Self-oscillation in agricultural mobile machine units

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Abstract. The article presents research on improving the dynamic properties of mobile equipment due to the optimal choice of mass-geometric and elastic-dissipative parameters of the running system. In the study of the sensitivity of systems to synchronization and the capture of these oscillations, a frequency analysis of the oscillations of the output shafts of the propulsion machines and executive units under various operating conditions was carried out. It has been established that under certain operating conditions, the coincidence of these oscillations frequencies is possible, which is the main reason for the appearance of the main dynamic loads. Measures to reduce these loads have been proposed.

1. Introduction
A high level of mechanization and automation of working movements is a special feature of the modern machines functioning. In pneumatic equipment, the inertia of the output links is relevant [1-3]. In the case of a mechanical and hydromechanical drive, often several machine units make movements with different frequencies. If there are connections between them, the movement is realized with the same or multiple frequencies, while certain phase relations are established between these oscillations. In some cases, the behavior of aggregates can be described using differential equations of motion, which can be written in the operator form [4,5]. The main task of the theory of synchronization is to establish the fact of the presence of synchronous movements and the stability of the solution. It is known that if the system allows synchronous movement, then the aggregates show a tendency to synchronization [6,7].

2. The main part of the research
In many cases, the behavior of aggregates can be described using differential equations, which can be represented as follows:

\[ z(t) = z_0 + \mu \cdot y_i \]  \hspace{1cm} (1)

where: \( \mu \) - communication parameter; \( y_i \) - communication function.

For system (1), a periodic solution is sought with periodic coefficients according to integral positive values in the form:

\[ z(t) = z^*(t) + \mu \cdot z_i(t) \ldots \]  \hspace{1cm} (2)
In this case, a system of equations is obtained, which is a sufficient and necessary condition for the existence of a periodic solution. The main task of synchronization is to establish the stability conditions for these solutions.

When conducting field trials [8], it was found that between low-frequency angular vibrations of the output shafts of the drives of agricultural machinery (combustion engine of the diesel type) and actuating units (driving wheels of combines, milling cutters of cultivators, rotary plows, etc.) interconnected low-frequency oscillations occur [9,10], which depend on the mass-geometric and elastic-dissipative characteristics of the systems [11,12].

Figure 1.2 show the oscillations oscillograms of the cutter of frontal rotary plow PR-2.7, the thresher case of combine Don-1500 and the corresponding angular oscillations of the output shafts of the T-150 wheeled tractor propeller. From Fig. 1.2 it is seen that the vertical oscillations of the case of the combine thresher, the angular vibrations of the milling drum of the plow, are associated with the oscillations of the output shafts of the propellers. The correlation coefficient of the processes here lies within $K = 0.61 \div 0.84$. The dynamics of the machine drive is mainly determined by the oscillations caused by the movement on the field microrelief roughness.

![Oscillogram](image)

**Figure 1.** Summary oscillogram of torques in the propeller($M_d$) and in the milling drum ($M_f$) of the PR-2.7 plow.

When studying the sensitivity of systems to synchronization and capturing these oscillations, differential equations were compiled using the well-known Lagrange equations. In the operator form of writing, with the right-hand side equal to zero, these equations have the following form:

for the propeller:

$$T_0^2 \rho^2 + 2T_0 \eta_1 \rho + 1 = 0 \quad (3)$$

for the drive unit:

$$\rho^2 + 2\eta_2 \rho + \omega_0^2 = 0 \quad (4)$$

where: $T_0$ - time constant of the drive of the propeller; $\eta_1, \eta_2$ damping of the oscillation’s coefficients in the propeller and drive unit; $\omega_0$ - circular frequency of natural oscillations; $\rho$ - the differentiation operator.
Figure 2. Oscillograms of the torques in the propeller (Md) and vertical accelerations (z') of the combine thresher case.

The damping decrements for equations (3) and (4) are respectively equal to:

\[ D_1 = \omega_0^2 - \eta_2^2 \]
\[ D_2 = 1 - \eta_2^2 \]

For \( \eta_1 \geq 1 \) and \( \eta_2 \geq \omega^2 \), equations (3) and (4) have at least one solution when \( \eta_1 = 1 \) and \( \eta_2 = \omega_0^2 \), which is a sufficient and necessary synchronization condition. Therefore, when making oscillatory movements, they oscillate with the same frequency.

The analysis of the results of field trials was carried out using well-known methods of mathematical statistics [13,14]. Figure 3 shows the graphs of the normalized spectral densities of the vertical accelerations of the combine thresher case during movement on the roughness of the field (1) and country road (2).

Spectral analysis at various speeds along a different field agricultural background allowed us to establish that the oscillation frequencies are low-frequency (\( \omega_s \leq 6 \div 10 \text{ s}^{-1} \)), and the energy spectrum does not exceed \( \Delta \omega_0 \approx 5 \div 7 \text{ s}^{-1} \).

Figure 3. Normalized spectral densities of vertical oscillations of the of the combine thresher case during movement on a field (1) and on a country road (2).

Figure 4 shows graphs of normalized spectral densities \( S(\nu) \), where: \( \nu \) is the circular frequency of torsional oscillations of the drive (curve 1) and vertical vibrations of the case \( S(\omega) \); \( \omega \) is the circular frequency of the vertical oscillations of the case(curve 3) in transport modes of operation. From graphs 2 and 3 it can be seen that the drive and the case oscillate at almost the same frequency \( \Delta \omega 1 \approx 0.8 \text{ s}^{-1} \).
3. Conclusions
In operating modes, the difference in natural frequencies is greater than $\Delta \omega_2 \approx 5 \text{ s}^{-1}$ (figure 4, curves 1 and 3). This made it possible to establish that the main reason for synchronization of oscillations is the coincidence of the frequencies of the natural oscillations of the case and the drive-in transport modes. To disrupt resonance phenomena, a mismatch of these oscillations is necessary. Presented in figure 4, the estimated spectral density graph (curve 4) of the drive vibrations can serve as the basis for choosing the oscillation parameters of the machine. Frequency analysis allows to reasonably choose the values of the mass-geometric and elastic-dissipative characteristics of the chassis of the machine, for example, the oscillations parameters $T_0$ and $\eta_1$ of the propeller.

It should be noted that the increase in the level of reliability is affected by a decrease in the angular drive stiffness of the running system of combine Don by 20%, which leads to a decrease in the amplitudes level of torque oscillations by 28% during transport modes.

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