Increasing the smoothness of the course of the forage harvester by optimizing the mass-dimensional and inertial parameters of its body

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The scientific-methodical substantiation and practical recommendations on stabilizing the movement of forage harvesters are proposed. Calculation of optimal mass and inertial parameters of the hull, as well as elastic properties of tires as the main element of the combine suspension system is theoretically grounded. The developed technique on the example of the combine prototype shows the possibility of calculating the optimal stiffness of tires and weight of additional counterweights installed in the base of the machine to ensure the equality of natural frequencies of front and rear axle oscillations on elastic tires. The modeling of the combine's movement on the subsoil road and asphalt concrete highway with different speeds has been carried out on the basis of mathematical and imitation model of the combine developed by the authors earlier. In the order to evaluate the efficiency of the proposed method of improving the smoothness of running the comparison of peak and mean square values of the pitch angle of the hull, as well as the levels of vertical vibroaccelerations in the cabin on the floor under the seat of the operator of the combine in its basic design and improved version due to the use of counterweights of a given mass and optimized stiffness of tires. A simulation was used to perform an octave analysis of the workplace vibration load. It is shown that optimization of mass and inertial parameters of the body, as well as elastic properties of the combine harvester tires on the main driving modes provides improvement of comfort of the operator's working place, especially in the most dangerous for human frequency range. The driving modes are defined, on which the best effect of rework is achieved. The stabilization process is described. The conclusions on the direction of further research are presented.

Introduction

The existing system of classification of motor transport, self-propelled vehicles of agricultural, road construction, military and other purposes assumes their division not only by their purpose, design and layout, but also predetermines the approaches and methods of design [1, 2]. Until recently, the self-propelled combine harvester was considered by designers and marketers, first of all, as a technological machine, and in this connection in the process of its creation they tried to provide a competitive level of only technological properties. As a result, they received a self-propelled machine, effective in terms of technological indicators, but having an extremely low level of transport and operational properties, due to a significant imbalance of mass and size indicators, as well as the lack of a suspension system. In this connection at operation of combines it is noticed: reconsolidation of agrophones because of the
excessive weight which is falling on wheels of the forward axis; separation of wheels from a basic surface at movement with speed more than 8-10 km/h and emergency braking; galloping of the case on elastic tyres with formation of considerable dynamic loadings on the case, and also high vibronic loading of a workplace of the operator [3-10]. Especially such problems are shown at operation of forage harvesters (FH).

Purpose of the study
The listed disadvantages are determined, first of all, by mass and inertial parameters of the body, as well as by unbalanced level of elastic dissipative properties of tyres as the only element of the body suspension system. Despite this, the designers did not take into account the stiffness of the tires, assigning the pressure in them based on the recommendations of manufacturers, which in turn are guided by the requirements of reliability and provide a given pressure in the contact spot [11-13]. However, recently combine manufacturers began to consider the properties of tires as a factor of smooth running control. For example, the CLAAS FH model JAGUAR 900 (Fig. 1 a) is equipped with a tire pressure control system, which allows for the change of the area of the wheel's spot contact with the bearing surface, the reduction of the rolling resistance of the wheels when driving on hard roads and the rigidity of the tires in order to control the smooth running parameters.

Another obvious and easiest way to reduce the dynamic load on the combine's body is to level the weights on the axles by means of additional counterweights mounted on the rear of the FH. The weight of the counterweights is chosen on the basis of the designers' recommendations or after a set of experimental measurements on the prototypes of combines with different variants of the attached implements.

Despite the fact that such a solution is currently used by all major manufacturers of FH, however, to date there is no information on the application of specific techniques and practical recommendations for the implementation of these technical solutions. The lack of methodology and practical recommendations for selecting the parameters of counterweights and stiffness of elastic tyres predetermined the purpose of this work.

Materials and methods
The study was carried out with the help of the mathematical and corresponding simulation model FH RSM 2650, developed on the basis of the results obtained in the works [14-19]. The model describes the dynamics of the combine harvester motion taking into account its weight and dimensions, elastic and viscous bonds, peculiarities of the interaction of the propulsor with the roughness of the support base and other parameters. It is assumed that the masses of the harvester's suspension elements are reduced to the load-bearing system, and the support base is supposed to be non-deformable. When modeling the dynamics of the harvester's movement its body was considered as absolutely rigid structure. The relationship between the kinematic parameters and external disturbances is described by means of differential equations, which make up the mathematical model of the combine's motion. The modeling is carried out on an uneven non-deformable support base of the type "asphalt concrete highway" and "dirt road", to describe the profile of which used the correlative functions given in the well-known works [20-21].

The study of the distribution patterns of mass and inertial pairs-meters was carried out on the basis of the method of conservative dynamic systems with many related degrees of freedom [22].

As efficiency indicators of the proposed method are accepted: vertical vibroacceleration at the operator's workplace in the first five octaves; standard deviation $\phi_{sd}$ and peak value of longitudinal inclination of the combine harvester body $\phi_{max}$. The simulation measurements of vertical vibroaccelerations in the operator's cabin on the floor under the driver's seat are carried out in accordance with GOST 12.1.012.

