THEORETICAL INVESTIGATION OF A REAR-MOUNTED LINKAGE FOR WIDE-SPAN TRACTORS

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Abstract: *The work is devoted to the investigation of the peculiarities of aggregation of wide-span tractors (vehicles) with agricultural machines and implements. As a result of the conducted studies, it has been established that, from the point of view of the control impact (the turning angle of the driven wheels) and the disturbing impact (the turning angle of the mounted implement in a horizontal plane) exerted by the wide-span tractor, the most desirable is such a variant of configuration of the mounted device in which the instantaneous centre of rotation, formed by the convergence rays of its lower links, is in the centre of the mass of the tractor. When using wide-span tractors (vehicles) with mounted implements of the high 4N and 4 categories in accordance with the ISO 730: 2009, one should consider a possibility to configure them both for a three-point and a two-point scheme.*

KEYWORDS: WIDE SPAN TRACTORS (VEHICLES), REAR-MOUNTED LINKAGE, STABILITY AND CONTROLLABILITY, ANGULAR MOBILITY, RESEARCH ON THE PC.

1. Introduction

Wide-span tractors (vehicles) [1, 2] are lately gaining popularity in the world, which allow implementing a controlled traffic farming technology.

Owing to their rather high versatility the wide-span tractors (vehicles) can (and undoubtedly should) be aggregated with mounted, semi-mounted and trailed agricultural machines and implements [1, 2]. At the same time, the wide-span tractor (vehicle), by its layout scheme, is intended for linking of the technological part (with or without a mounted device) in sufficient vicinity to its centre of mass [1]. In such a situation, to ensure that the deviations from the course of the wide-span tractor (vehicle) did not adversely affect the stability of the movement of the mounted aggregated machines and implements, the design of the mounted implement must (to some extent, naturally) be able to independently rotate in a horizontal plane. That is why the issue of justifying the parameters of a wide-span tractor (vehicle) from the standpoint of steady movement of its technological part in a horizontal plane is relevant.

It is known that the three-point adjustment scheme of a rear-mounted mechanism of the conventional tractor can provide sufficient mutual angular turning ability of its framework and the technological part [3]. Considering a possibility to unify the mounted implements of the wide-span tractors (vehicles) with the conventional ones, the parameters of which are determined by the International Standard ISO 730: 2009, it is necessary to establish a correspondence of the sizes and requirements for a three-point mounting, designed for the connection of agricultural machines and implements.

In addition, certain trends have been outlined lately in the improvement of the design of the three-point mounting devices of tractors [4-9]. However, insufficient attention has been paid to the problem of their angular mobility, especially with respect to the wide-span tractors (vehicles) characterised by an untypical layout, the specificity of their use and the operating conditions.

The purpose of the work is to increase the efficiency of operation and use of wide-span tractors (vehicles) by justifying the parameters of their mounted devices.

2. Materials and method

As a physical object of research, we will consider a wide-span vehicle of the authors' design, shown in Fig. 1.

***Fig. 1.** A design of a wide-span vehicle developed in Ukraine*

When making a plane-parallel movement in a horizontal plane, a number of corresponding forces are acting upon the wide-span vehicle and its technological part. For the purpose of mathematical simulation of its plane-parallel movement, we will construct an equivalent calculation scheme for the wide-span vehicle (Fig. 2).
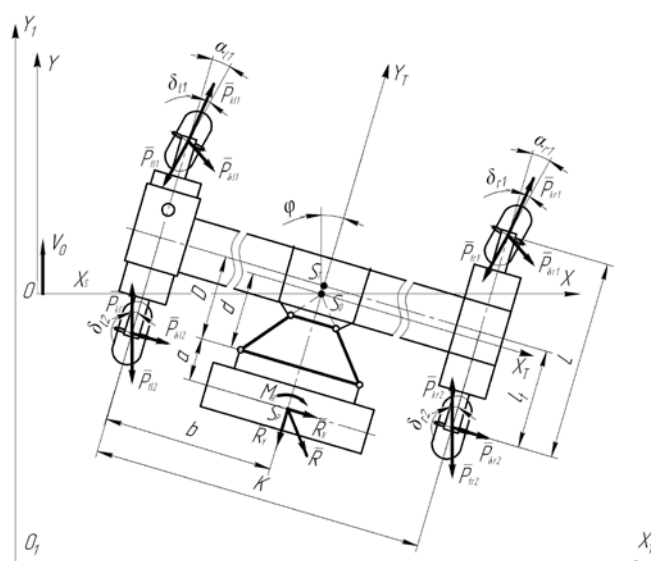

