Wheeled tractors in the agricultural machine-tractor aggregates work efficiency opportunities

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Abstract. Improving the machine-tractor aggregate stability and performance is to create conditions for smooth regulation of the translational speed and the formation of traction resistance, depending on the kinematic parameters of the engine-transmission installation. Performing this task requires a phased solution. The first stage in the implementation of this task is to form a transition process from a stepwise change in translational speed to stepless one, which will provide the opportunity to coordinate the kinematic parameters of the agricultural implement working body and the tractor translational speed to create optimal operating conditions for the machine-tractor aggregate. This will protect the transmission from vibrations arising in the engine and transmission design, which will be significantly reduced due to special devices with elastic - damping properties, which will stabilize the loading conditions and dynamic loading of the machine-tractor aggregate transmission. The presence of elastic devices various types located in the structure of the transmission, mounted or trailed devices that generate traction on the tractor hook, reduce dynamic loads in all the transmission nodes. The ability to most effectively reduce dynamic loads provides an elastic pneumohydraulic element, which has the ability to control the optimal parameters in a wide range of elastic properties.

1. Introduction

It should be noted that the increase in the efficiency of the motor-transmission installation of a wheeled tractor for agricultural purposes is due to the creation of an original controlled transmission with elastic elements in the clutch. The introduction of such devices will provide the opportunity to create modern high-technical and technically advanced mobile machines [1,2,3].

A characteristic feature of this design is the use of an elastic element that contributes to a shift in the frequency of natural vibrations by selecting the elastic characteristics of a pneumohydraulic accumulator, as well as the implementation of artificial damping.

The use of an elastic element in the tractor’s transmission [4, 5] helps to provide an effective way to reduce the amplitudes of hook load oscillations and torsional vibrations from the engine in power transmissions. In order for the resonance phenomena to occur not at the operating conditions of the engine speed, the elastic elements in the clutch should be selected with the minimal rigidity.

2. Materials and methods

Therefore, we consider the issues of schematization of actual machine-tractor aggregates taking into account real stiffnesses in the system of load conditions formation in the nodes and mechanisms of
power installations equipped with pneumohydraulic elastically damping elements in the transmission. On the example of the machine-tractor aggregate with a wheeled tractor of class 1.4, mathematical models are compiled to determine the optimal parameters of the pneumohydraulic elastic element [6]. The research is based on theoretical and experimental studies of tractors in agricultural production in the arid zones of the south-east of the country, as well as on the basis of engines stationary characteristics. The obtained models demonstrate the methodology for compiling algorithms for computer calculating of the equivalent system elastically damping elements optimal parameters that evaluate the economic performance of a machine-tractor aggregate under random loads.

3. Discussion

The main quality indicator of the planetary flywheel clutch, is its components reliability. A transmission elastic element is considered one of the main design components, its operation can provide timely redistribution of energy from the power aggregate along the propulsion devices [7]. The main characteristic of an elastic element is its rigidity, its calculation is one of the priority tasks in the design development.

The force developed by the pneumohydraulic accumulator (PHA) and the fluid pressure in the hydraulic system depend on the elastic element stiffness. Then, the elastic element stiffness with constant engine load by the torque from the transmission side can be determined by using equation (1):

$$C_{\phi e} = \frac{1}{2\pi^2} \cdot \frac{\left(\delta_{BP} \cdot m_{TP} + m\right)^2 \cdot r^2 \cdot \lambda^2}{(m_{TP} + m + \frac{I_{TP}}{r^2}) \cdot \eta_{TP}}$$

where $\delta_{BP}$ is the coefficient of accounting for the rotating mass of the tractor; $m_{TP}$ is the mass of the tractor, kg; $m$ is the mass of agricultural implements, kg; $I_{TP}$ is the reduced moment of transmission inertia to the tractor drive wheel, kg m$^2$; $r$ is the dynamic radius of the wheel, m; $i_{TP}$ is the gear ratio; $\eta_{TP}$ is the transmission efficiency; $\lambda$ is the frequency (prevailing) of the transmission forced oscillations, c$^{-1}$.