Results and discussion
The FH dynamic model is based on the type of "bicycle" calculation scheme on elastic tires (Fig. 1) to evaluate the effectiveness of the application and to develop practical recommendations for the method of motion stabilization by adjusting the tire stiffness and application of counterweights. Such representation of the FH model is possible because of the lack of connection between transverse angular oscillations of the FH body and other types of its motion [2, 22].

![Calculation diagram of the forage harvester on elastic tyres](image)

**Fig. 1.** Calculation diagram of the forage harvester on elastic tyres:

- *C* – center of mass of the combine; *X, Z* – axes of stationary coordinate system; *C₁, C₂* – tire stiffness of front and rear axles respectively; *l₁, l₂* – distance from the center of masses of the combine to the front and rear axles respectively; *z₁, z₂* – vertical coordinates of points of fixing the wheels of front and rear axles respectively; *φ* – angle of longitudinal inclination of the combine body; *M, J* – total mass and the main moment of inertia relative to the transverse axis of the combine.

According to Fig. 1 of the equation of motion of the combine on elastic tires we write as

\[
\begin{align*}
M \dddot{z}_c +& (C_1 + C_2) z_c + (C_1 l_1 - C_2 l_2) \varphi = 0; \\
J \dddot{\varphi} +& (C_1 l_1 - C_2 l_2) z_c + (C_1 l_1^2 + C_2 l_2^2) \varphi = 0,
\end{align*}
\]

where \(z_c\) – vertical movement of the combine's centre of gravity; \(z_c\) – acceleration of the combine's centre of gravity in the vertical direction.

We enter the symbols:

\[
\omega_1^2 = \frac{C_1 + C_2}{M}; \omega_2^2 = \frac{C_1 l_1^2 + C_2 l_2^2}{J}; \frac{d}{dt} = p; \frac{d^2}{dt^2} = p^2,
\]

where \(\omega_1\) – natural frequency of vertical oscillations of the enclosure; \(\omega_2\) – natural frequency of longitudinal and angular vibrations of the enclosure.

The system identifier (1) can be recorded as [22]

\[
\begin{bmatrix}
p^2 + \omega_1^2 \\
\omega_1 \omega_2 \\
\omega_1 \omega_2 \\
p^2 + \omega_2^2
\end{bmatrix}
\begin{bmatrix}
\alpha \omega_1 \omega_2 \\
\alpha \omega_1 \omega_2 \\
p^2 + \omega_2^2
\end{bmatrix} = 0;
\]

\[
p^4 + (\omega_1^2 + \omega_2^2) p^2 + \omega_1^2 \omega_2^2 (1 - \alpha^2) = 0;
\]

\[
\alpha^2 = \frac{(C_1 l_1 - C_2 l_2)^2}{(C_1 + C_2)(C_1 l_1^2 + C_2 l_2^2)},
\]

where \(\alpha\) – the bond coefficient.

Then the roots of equation (2) can be defined as
where $\Omega_1$, $\Omega_2$ – natural angular frequencies of vertical and longitudinal angular oscillations respectively.

It has been shown in [22] that, for biaxial wheeled machines, an improvement in smooth running can be achieved by ensuring the equality of $\omega_1$ and $\omega_2$. In order for the two natural frequencies of the oscillating system (1) to coincide with some optimum frequency, the condition must be fulfilled $\Omega_1^2 = \Omega_2^2$. It follows from this that

$$C_1 l_1 = C_2 l_2;$$

$$\frac{C_1 l_1^2 + C_2 l_2^2}{J} = \frac{C_1 + C_2}{M}.$$

Therefore, the geometric $l_1$, $l_2$, inertial $M$, $J$ and stiffness parameters $C_1$, $C_2$ should be in a certain ratio in order to minimize the system oscillations (1) caused by the influence of the supporting base, in certain expressions (3) and (4). Expressions (3) and (4) allow optimizing mass and dimensional properties and inertial parameters of FH to provide a given level of vertical oscillations and minimize longitudinal-angle oscillations.

It is often difficult to maintain these ratios during design. The tyre rigidity can be changed by changing the air pressure in the tyres. In order to ensure smooth running, FH tyres are better suited for optimising the properties of the front axle tyres, as they are larger than the rear axle tyres and therefore have a larger stiffness control range.

The expression (3) can then be rewritten as

$$C_1 = \frac{l_2}{l_1} C_2.$$
\[ x = \frac{m_p}{M + m_p} \left( I_2 + l_p \right). \] (6)

If we subtract expression (5) from (4), we shall receive, subject to (6)

\[ \frac{(l_2 - x)(l_2 + x + 1)}{J_o + m_p(l_2 - x + l_p)^2} = \frac{L}{(l_1 + x)(M + m_p)}; \]

\[ J_o = J + J_p; L = l_1 + l_2, \]

where \( J_o \) – main moment of inertia with respect to the transverse axis of the combine with counterweights.

