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INVESTIGATION OF THE WORKING CAPACITY OF THE OPERATING BODY SUSPENSION FUNCTIONAL-TRANSPORTING MACHINE

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Summary. *The investigation concerning the identification of the effectiveness of reducing the oscillations of the sprayer boom developed by independent pendulum suspension is carried out in this paper. In order to achieve this goal, the field microrelief, which creates kinematic perturbation of the rod oscillations is simulated. Dynamic model of rod mass oscillations on this suspension is also constructed, numerical solutions for a given sprayer operation mode are obtained, and numerical results are analyzed.*
Key words: *oscillations, rod, sprayer, dynamic model, independent suspension, vertical displacement, angular displacement, field irregularities.*

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Statement of the problem. While performing the technological process of spraying crops, one of the important factors in the quality of boom sprayer operation is the uniformity of application of the working preparation on the plant surface. The key role here is played by the sprayer boom when it is hung on a specially designed suspension, which should dampen the oscillations that occur during the movement of the sprayer field irregularities. Only in the case of the most stabilized bar operation, you can clearly and relatively accurately calculate the cost of the working preparation for the treated area. Therefore, there is a need to develop and analyze the work of new designs of sprayer boom suspensions in order to identify the effect of maximum stabilization of the boom to improve the quality of the process.

Analysis of available researches and publications. The problem of improving the quality of spraying crops in the literature sources is very important [1, 2], but there are few investigations where specific investigations of identifying the effectiveness of different types of suspensions for sprayers are carried out.

First of all, to model the operation of the sprayer boom, its kinematic perturbation, you need a model which realistically reproduce the agricultural field. Moreover, this model should be as simple as possible, in order not to complicate the dynamic model of oscillations of the rod mass on the suspension.

It is clear that field inequalities are random, as field preparation and technology of growing different crops form this agricultural background [3, 4]. But in order to identify the ability of the suspension to dampen vibrations, we can consider partial cases where the inequalities are regular, described by some harmonic function [5]. These assumptions are confirmed by other researchers of similar processes [6].

According to the analysis of a number of literature sources, particularly the works of Vikovich I. A., who studied the influence of field irregularities taking into account the elastic properties of soil, fluid fluctuations in the tank and other factors on the laws of kinematic perturbation of the rod in the trailed sprayer and proved that such perturbations can be represented in the form of simple harmonic laws [7].

Therefore we will consider that the sprayer moves with constant linear speed v_m and during time t passes the distance $x = v_m t$, and its wheels carry move in the vertical plane according to the law

$$z = z_p \cdot \sin \frac{\pi x}{a_p} \tag{1}$$

or let us rewrite (1) in the following way

$$z = z_p \cdot \sin \omega t , \tag{2}$$

where z_p is half the height of the field inequality;

a_p is the length of the wavelength of the field inequality;

ω is frequency characteristic of perturbation (circular frequency), $\omega = \frac{\pi v_m}{a_p}$;

Here the period of coercive force is $T_v = \frac{2\pi}{\omega}$, frequency is $f_v = \frac{\omega}{2\pi}$.

In order to reduce the cumbersomeness of the material, let us assume that the sprayer wheels move synchronously with field irregularities, and the resulting angular oscillations in the transverse-vertical plane are described by harmonic function in the following form

$$\psi(t) = \psi_p \cdot \cos \omega t , \tag{3}$$

where ψ_p is angular movement of the sprayer frame.

On the basis of the specified dependences we describe field irregularities on which the rod sprayer move.

The objective of the paper is to carry out analysis of the ability to dampen the vibrations of the rod by the developed suspension structure during the given typical mode of trailed sprayer operation.

Presentation of the main material. There are many patent designs of the sprayer boom suspensions by both domestic and foreign researchers. Among Ukrainian scientists, the theoretical foundations of the suspension and oscillations of the sprayer booms are thoroughly covered in the papers by Vikovych I. A., Diveieva B. M., Dmytrychenko M. F., Dorosh I. R. [7–9]. But there is a need for theoretical investigation of new developed suspens ions, which should meet the requirements for the level of rod stabilization.

A new design of the sprayer boom suspension which belongs to the four-link lever suspensions is proposed [10].

Let us consider the functionality of such suspension, Fig. 1.

