Research of parameters of hydrostatic transmission with axial flow divider for cases of multi-drive vehicle maneuvering

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Abstract. The article considers a mathematical model of hydrostatic transmission with an axial flow divider for a multi-drive vehicle during its maneuvering. The new method for controlling the energy parameters of the wheel drive by adjusting the angles of swash plates for each section of the axial flow divider depending on the radius of rotation of the multi-drive vehicle is presented and static characteristics are constructed.

1. Introduction The use of hydrostatic transmission (HST) for multi-drive vehicles has several significant advantages relative to transmissions with electrodynamic or mechanical drives. Therefore, the works of many authors are devoted to the study of new designs of HST and its control systems to increase its energy efficiency. In particular, the works [1-4] consider mathematical models of the HST with special emphasis on the swash-plate mechanism of adjustable pump [1], the HST using a hydraulic accumulator for energy recovery [2] and the HST of off-road multi-axle wheeled vehicles [3, 4].

One of the functions of transmission is the distribution of moments and frequencies of rotation of the wheels when the vehicle is moving along a curved path. When using hydrostatic transmissions, the wheels speed is controlled by changing the fluid flow through the hydraulic motor-wheels.

2. Materials and methods
In this paper, the use of an axial flow divider [5] is considered for independent control of angular velocity of each hydraulic motor in case of curvilinear movement of a vehicle. For the plotting of static characteristics of HST there was used the theoretical method of numerical research using the mathematical equations presented in the article.

Calculation of static characteristics was carried out using MathCad v.14 program under the following assumptions: the roughness of the roadway, the leakage and compressibility of the working fluid was not taken into account.

To calculate the kinematics of the vehicle’s turning there was used the scheme shown in Fig.1.
Fig.1. Scheme of the turning kinematics of two-axial vehicle

The turning radii of the rear left, rear right, front left and front right wheels are determined from equations (1) - (4), respectively.

\[ R_{z,lev} = R - \frac{b}{2}, \]
\[ R_{z,pr} = R + \frac{b}{2} = R_{z,lev} + b, \]
\[ R_{p,lev} = ((R + b_p / 2)^2 + a^2)^{0.5} - b_k, \]
\[ R_{p,pr} = ((R - b_p / 2)^2 + a^2)^{0.5} + b_k. \]

here \( R \) – turning radius of the center of the rear axle, m; \( a \) – wheelbase of a vehicle, m; \( b \) – distance between centers of rear left and rear right wheels, m; \( b_p \) – distance between front axle joints, m; \( b_k \) – distance from joint to wheel center, m.

The turning angles of kinematically connected right and left front wheels are determined from equations (5), (6):

\[ \gamma_{pr} = \arctg(a/(R + b/2)), \]
\[ \gamma_{lev} = \arctg(a/(R - b/2)). \]

Transforming equations (5), (6), we obtain expressions for determining the radius of the center of rear axis:

\[ R = a / \tan(\gamma_{pr}) - b_p / 2 = a / \tan(\gamma_{lev}) + b_p / 2. \]

Angular velocity of the vehicle when turning

\[ \omega_a = \nu / R, \]

here \( \nu \) – linear velocity of the vehicle, m/s.

Rotation frequencies of each wheel

\[ n_{z,lev} = 2 \omega_a R_{z,lev} / (\pi^2 R_k), \]
\[ n_{z,pr} = 2 \omega_a R_{z,pr} / (\pi^2 R_k), \]
\[ n_{p,lev} = 2 \omega_a R_{p,lev} / (\pi^2 R_k), \]
\[ n_{p,pr} = 2 \omega_a R_{p,pr} / (\pi^2 R_k). \]
\[ n_{p,pr} = 2\omega \frac{R_{p,pr}}{R_k^2}, \]  

(12)

here \( R_k \) – the wheel radius, m.