***Fig. 2.** A scheme of forces acting upon the wide-span vehicle in a horizontal plane with a kinematic method of its control*

With respect to the XOY plane the investigated wide-span vehicle with a technological part attached to it has two degrees of

freedom, corresponding to two generalised coordinates: angle φ and abscissa X_S (Fig. 2).

To the number of external forces acting upon the wide-span vehicle in its plane-parallel movement (see Fig. 2) belong:
- the forces of rolling resistance of the frontal (P_{fl1}, P_{fl1}) and the rear $(P_{fl2}, Pfl2)$ wheels;
- the tangential traction forces $P_{kl1}, P_{kr1}, P_{kl2}, P_{kr2}$, on the frontal and rear wheels;
- lateral forces $P_{\delta l1}, P_{\delta l2}, P_{\delta r1}, P_{\delta r2}$, which lead to the appearance of displacement angles of the frontal δ_{l1}, δ_{r1} and the rear δ_{l2}, δ_{r2} wheels;
- the main vector R (components R_X and R_Y, respectively) and the main moment M_R of forces brought to the centre of resistance (point S_p), acting from the side of the agricultural implements.

The value of force R_X and moment M_R will depend on the turning angle β of the lower links of the wide-span vehicle (Fig. 3).

Taking into account the small value of these angles in Fig. 3 we have:

$$R_X = R\left[\varepsilon \pm (1 - c/1)\beta\right],$$
$$M_R = R(d_0 + D - d)\left[\varepsilon \pm (1 - c/1)\beta\right], \qquad (1)$$

where ε – an angle between the direction of the traction force of the tractor and the longitudinal axis of symmetry of its technological part;

β – the turning angle of the lower links of the mounted wide-span vehicle;

d_0 – the distance in a longitudinal direction from the point of application of the hook effort to the instantaneous turning centre of the mounted tractor attachment (Fig. 3);

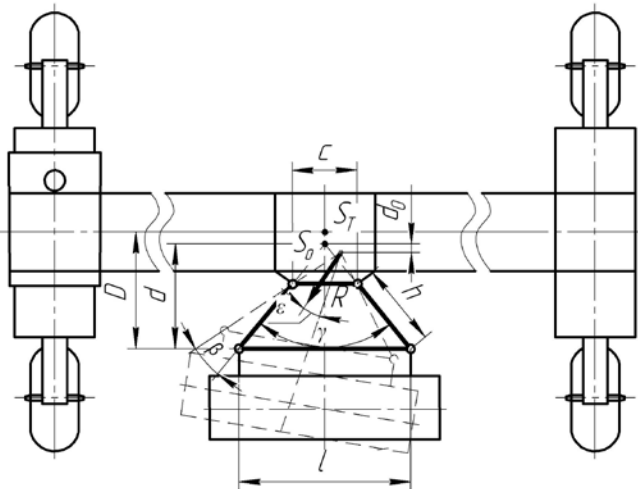c, l – the design parameters of the mounted wide-span vehicle (Fig. 3).

Fig. 3. *A scheme of turning of the lower links of the mounted wide-span vehicle attached at angle β*

In dependencies (1), the choice of the sign from the proposed record "±" is determined by the correspondence of the direction of angles ε and β. So the "-" sign is accepted when the indicated angles of the agricultural implement change in the process of the work of the wide-span vehicle in antiphase. In case of their in-phase change sign "+" is adopted.

