THEORETICAL INVESTIGATIONS OF VEHICLE OPERATING WITH AUXILIARY DRIVING WHEELS WITH HYDROSTATIC TRANSMISSION

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Abstract. In the work a task of driving wheel towing reduction of a mobile vehicle, which is exploited in bad passing conditions, is investigated. For that purpose transmission is used, when the engine-developed torque is transmitted to all driving wheels of the vehicle via the mean driving bridge differentials. For the auxiliary driving wheel transmission a hydrostatic system is used, made of constant operating volume hydraulic machines. In the work a mathematical model of hydraulic transmission of a mobile vehicle is given, being under analysis, basic properties of such transmission have been investigated, constructional parameter values, in which the torque distribution is realized so that the towing of driving wheels is minimum, have been detected. Also a hydraulic transmission power losses and the influence of the properties of some supporting surfaces to kinematical indexes of compatibility have been analyzed.

Key words: transmission, hydrostatic drive, towing, hydraulic motor, pump.

Introduction

Some mobile technological equipment, for example agricultural, wood works, rescue works, is exploited in complicated driving conditions, like soft ground, different dirt, etc. This may cause different negative confluence: draught force reduction, towing, fuel contribution increase, destruction of the soil and biological covering, etc. There are known miscellaneous mobile equipment, driving in bad conditions, technological solutions [1-4]. To reach this target the efficiency has the means, rising the supporting surface of driving wheels; starting with pressure reduction in the driving wheel tires [4], ending with miscellaneous type caterpillar chassis [5; 6]. The wheels supporting surface rising have shortages, worsening other important parameters of the vehicle. That is why the research of the wheeled vehicle drive ability improvement is a topicality.

The passing of wheeled mobile equipment can be effectively improved by using transmissions, committed for this purpose [7; 8]. The transmission efficiency can be evaluated by an index, showing the vehicle wheels equal draught. In creation of such transmissions it is prospective to use in them volume drives.

The aim of the work **–** to investigate the properties of one of the transmissions with hydrostatic drive.

The Investigation Object and Methodics

The investigated transmission scheme is shown in Fig. 1. In the given scheme there are designated: *CM* – general motor; *CU* – general drive; *D* – differential; M_T, *M* – torque of the differential input and output correspondingly; *A*, *B* – driving and auxiliary wheels correspondingly; U_A, U_B, U_{HS} – reducers; *HS* – pump; *HM* – hydraulic motor.

The torque, created in the general motor *CM* and reduced in the general reducer *CU* until the value M_T, is divided in the transmission into equal parts M_1=M_2=M_T/2=*M* and is distributed to semi axel 1 and 2 of the differential *D*. Next, the torque *M* is distributed into two directions:

- to the driving wheel *A* via the reducer U_A;
- via the reducer U_{HS}, pump *HS*, hydraulic motor *HM* and reducer U_B to the auxiliary wheel *B*.

In further analysis we assume, that the part of the forces of the resistance of the vehicle motion F_*, which is applied to the wheel *, being under investigation, does not exceed its force F_{S*} of coupling with the road coating. Also we assume, that the wheel * the draught force $F_{tr*} = F_*$, is expressed by the torque M_*, applied to this wheel *, and is equal to the force F_{p*}, which is the resistance to the wheel * creeping motion of the axis.

$$F_{tr*} = \frac{M_*}{R_*} = F_* = F_{p*}, \tag{1}$$

where R_* – the radius of the wheel *, m.