It should be noted that a further decrease in stiffness gives rise to resonance modes in certain types of work [8] during ordinary tractor operation. However, the use of combined elastic elements can lead to an increase in more dangerous vibrations during the formation of the tractor average hook load; therefore, it is necessary to check the oscillatory system for the impossibility of the natural vibrations frequencies calculated from the partial (individual masses) frequencies in the vibration zone of the tractor loading spectral density to be cancellation. They can be determined by using the formula (2):

$$\nu = \frac{\sqrt{2}}{2\pi} \cdot \frac{\delta_{BP} \cdot m_{TP} + m}{m_{TP} + m + \frac{I_{TP}}{r^2}} \cdot \lambda$$

where $\nu$ is the frequency of natural oscillations, c$^{-1}$; $\delta_{BP}$ is the coefficient of accounting for the tractor rotating masses; $m_{TP}$ is the mass of the tractor, kg; $m$ is the mass of agricultural implements, kg; $I_{TP}$ is the reduced moment of transmission inertia to the tractor drive wheel, kg m$^2$; $r$ is the dynamic radius of the wheel, m; $\lambda$ is the frequency (prevailing) of the transmission forced oscillations, c$^{-1}$.

Then the optimal stiffness of the PHA elastic element, reduced to rectilinearly moving masses, can be calculated (3):

$$C_{M(\eta)} = \left(m_{TP} + m + \frac{I_{TP}}{r^2}\right) \cdot \gamma^2$$
If we substitute the natural oscillations frequency value from (1) in this dependence, we obtain:

$$C_{M(\text{TP})} = \frac{1}{2\pi^2} \left( \delta_{\text{np}} \cdot m_{\text{TP}} + m \right)^2 \left( m_{\text{TP}} + m + \frac{I_{\text{TP}}}{r^2} \right) \cdot \lambda^2$$  \hspace{1cm} (4)

To calculate the flywheel clutch stiffness, reduced to the gearbox input shaft (carrier planetary clutch), we first bring this stiffness to the drive wheel shaft by the following method.

$$C_{\text{MK}} = \frac{\Delta M_K}{\Delta \phi_K}$$  \hspace{1cm} (5)

where $\Delta M_K$ is the change in moment on the wheel, $\Delta \phi_K$ is the wheel shaft twist angle corresponding to this change in moment, $\Delta M = \Delta P_{K_P} / r$, $\Delta \phi_K = \Delta x/r$, ($\Delta x$ is the displacement of a horizontal elastic element with the rigidity $C_{M(\text{TP})}$).

Taking into account these changes, we get:

$$C_{\text{MK}} = \frac{\Delta P_{K_P} \cdot r}{\Delta x / r} = \frac{\Delta P_{K_P} \cdot r^2}{\Delta x}$$  \hspace{1cm} (6)

In this equation, $\frac{\Delta P_{K_P}}{\Delta x}C_{M(\text{TP})}$ is the stiffness of the clutch reduced to the linear movement of the machine-tractor aggregate.

And then we determine the stiffness reduced to the carrier of the planetary gear.

$$\frac{\Delta M_u}{\Delta \phi_u} = C_{\text{gu}} = \frac{\Delta M_K}{\Delta \phi_K} \cdot \frac{1}{i_{\text{mp}} \cdot \eta_{\text{TP}}} = \frac{\Delta M}{\Delta \phi_K} \cdot \frac{1}{i_{\text{mp}}^2 \cdot \eta_{\text{TP}}}$$  \hspace{1cm} (7)

where $i_{\text{mp}}$ is the gear ratio of the transmission, $\Delta M_B$ is the change in moment on the planet carrier, corresponding to its rotation by the angle $\Delta \phi_B$, $\eta_{\text{TP}}$ is the transmission efficiency.

The main purpose of mechanisms using with elastic elements in various places of the power drive is not to damp the incoming signal ($P_{\text{tr}}$), but to implement the conditions for the optimal interaction of the agricultural implements working bodies with the processed material, which are determined by a decrease in the tractor hook average loads and their dispersions.