3. Results and discussion

The differential equations of the movement of the wide-span vehicle with respect to plane XOY (see Fig. 2) will look like:

$$A_{11}\ddot{X}_S + A_{12}\dot{X}_S + A_{13}\dot{\varphi} + A_{14}\varphi = f_{11}\alpha + f_{12}\beta + f_{13};$$
$$A_{21}\ddot{\varphi} + A_{22}\dot{\varphi} + A_{23}\varphi + A_{24}\dot{X}_S = f_{21}\alpha + f_{22}\beta + f_{23}, \qquad (2)$$

where $A_{11} = M_T + M_R;$

$$A_{12} = \begin{pmatrix} k_{l1} + k_{l2} + k_{r1} + k_{r2} - P_{kl1} + P_{fl1} - P_{kr1} + \\ + P_{fr1} - P_{kl2} + P_{fl2} - P_{kr2} + P_{fr2} \end{pmatrix} / V_0;$$

$$A_{13} = \left[\begin{array}{l}(k_{l1} + k_{r1} - P_{kl1} + P_{fl1} - P_{kr1} + P_{fr1})(L - l_t + D - d) + \\ + (P_{kl2} - P_{fl2} + P_{kr2} - P_{fr2} - k_{l2} - k_{r2})(l_t - D + d)\end{array}\right] / V_0;$$

$A_{14} = -A_{12}V_0;$ $A_{21} = J_{SO};$

$$A_{22} = \left[\begin{array}{l}(k_{l1} + k_{r1} - P_{kl1} + P_{fl1} - P_{kr1} + P_{fr1})(L - l_t + D - d)^2 + \\ + (P_{kl2} - P_{fl2} + P_{kr2} - P_{fr2} - k_{l2} - k_{r2})(l_t - D + d)^2\end{array}\right] / V_0;$$

$A_{23} = -A_{13}V_0;$ $A_{24} = A_{13};$

$f_{11} = k_{l1} + k_{r1};$ $f_{12} = R(1 - c/1);$ $f_{13} = R\varepsilon;$

$f_{21} = (k_{l1} + k_{r1} + P_{kl1} - P_{fl1} + P_{kr1} - P_{fr1})(L - l_t + D - d);$

$f_{22} = R(1 - c/1)(d_0 + D - a - 2d) + (P_{kl1} - P_{fl1} + P_{kl2} - P_{fl2})b -$
$(P_{kr1} - P_{fr1} + P_{kr2} - P_{fr2})(K - b);$

$f_{23} = R(d_0 + D - a - 2d)\varepsilon.$

In the mathematical model (2) the following designations are accepted: M_T, M_P - masses of the tractor and its technological part, respectively; V_0 – the forward speed of the tractor; J_{SO} – the moment of inertia of the tractor with the technological part relative to point S_0; k_{l1}, k_{r1} and k_{l2}, k_{r2} – coefficients of the drag resistance of the frontal and rear wheels of the wide-span vehicle, respectively L, l_t, K, a, b, d, D – the design parameters of the wide-span vehicle the nature of which is clear from Fig. 2 and 3; α_{l1}, α_{r1} – the turning angles of the driven wheels of the tractor at its kinematic method of control.

Applying transformation $d/dt = s$, we will obtain a mathematical model of the plane-parallel movement of the wide-span vehicle in a horizontal plane, recorded in an operator form:

$$\left.\begin{array}{l}K_{11}X_S(s) + K_{12}\varphi(s) = F_{11}\alpha(s) + F_{12}\beta(s) + F_{13}(s); \\ K_{21}X_S(s) + K_{22}\varphi(s) = F_{21}\alpha(s) + F_{22}\beta(s) + F_{23}(s),\end{array}\right\} \qquad (3)$$

where $K_{11} = A_{11}s^2 + A_{12}s;$ $K_{12} = A_{13}s + A_{14};$ $K_{21} = A_{24}s;$

$K_{22} = A_{21}s^2 + A_{22}s + A_{23};$ $F_{11} = f_{11};$ $F_{12} = f_{12};$ $F_{13} = f_{13};$

$F_{21} = f_{21};$ $F_{22} = f_{22};$ $F_{23} = f_{23};$

$\dfrac{d}{dt} = s$ – the operator of differentiation.