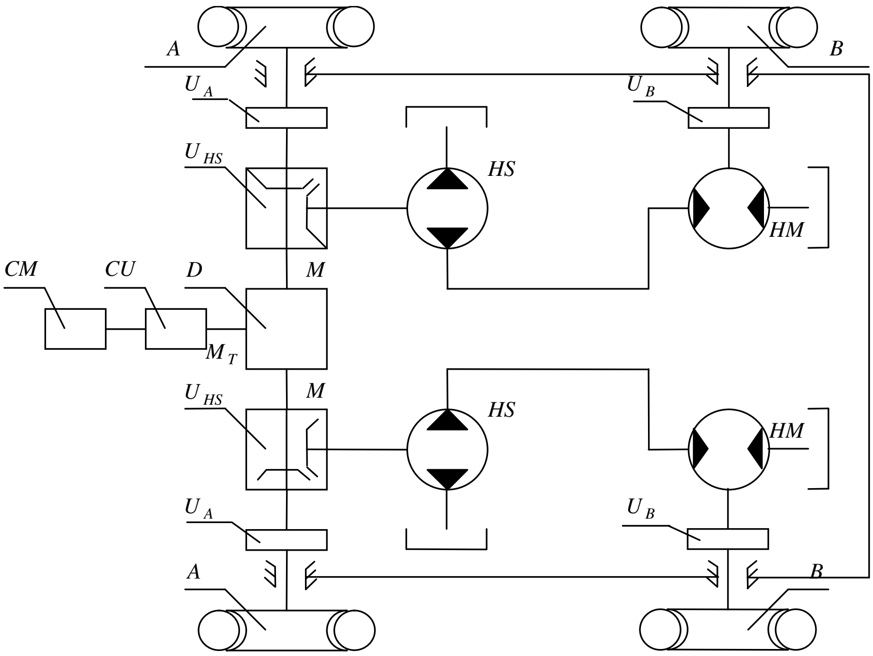

Fig. 1. **Scheme of transmission**

While the wheel * is applied with the torque M_* and the resistance to its axis creeping motion, in its contact with the road coating appears the contact deformation processes. Outwardly it is seen as wheel towing [1].

$$b = \frac{2\pi R_* n_* - v}{2\pi R_* n_*}, \tag{2}$$

where R_* – the radius of the wheel *, m;

n_* – wheel * rotational speed, s^{-1};

v – wheel * axis linear speed, $m \cdot s^{-1}$.

Empirically it is detected [3; 4; 8], that when the wheel draught force F_* does not exceed the coupling with the road coating force F_{S*}, its towing coefficient can be expressed in a linear dependence.

$$b = \frac{1}{k_*} F_*, \quad F_* \le F_{S*}, \tag{3}$$

where k_* – wheel'* contact with the road coating rigidity, [N], further named as "road rigidity".

Using (1)-(3), we express the rotational speed of a driving wheel A:

$$n_A = \frac{v}{2\pi R_A \left(1 - \frac{1}{k_A}\left(F_{PA} - F_B\right)\right)}, \tag{4}$$

Next we analyze the transmitting processes of the torque, applied to the axillary wheel B.

We express the oil flow debit in the pump HS in its output

$$\begin{cases} q_{HS} = n_{HS} V_{HS} \left(1 - \dfrac{k_{HS}}{100}\right) \\ n_{HS} = \dfrac{n_A U_{HS}}{U_A} \end{cases}, \tag{5}$$

where V_{HS} – pump *HS* cyclical volume, m^3;
k_{HS} – pump *HS* oil losses ratio, %;
U_{HS}, U_A – transmission of torques in the reducers U_{HS} and U_A correspondingly;

oil flow debit in the hydraulic motor *HM* input,

$$\begin{cases} q_{HS} = n_{HM} V_{HM} \left(1 + \dfrac{k_{HM}}{100}\right) \\ n_{HM} = \dfrac{n_B}{U_B} \\ n_B = \dfrac{v}{2\pi R_B \left(1 - \dfrac{1}{k_B} \dfrac{M_B}{R_B}\right)} \\ M_B = \dfrac{p V_{HM}}{U_B} \end{cases}, \tag{6}$$

where V_{HM} – hydraulic motor *HM* cyclical volume, m^3;
k_{HM} – pump *HM* oil losses ratio, %;
U_B – reducer U_B torque transfer;
R_B – wheel *B* radius, m.

