} and L_6.

In accordance with the established design practice, there are two output kinematic parameters LM – gear ratios on the axis of suspension I_m and gear ratio at a distance of 610 mm from the axis of suspension I_{610}. The gear ratio LM on the axis of suspension is determined under the assumption that the center of gravity of the MI is located on the axis of suspension of the LM and it is determined by the first component in expression (10).

The calculation of the gear ratios of the LM of "Belarus 1523" tractor was performed in accordance with the analytical expressions obtained and practically coincided with the results obtained by the graph-analytical method. Diagrams of change of LM gear ratios presented in Fig. 3.

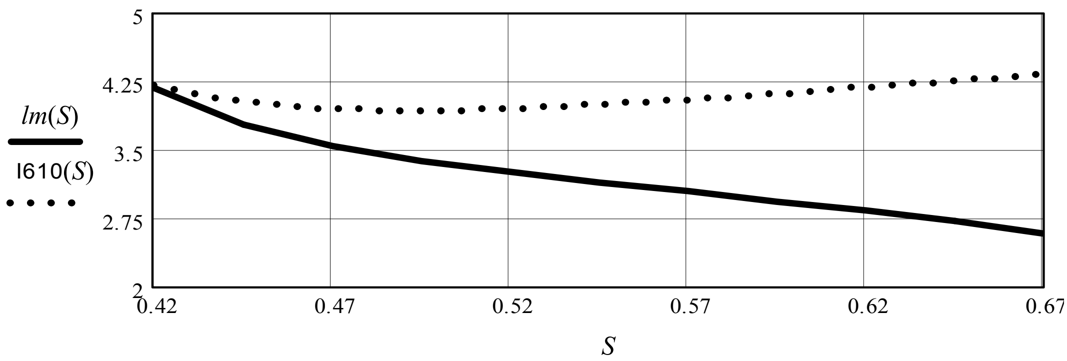

Fig. 3. Dependencies of gear ratios LM on the suspension axis (solid line) and at a distance of 610 mm from it (dashed line) from the generalized coordinate *S*

The formalization of the description of force analysis of LM consists in determining the forces acting in the hinges of links, and is carried out in Assur groups in the reverse order, according to the well-known method [5]. It does not take into account the weight of links LM and its inertia forces arising in the process of movement of links. The friction force reduced to the hydraulic cylinder piston is calculated on the basis of the fact that it is equal to the ratio of dividing the sum of the instantaneous friction powers expended in the hinges of LM by the piston velocity HC plus the piston cuff friction towards the HC sleeve (F_{frhc}):

$$F_{\text{fr}}^{\text{R}}(S) = F_{\text{frhc}} + rf_{\text{fr}}\left\{\sum_{i=1}^{7} R_{0i}(S)\varphi_i'(S) + \sum R_{ij}(S)\left[\varphi_i'(S) \pm \varphi_{i+1}'(S)\right]\right\}, \tag{11}$$

where $r-$ is the radius of the hinge; f_{fr} – friction coefficient; $R_{0i}(S), R_{ij}(S)$ – forces acting respectively in fixed and movable hinges LM; $\varphi_i', \varphi_{i+1}'$ – analogues of angular velocities of LM links.

To simplify the expression (11) we assume the hinge radii and friction coefficients are the same for all kinematic pairs. The friction force of the piston cuff on the inner surface of the HC sleeve is determined by the expression derived from [6]:

$$F_{\text{frhc}} = \pi D l f_{\text{cfr}} p_m, \tag{12}$$

where D – piston diameter; l – cuff width; f_{cfr} – the coefficient of friction of the cuff about the sleeve HC; p_m – average pressure in the pressure cavity of HC.

Analysis of expressions (11) and (12) shows that the friction losses in the hinges do not depend on the piston velocity and are determined only by the internal parameters of the LM and HC.

The expansion of the train of agricultural machines and working tools, aggregated with the tractor accompanied by the growth of their mass-geometric characteristics, conflicts with the limited power of the hydraulic drive of the LMD. That is why for the modernization of LMD in the mode of computer-aided design, a more accurate description of its lifting capacity is necessary. With this aim it is proposed to clarify the expression (1). Add expressions (2)–(5) and (10) to (1) and perform the transformations. As a result the analytical expression for the lifting capacity of the LMD takes the form:

$$G_{S6} = \frac{p_{\text{HC}}^{\max} F_{\text{HC}} - \left[F_{\text{fr}}^{\text{R}} + F_{\text{i}}^{\text{R}}\right]}{I_S}. \tag{13}$$

In this expression the lifting capacity is the weight of the MI transferred from the working to the transport position, measured in newtons.

As follows from the expression (13), the lifting capacity of the LMD is an integral indicator depending simultaneously on the parameters of the hydraulic drive, the linkage mechanism and the mass-geometric characteristics of the MI. It should also be noted that the liftng capacity of the LMD is inversely proportional to its gear ratio and changes as the MI rises. In this case the maximum value of gear ratio limits the weight of the MI, which is possible to be transferred in the transport position.

Calculation of output parameters of the LMD

On the basis of the clarified expression for determining the lifting capacity (13), the lifting capacity of the "Belarus 1523" tractor LMD was determined on the hitch axis, as well as at a distance of 610 mm and 950 mm from it. The results of the calculation of the output parameters of the LMD of the "Belarus 1523" tractor aggregated with the "Polesie-3000" mounted forage harvester, performed with the help of the generated functional mathematical model (FMM), are presented in Tabl. 1 and 2.