The nature of the response of the dynamic system to the control and disturbing impacts is determined by the design and technological parameters of the wide-span vehicle, which enter into the coefficients of the right-hand parts of the system (3). In our case the point of interest is the nature of the response of the control and disturbing impacts of the considered dynamic system at various ratios of the design parameters d and D (see Fig. 3). These parameters determine the distance from the axis that passes through the attachment points of the lower links of the mounted device to the instantaneous turning centre (point S_0) and the centre of the mass (point S_T) of the tractor. Therefore, in the process of mathematical simulation, this ratio of parameters was considered for three cases: $d < D$; $d = D$ and $d > D$.

The output values of the mathematical model (3) are transverse displacements X_S of the wide-span vehicle, and its relative bearing φ.

The controllability of the movement of the wide-span vehicle was estimated by analysing the corresponding transfer functions with the help of the constructed amplitude and phase frequency characteristics.

The transfer function $W^\alpha(s)$, which describes variations in

the relative bearing φ of the wide-span vehicle when the turning angle α of its wheels change, can be presented according to the system of equations (3) as:

$$W^\alpha(s) = \frac{C_{1\alpha}s + C_{0\alpha}}{s\left(B_2 s^2 + B_1 s + B_0\right)}, \qquad (4)$$

where $\quad C_{1\alpha} = A_{11}f_{21}; \quad C_{0\alpha} = A_{12}f_{21} - A_{24}f_{11}; \quad B_2 = A_{11}A_{21};$
$B_1 = A_{12}A_{21} + A_{11}A_{22}; \quad B_0 = A_{12}A_{22} + A_{11}A_{23} - A_{13}A_{24}.$

Analysis of the conducted theoretical studies showed that the controllability of the wide-span vehicle of the TGATU design at the speed of its movement 2 m • s^{-1} depends little on the change in the distance from the axis that passes through the attachment points of the lower links of the mounted device to the instantaneous turning centre (point S_0) with respect to the centre of mass (point S_T). At low fluctuation frequencies (up to 0.8 ... 0.9 s^{-1}) of the turning angle α of the frontal wheels of the tractor, in possible variants of the ratio of parameters d and D (Figure 2), we have significant overregulation but at high frequencies - underregulation of the control impact by the dynamic system. Besides the desired fluctuation frequency of the turning angle α of the frontal wheels of the tractor is on the level 0.8 ... 0.9 s^{-1}, where the value of the amplification coefficient of the input control impact is close to one.

The worst amplitude and phase frequency characteristics of the control impact developed by the dynamic system are observed when $d > D$ since in the operating range of frequencies these characteristics are most distant from the desired ones.

The most desirable amplitude and phase frequency characteristics of the control impact developed by the dynamic system are observed when $d < D$. In this case, in the operating range of frequencies, these characteristics are closest to the desired ones.

On the whole, the following trend emerges: in order to ensure satisfactory controllability of the wide- span vehicle of a new design, the fluctuation frequency of the turning angle of its wheels at a speed of 2 m • s^{-1} should be equal to 0.8 ... 0.9 s^{-1}. Otherwise the amplitude and phase frequency characteristics of the control impact developed by the dynamic system are approximated to the desired ones in the case when the instantaneous turning centre of the wide-span vehicle is in the centre of its mass.

Delayed response of the wide-span vehicle to the control action is the less, the less is distance d. It is another matter that the difference between the obtained phase-frequency characteristics is not significant and generally manifests itself only at the frequencies of the control impact, greater than 0.8 ... 0.9 s^{-1}.