For the control system it is necessary to determine the total rotation frequency of all hydraulic motor-wheels \( n_{\text{max}} \) according to the equation:

\[ n_{\text{max}} = n_{z,\text{lev}} + n_{z,\text{pr}} + n_{p,\text{lev}} + n_{p,\text{pr}}. \]  

(13)

and inclination rates of swash plates of divider’s sections

\[ k_{z,\text{lev}} = \frac{n_{z,\text{lev}}}{n_{\text{max}}}, \]  

(14)

\[ k_{z,\text{pr}} = \frac{n_{z,\text{pr}}}{n_{\text{max}}}, \]  

(15)

\[ k_{p,\text{lev}} = \frac{n_{p,\text{lev}}}{n_{\text{max}}}, \]  

(16)

\[ k_{p,\text{pr}} = \frac{n_{p,\text{pr}}}{n_{\text{max}}}. \]  

(17)

Then the rotation frequency of the axial divider shaft

\[ n_d = \frac{q_m}{q_{\text{max,s}}} \cdot \frac{n_{z,\text{lev}} + n_{z,\text{pr}} + n_{p,\text{lev}} + n_{p,\text{pr}}}{k_{z,\text{lev}} + k_{z,\text{pr}} + k_{p,\text{lev}} + k_{p,\text{pr}}}, \]  

(18)

here \( q_m \) – displacement of hydraulic motor, cm\(^3\); \( q_{\text{max,s}} \) – maximum displacement of divider section, cm\(^3\).

Fluid flow rates through hydraulic motors without leakage

\[ Q_{z,\text{lev}} = n_{z,\text{lev}} \cdot q_m, \]  

(19)

\[ Q_{z,\text{pr}} = n_{z,\text{pr}} \cdot q_m, \]  

(20)

\[ Q_{p,\text{lev}} = n_{p,\text{lev}} \cdot q_m, \]  

(21)

\[ Q_{p,\text{pr}} = n_{p,\text{pr}} \cdot q_m. \]  

(22)

Total fluid flow rate supplied by the pump group:

\[ Q_{\text{pump}} = Q_{z,\text{lev}} + Q_{z,\text{pr}} + Q_{p,\text{lev}} + Q_{p,\text{pr}}. \]  

(23)

According to the presented equations, the calculation of the static characteristics of HST for curvilinear motion of a vehicle was performed using the KAMAZ-43501 as an example. As input data for calculations the actual parameters of the vehicle were used: \( R_k=570 \) mm; \( q_m=q_{\text{max,s}}=8 \cdot 10^{-5} \) m\(^3\); \( a=3670 \) mm; \( b=2105 \) mm; \( b_L=1710 \) mm; \( b_k=197.5 \) mm; \( v=50 \) km/h.

3. Results and Discussion

Figure 2 presents the results of calculations obtained using the mathematical model. It can be noticed that in case of one front driven axle, the rotation frequencies of hydraulic motor-wheels on the right side in the motion direction of the vehicle turning to the left increase with increasing turning angle (fig.2a). At the same time, the rear left motor-wheel slows down, and the front left motor-wheel first slows down in the range of rotation angles \( 5^\circ \ldots 25^\circ \), and then it smoothly accelerates. This is explained by different values of the radii \( R_{z,\text{lev}}, R_{z,\text{pr}}, R_{p,\text{lev}}, R_{p,\text{pr}} \) relatively to instantaneous turning center of O (fig. 1). The flow rate of the group of hydraulic pumps increases with increasing turning angle of the vehicle (fig. 2b).
Fig. 2. Fluid flows rates of hydraulic motors and required pump flow rate (a) and rotation frequencies of hydraulic motor-wheels and the divider’s shaft (b) depending on the turning angle of the front left wheel.

4. Conclusion
The presented static characteristics show the possibility of implementing independent volumetric regulation of rotation frequency of several undjustable hydraulic motor-wheels (for example, high-torque radial-piston hydraulic motors) having a common drive from one or a group of adjustable pumps. In this case, the electronic control system for the drive of considered HST, based on the developed mathematical model, will provide a qualitative ratio between rotation frequencies of the hydraulic motor-wheel shafts by adjusting the inclination of swash-plates of corresponding sections of the axial flow divider.

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