Table 1

Geometric and kinematic output parameters LM

	Geometric parameters				Kinematic parameters				
S, m	$Y_{56}(S)$, m	$X_{S6}(S)$, m	$Y_{S6}(S)$, m	$\varphi_6(S)$, grad	$U_{53}(S)$**	$\varphi_6'(S)$, 1/m	$I_m(S)$	$I_{610}(S)$	$I_{S6}(S)$
0.420	0.300*	–	–	–	–	–	–	–	–
0.445	0.401	1.681	0.515	90.02	0.692	0.401	3.763	4.007	5.009
0.470	0.472	1.702	0.603	90.17	0.689	0.662	3.529	3.932	5.022
0.495	0.562	1.711	0.693	90.93	0.682	0.892	3.370	3.914	5.047
0.520	0.641	1.712	0.784	92.31	0.670	1.121	3.247	3.930	5.073
0.545	0.724	1.701	0.882	94.09	0.654	1.362	3.141	3.969	5.085
0.570	0,791	1.692	0.979	96.21	0.633	1.626	3.039	4.025	5.097
0.595	0,863	1.671	1.061	98.83	0.607	1.919	3.036	4.093	5.109
0.620	0.932	1.643	1.099	100.82	0.574	2.247	2.826	4.167	5.123
0.645	1.067	1.598	1.191	102.19	0.535	2.610	2.705	4.240	5.139
0.670	1.132	1.561	1.283	105.11	0.486	3.003	2.571	4.298	5.167

*The connection of the mounted forage harvester "Polesie-3000" is made when the height of the suspension axis (Y_{56}) is 0.4 m.

**Dimensionless quantity.

Table 2

LMD power parameters

S, m	$G_m(S)$, kN	$G_{610}(S)$, kN	$G_{S6}(S)$, kN	R_{01} (S), kN	$R_{03}(S)$, kN	$R_{05}(S)$, kN	$R_{07}(S)$, kN	$F_R(S)$, kN	$p_2(S)$, MPa
0.420	–	–	–	–	–	–	–	–	–
0.445	46.39	42.63	33.87	114.02	58.45	78.34	60.97	114.02	12.45
0.470	48.41	43.43	34.01	115.0	49.54	76.58	59.01	115.0	12.55
0.495	50.68	43.63	33.77	117.11	48.14	75.05	59.23	117.11	12.78
0.520	52.60	43.46	33.31	119.80	47.59	73.62	59.23	119.80	13.08
0.545	54.38	43.03	32.79	123.22	47.53	72.22	59.53	123.22	13.45
0.570	56.19	42.43	32.32	126.98	47.74	70.76	59.99	126.98	13.86
0.595	58.17	41.73	32.17	131.03	48.05	69.28	60.45	131.03	14.3
0.620	60.43	40.98	32.05	135.12	48.33	67.47	60.75	135.12	14.75
0.645	63.14	40.28	31.93	138.79	48.44	65.57	60.65	138.79	15.15
0.670	66.46	40.04	31.72	141.39	48.28	63.54	59.89	141.39	15.43

The calculation showed that the lifting capacity of the LMD on the suspension axis was 46.39 kN and at a distance of 610 mm and $X_6 =$ 950 mm from the axis of suspension, respectively, 40.04 kN and 31.72 kN. Consequently, in the energy aspect, the aggregation of the "Belarus-1523" tractor with the mounted forage harvester "Polesie-3000" ($P_6 = 28.5$ kN) is quite possible.

From Tabl. 2 it can be seen that the lifting capacity of the LMD is considered to be equal to its minimum value, taken in the range of variation of the generalized coordinate S and simultaneously corresponding to the maximum of gear ratio.

Calculation of lifting capacity reserve

In the first approximation it is possible to compare the moment of load $(M = P_6 X_6)$ relative to the axis of suspension of the LM, created by the original MI with the upgraded MI. In this case the load moment of the upgraded MI should not exceed the initial one:

$$P_6^{\text{mod}} X_6^{\text{mod}} \le P_6^{\text{in}} X_6^{\text{in}}.$$

Considering the initial load moment as a constant value ($P_6^{\text{mod}} X_6^{\text{mod}} = \text{const}$), we determine the lifting capacity LMD as the difference between the minimum value of the lifting capacity in the process of lifting MI and its weight:

$$\Delta G_{S6} = G_{S6}^{\min} - P_6 ; \quad \Delta G_{S6} = 31.72 - 28.5 = 3.22\,\text{kN}.$$

Thus, the lifting capacity reserve of the LMD is directly related to the coordinate X_6. Modernization MI is accompanied by a change in its weight, which entails a change in the position of the center of gravity MI (S_6) relative to the axis of suspension LM (Π_{56}) and gear ratio (I_{S6}). Therefore the lifting capacity of the upgraded LMD is determined taking into account possible changes to the components of the following formula:

$$G_{S6}^{\text{mod}} = \frac{p_{\text{HC}}^{\max} F_{\text{HC}} - \left[F_{\text{fr}}^{\text{R}} + F_{\text{i}}^{\text{R}}\right]}{I_S}.$$

Accordingly, the lifting capacity reserve according to the results of modernization MI is calculated by the formula:

$$\Delta G_{S6}^{\text{mod}} = G_{S6}^{\text{mod}} - P_6^{\text{in}} \frac{P_6^{\text{in}} X_6^{\text{in}}}{P_6^{\text{mod}} X_6^{\text{mod}}}.$$

Conclusion

The given method of calculating the lifting capacity of LMD of an agricultural tractor and its reserve makes it possible to evaluate, in the energy aspect, the possibility of aggregating any MI with other wheeled mobile power units having a structurally identical LMD.

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