The transfer function $W^\beta(s)$, which reflects a measurement of the change in the relative bearing of the wide-span vehicle from value β, according to the system of equations (3), can be presented as:

$$W^\beta(s) = \frac{C_{1\beta}s + C_{0\beta}}{s\left(B_2 s^2 + B_1 s + B_0\right)}, \qquad (5)$$

where $C_{1\beta} = A_{11}f_{22}; \quad C_{0\beta} = A_{12}f_{22} - A_{24}f_{12}.$

From the analysis of the development of a disturbing impact by the dynamic system it follows that with an increase in parameter d the amplitude-frequency characteristics undesirably increase. It is true that this process is more or less significant only at low fluctuation frequencies of angle β (0.5 ... 1.3 s^{-1}). The most desirable variant of the ratio of the considered parameters is when $d > D$. The latter condition indicates that the instantaneous turning centre of the wide-span vehicle should be in the centre of its mass.

This condition can be ensured by choosing such a convergence angle γ of the lower links of the mounted wide-span vehicle in which the instantaneous turning centre of the first link will be located in the zone of the centre of mass of the second link. In practice this can be achieved by setting the required value of c (Fig. 3).When $d = D$, parameter c in Fig. 3 should be equal to:

$$c = 1\left(1 - \frac{2h}{\sqrt{l^2 + 4D^2}}\right). \qquad (6)$$

And the convergence angle γ of the lower links of the mounted wide-span vehicle is equal to:

$$\gamma = 2\,\text{ATAN}\left(\frac{1}{2D}\right). \qquad (7)$$

Analysis of dependencies (6) and (7) for the mounted devices of high categories 4N and 4, according to ISO 730: 2009, showed that, increasing distance D from the axis that passes through the attachment points of the lower links to its centre of mass, the required increase in length c varies nonproportionally (Fig. 4). As nonproportional is also reduction of the convergence angle γ of the lower links. On the whole, increasing length D from 0.6 to 2.6 m, c increases from 0.02 m, which practically corresponds to full reduction of the lower links of the wide-span vehicle to one point, up to 0.77 m. In this case angle γ decreases from 85 to 24 deg.
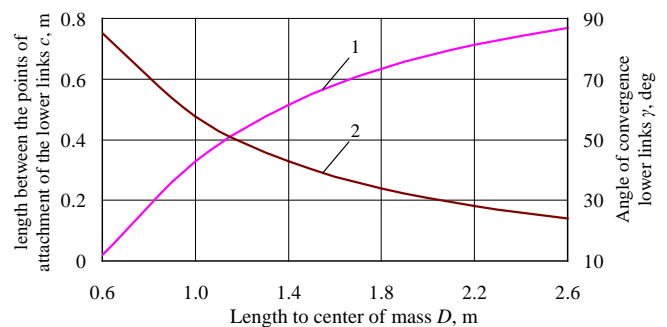
Fig. 4. *Dependence of the distance between the points of attachment of the lower links of the mounted wide-span vehicle (1) and the convergence angle of its lower links (2) upon the distance to its centre of mass*

The conducted investigations showed that, in order to ensure sufficient angular mobility of the technological part of the wide-span vehicle in relation to its framework, the required distance between the attachment points of the lower links of the mounted device is substantially determined by the distance to its centre of mass. Besides, the proximity of the location of the mounted device to the centre of the mass of the wide-span vehicle is determined by the fact that the required distance between the attachment points of the lower links of the mounted device should be so small that it corresponds to their complete reduction to one point. Therefore, from a position of designing a mounted multi-purpose device for the wide-span tractors (vehicles), one should consider a possibility to configure it, both according to a three-point and a two-point scheme.

4. Conclusion

1. It has been established during the research that the most desirable is such a variant of configuration of the mounted device of the wide-span tractor (vehicle) in which the instantaneous turning centre, formed by convergent rays of its lower links, is in the centre of the mass of the tractor.

2. To ensure sufficient angular mobility of the technological part of the wide-span tractor (vehicle) relative to its framework, the required distance between the attachment points and the convergence angle of the lower links of the mounted device is substantially determined by the distance to its centre of mass. When the latter is reduced, the required distance between the attachment points of the lower links of the mounted device can be so small that it corresponds to their complete reduction to one point.

3. From the position of designing a multi-purpose mounted device for wide-span tractors (vehicles), one should consider a possibility to configure it, both according to a three-point and a two-point scheme.

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