But the peculiarity of the planetary clutch device allows to realize the ability to control the pneumohydraulic accumulator stiffness by changing the transmission gear ratio and the planetary gear set gear ratio [9]. It should be taken into account that the gear ratio $i_{\text{tp}}$ is not a complete gear ratio determined by the ratio:

$$i_{\text{tp}} = \omega_{\text{p}} / \omega_{\text{e}}$$

Then the adjustment of the gear ratio can be carried out by changing of the clutch planetary gearbox internal gear ratio. With the planetary center gear stopped, the peripheral speed of the satellite centers will be equal to half of the hollow gear peripheral speed.

$$\Omega_{\text{e}} r_e = \frac{1}{2} \cdot \omega_{\text{e}} r_e$$  \hspace{1cm} (8)

where $r_{\text{e}}$, $r_\text{e}$ are the radii of the carrier and the hollow gear.

The peripheral speed of the satellites relative to the carrier will be equal to the difference between the relative speeds of the hollow gear and the carrier:
ωₜ; τₑ=ωₜ·τₑ

and the engine rotational rate can be expressed in terms of the carrier rotational rate:

ωₑ=ωᵦ· \frac{K+1}{K} \text{ or } ωₑ=ωᵦ· \frac{K+1}{K}, \text{ then: } \omega_{\text{car}}=ωᵦ· \frac{K}{K-1}· \frac{K+1}{K} = ωᵦ· \frac{K+1}{K-1}.

where \(\omegaᵦ\), \(\omega_{\text{car}}\) - is the carrier and satellite rotational rate.

When the tractor performs any specific type of work: plowing, harrowing, sowing or fertilizing, the hook force of \(P_{\text{КР}}\) changes, and, therefore, gear shifting in the box is required (new gear ratio of the \(i_{\text{TP}}\) transmission). In this case, gear shifting is considered as the engine load controlling means. But this is quite difficult to accomplish when the machine-tractor aggregate with a load on the hook moves. Therefore, we propose to make adjustment of stiffness by changing the planetary clutch gear ratio.

Then the \(i_{\text{TP}}\) transmission gear ratio in the actual gear without taking into account the planetary gear ratio can be determined:

\[ i_{\text{TP}} = \frac{i_{\text{TP}}}{(K+1)/K} = i_{\text{TP}} · \frac{K}{K+1} \]

Gear shifting in this case is considered as the need to control the engine load. This operation is quite difficult to perform during the machine-tractor aggregate work with a load on the hook. At the same time as the gearshift, the following parameters change: the machine-tractor aggregate speed, calculated frequency of shock phenomena \(λ_{\text{с}}^1\) and cofactor \((δ_{\text{пр}}·m_{\text{TP}}+m)\) in the numerator of the formula, since \(δ_{\text{пр}}\) is a function of \(i_{\text{TP}}\).

Then, the change in the optimal stiffness limits of the pneumohydraulic accumulator ensuring the operability of the planetary clutch, depending on the work type performed, is calculated according to (6):

\[ C_{\text{pil}} = \frac{1}{2π^2} \cdot \frac{(δ_{\text{пр}}·m_{\text{TP}}+m)^2 · r^2}{(m_{\text{TP}} + m + \frac{I_{\text{TP}}}{r^2}) · i_{\text{TP}}^2 · i_{\text{TP}}^2 · (k+1)^2 · η_{\text{TP}} · η · η_{\text{TP}}}, \]

where \(i_{\text{TP}}\) is the transmission gear ratio; \(i_{\text{пр}}\) is the pump transmission gear ratio; \((k+1)\) is the gear ratio from the carrier to the center gear; \(η_{\text{пр}}\) is the transmission efficiency; \(η\) is the transfer efficiency from the carrier to the center gear; \(η_{\text{TP}}\) is the pump drive efficiency.

4. Conclusion

Comparative operational and technological tests revealed the fact that, in all types of work, an elastic drive, due to a decrease in slippage and an increase in the uniformity of the engine load regime, improves machine-tractor aggregate performance; this is especially noticeable on soil backgrounds with reduced hardness and viscosity. The productivity of the experimental unit exceeds this serial indicator by 8.54% during plowing, by 17.73% during cultivation and by 5.68% during transport operations, and fuel consumption is lower by 13.47%; 21.34%; and 10.78% respectively.

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