In further analysis we assume, that the ratios k_{HS} and k_{HM} (k_*), showing the oil losses, have linear dependence on the oil pressure p

$$k_* = k_{*0} \frac{p}{p_0}, \tag{7}$$

where k_{*0} – nominal value of the loss ratio;
p_0 – nominal value of the oil pressure, Pa.

Oil pressure p is the function of the time t, which is expressed through differential equation

$$\dot{p} = C(q_{HS} - q_{HM}), \tag{8}$$

where C – rigidity ratio of the pumps *HS*, *HM* volumes, Pa·m^{-3}.

When the regime has stabilized, $\dot{p} = 0$. Due to the stabilized regime we put (4), (5) and (6) into (8), and after solving we receive balance equation of the transmission under analysis

$$\begin{cases} m \dfrac{1 - \dfrac{k_{HS}}{100} \cdot \dfrac{p}{p_0}}{1 - \dfrac{1}{k}\left(F_{pA} - F_B\right)} - \dfrac{1 + \dfrac{k_{HM}}{100} \cdot \dfrac{p}{p_0}}{1 - \dfrac{1}{k} F_B} = 0 \\ m = \dfrac{U_{HS} V_{HS}}{V_{HM}} \cdot \dfrac{U_B R_B}{U_A R_A} \\ F_B = \dfrac{p V_{HM}}{U_B R_B} = \dfrac{p}{p_0} \cdot p_0 \dfrac{V_{HM}}{V_B R_B} = \dfrac{p}{p_0} F_{B_0} \end{cases}. \tag{9}$$

As it was remarked earlier, the major goal of the transmissions under analysis is the equalizing tension of the driving wheel *A* and auxiliary wheel *B*. We will find the transmission ideal, if the tensions of the wheel *A* and *B* are equal, i.e.

$$F_A = F_B = \frac{1}{2} F_p . \tag{10}$$

The condition of ideal transmission can be realized by changing the coordinating ratio

$$m = \frac{U_{HS} V_{HS}}{V_{HM}} \cdot \frac{U_B R_B}{U_A R_A} . \tag{11}$$

Depending on the hydraulic machines *HS* and *HM* losses k_{HS} and k_{HM} of the volumes, the equation

$$m_0 = \frac{100 p_0 - k_{H\dot{S}_0} \cdot p}{100 p_0 + k_{H\dot{M}_0} \cdot p} \tag{12}$$

is used to define the dependence of ratios *m* value m_0 on $k_\Sigma = k_{HS_0} + k_{HM_0}$, as shown in Fig. 2.

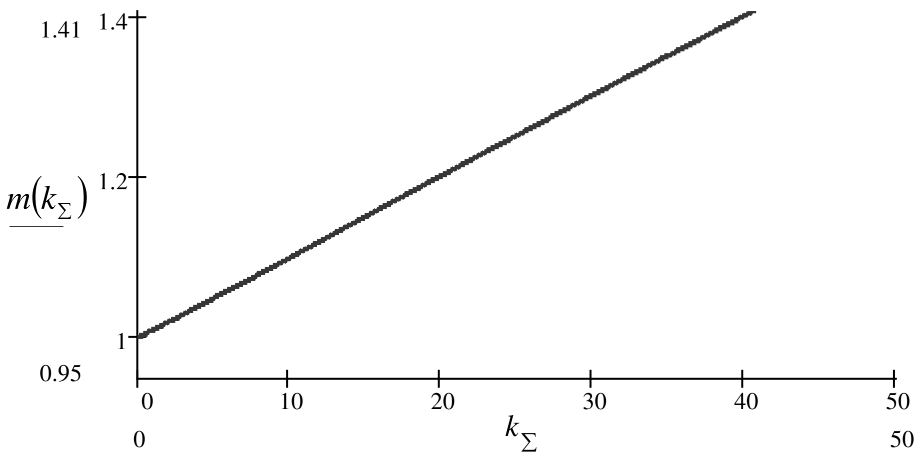

Fig. 2. **Dependence of coefficient** m_0 **on** k_Σ

Analysing transmission from the technological point of view, it is important to detect, how the maximum tensions of the wheels *A* and *B* vary, when the ratios k_{HS} and k_{HM} showing the pump *HS* and hydraulic motor *HM* losses of volumes vary among themselves independently. For this purpose we use variational calculation for the formulae (9), and through varying m, k_{HS_0}, k_{HM_0}, the maximum tensions of wheels *A* and *B* have been calculated