For simplification of the further transformations we will accept without essential loss of accuracy \( l_o \approx 0 \). Finally, we obtain a quadratic equation to determine the optimal \( m_p \):

\[ am_p^2 + bm_p + c = 0; \]

\[ a = 2l_1l_2 + l_2 + Ll_2 + l_1^2 + l_1; \]

\[ b = l_1^2 + 2Ml_1l_2 + Ml_2 + 2l_2^2M + 2Ml_2 - LJ_o - LMl_2; \]

\[ c = l_2^2M^2 + l_1^2M^2 - LJ_oM, \] (7)

where \( a, b, c \) – quadratic equation coefficients.

Solving equation (7) by known methods, it is possible to determine the optimal value of \( m_p \) taking into account the rigidity of tyres, mass and geometric characteristics of the body. On the basis of expression (7) it is established that for FH RSM 2650 the optimal \( m_p \) is 1450 kg, and the stiffness of front wheels should be reduced from 3125000 to 1856335 N/m. The stiffness of the rear axle tyres did not change, in all cases it corresponded to the level of production tyres of the company "Mitas" with the dimension of 21.3R24 and was 1230769 N/m.

According to the results of simulation it has been established that the vibration load of the operator's workplace on the improved version of FH in comparison with the basic version is reduced by 1÷7 dB, and the greatest effect is achieved in the most dangerous for the human body 3, 4 and 5 octaves (Fig. 3). In the first and second octaves it is possible to increase the active vibration level up to 3 dB, which is associated with some increase in the swinging of FH due to a decrease in the rigidity of the front axle tires and increase in the body weight due to counterweights. However, the permissible levels of exposure do not exceed the standard requirements.

As you can see from the fig. 4, FH refinement by the proposed method resulted in some increase of \( \phi_{sd} \) and \( \phi_{max} \) at the speed of 5 km/h. This is due to the fact that the roughness of the supporting surface forms power disturbances with a frequency close to the frequency of the body's own vibrations on elastic tyres. It should be taken into account that swaying is also facilitated by a reduction in the stiffness of the front axle tires, as the more solid body of the combine presses the softer tire into the larger foamy ground. At 10 km/h, there has been an improvement in running smoothness: \( \phi_{sd} \) has been reduced by 0.2˚ and \( \phi_{max} \) by 1.2˚.

In the case of traffic without an adapter on the asphalt concrete highway (mode "long-distance transport"), the level of vibration load of the operator's workplace FH as a whole increases with the increase in speed from 10 to 35 km / h (Fig. 5). Reduced front wheel stiffness and counterweights at speeds of up to 30 km/h result in some increase of vibration at the workplace only in the first octave (Fig. 5a-c), and in more dangerous from the point of view of labor protection third, fourth and fifth vibration loadings decrease is achieved. The greatest effect of finishing the combine is manifested in the third octave, which achieves a reduction in vibration from 2.5 to 8 dB (Fig. 5a-c). In the "long-distance transport" mode, the greatest result of the harvester's reworking was found at a speed of 20
km/h. At the maximum speed, the vibration level at the operator's workplace of the reworked combine has decreased noticeably only at the first octave (Fig. 5g) due to the reduction of galloping of the combine's body as a whole (Fig. 6). At 35 km/h, the level of vibration load in the cabin of the reworked combine is practically the same as in the basic version.

![Fig. 3. Current levels of vertical accelerations in the operator's cab for combine harvesters with an adapter when driving on a dirt road at 5 (a) and 10 (b) km/h](image)

When the combine moves along the asphalt without adapter, the finish of the combine significantly improves the smoothness of travel at a speed of 10 and 35 km/h (Fig. 6). However, at a speed of 20-30 km/h, the values of $\varphi_{sd}$ and $\varphi_{max}$ increased, primarily due to the prevalence of low-frequency hull oscillations (see Fig. 5b, d). Especially the smooth running parameters decrease at the speed of 30 km/h. It is obvious that at low speed the effect is achieved due to smoothening of vibrations by softer tires, and at the maximum because of the increase in the mass of the body sprung on the tires. At a speed of 10 km/h, the reduction of $\varphi_{sd}$ is due to the smoothing of vibrations by softer tyres, but for the same reason the level of $\varphi_{max}$ increases.

Thus, a set of measures has been developed to improve the design of the FH, consisting in changing the stiffness of the front axle tires and attaching additional counterweights to the rear of the combine. In contrast to the known methods of stabilization of self-propelled machines, the developed method involves bringing the parameters of the combine's oscillating system to a certain ratio, which ensures the minimization of vibration, primarily by narrowing the bandwidth of the body's own vibrations.
Fig. 5. Comparison of the current levels of vertical accelerations in the operator's cab for combine harvesters without an adapter when driving on an asphalt-concrete road at a speed of 10 (a), 20 (b), 30 (c) and 35 km/h (d)

Fig. 6. Comparison of peak (a) and standard pitch values of the hull (b) FH without adapter when driving on an asphalt-concrete road

Conclusion
1. When the combine is in transport mode, its reworking is efficient by optimizing the weight and inertia of the machine and helps to reduce the vibration load of the FH operator's workplace.
2. The optimization of wheel stiffness and inertial parameters of the body allows to improve the smoothness of the combine and comfort at the operator's workplace only for a given mode or speed. An effective way to stabilize the FH is probably to use some semi-active and active actuators in the hanger or attachments.

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