$$S = \max\left(\frac{F_p - F_B}{F_p} \cdot 100\%, \frac{F_B}{F_P} \cdot 100\% \right). \tag{13}$$

Generalized maximum tension values, received using different adjustment ratio *m* are shown in Fig. 3. The adjustment ratio *m* values, shown with numbers, which are written afterwards the symbol “s”, are shown in Fig. 3.

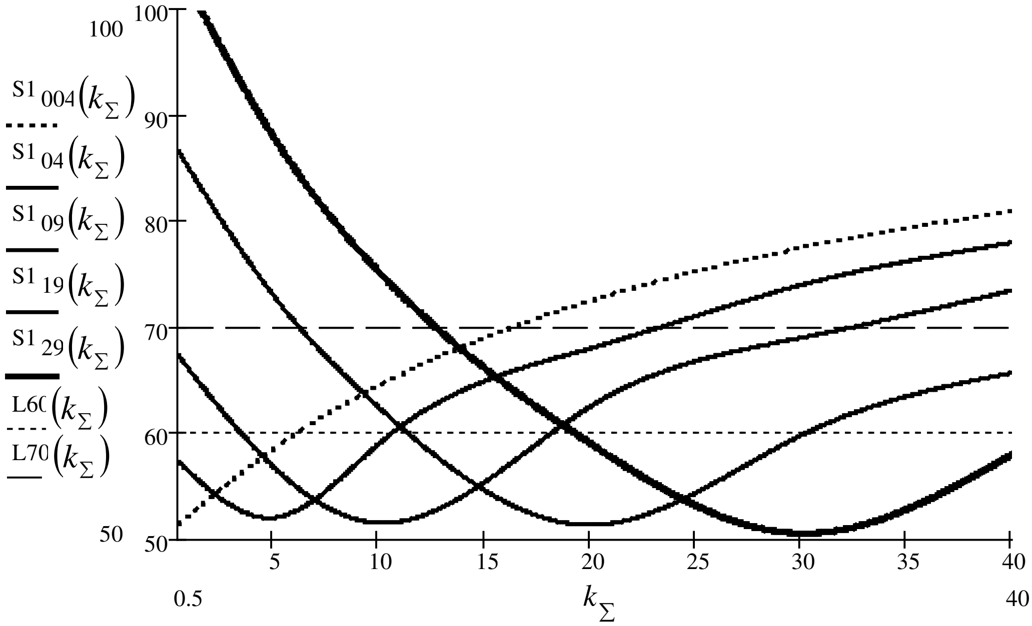

Fig. 3. **Dependence of wheel maximum tension on** k_Σ

Generalization of the Research Results and Conclusions

1. Generalized wheel maximum tensions S_*, calculated and shown in Fig. 3, and their dependences on the hydraulic machine total volume losses k_Σ allow us to make a conclusion, that application of transmission with hydrostatic drive practically reduces the tension values until not exceeding 60-70 % of the driving vehicle resistance to the motion value. The 60 % and 70 % of the tension values, shown in Fig. 3, are shown as lines $L60$ and $L70$ correspondingly.
2. In order to achieve the wheel tensions reduction until the value close to 50 % of the resistance force, it is necessary to achieve the adjustment parameter *m* (9) value within the interval 1.1-1.2. This can be achieved by varying the transmission element (see Fig. 1) U_A, U_B, U_{HS}, *HS*, *HM* parameters.

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