Depending on the biological development of plants that are subjected to chemical protection, the hitch 5 is installed at a given height above the treatment surface. This installation operation is performed by vertically moving and fixing the suspension crossbar of the rod 2 on the frame of the rod 1. Then, while performing the technological process, the boom sprayer moves with field irregularities that disturb the oscillations of the hitch, in particular boom 5. Kinematic perturbation is transmitted from the sprayer chassis to the boom frame 1, where the crossbar and to the hitch 5. Kinematic perturbation is transmitted from the chassis of the sprayer to the frame of the rod 1, where the crossbar of the suspension rod 2 is fixed, and then through hinged pendulums 3, 4 and to the hitch rod 5. In order to minimize oscillations of the hitch rod 5 each of the pendulums 3, 4 for their free movement limitation is connected to independent links 6, 7 with elastic-damping elements (PDE), which absorb the load from the movement of the pendulums 3, 4 and dampen their free oscillations.

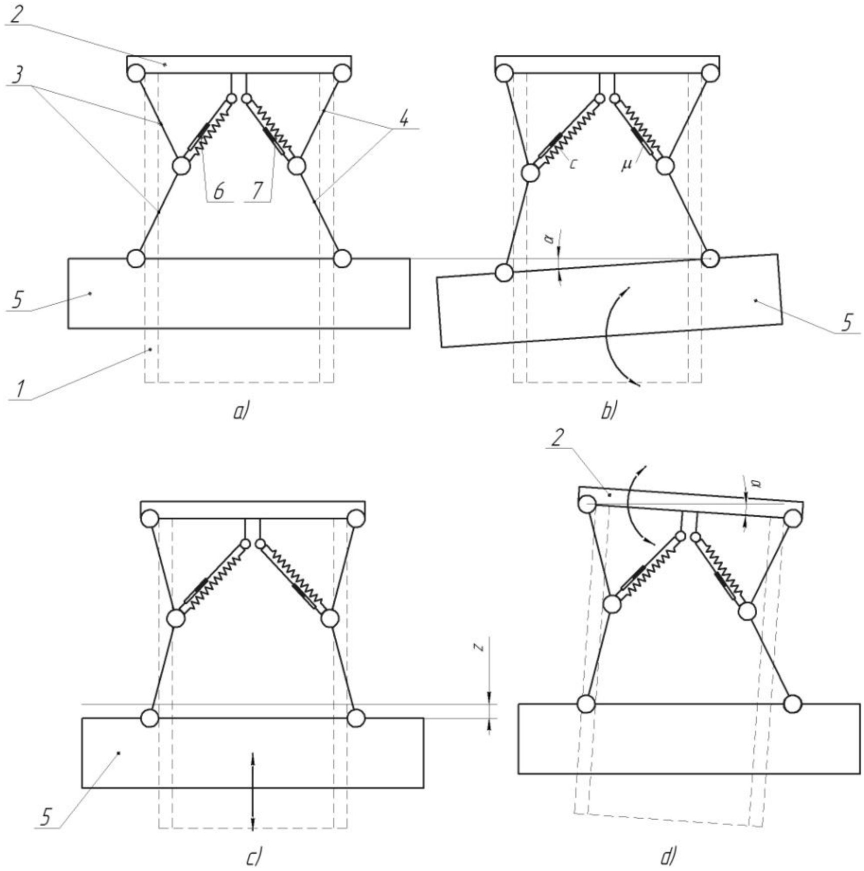

Figure 1. Functional diagrams of the independent suspension:
a – static equilibrium; b – operation of the suspension in case of transverse angular oscillations of the rod; c – the same in the case of vertical oscillations; d – operation of the suspension during kinematic perturbation of the hitch rod vibrations: 1 – rod frame; 2 – crossbar of the bar suspension; 3, 4 – pendulums; 5 – hinged rod; 6, 7 – independent sections with elastic-damping elements

Execution of spring-loaded sections 6, 7 in the form of two independent elastic-damping elements makes it possible for the suspension of the sprayer boom to be independent. This approach in this suspension structure allows to perceive different types of oscillations of the hitch 5. For example, due to kinematic perturbation there are angular oscillations of the hitch 5 (relative to the longitudinal axis of the unit), then such oscillations are perceived only by one pendulum 3 and are extinguished by independent link 6 with elastic-damping elements, or alternately with pendulum 4 and its independent link 7 with elastic-damping elements. In case of oscillations of the hitch rod 5 in the vertical plane, such suspension perceive and dampen the oscillations of both pendulums 3, 4 with their independent links 6, 7 with elastic-damping elements.

In the case of kinematic vibration perturbation, when, for example, one wheel of the sprayer hits the obstacle, the frame of the rod 1 deviates at angle α from the horizontal position together with the fixed crossbar of the suspension rod 2, respectively, this action is transmitted through pendulums 3, 4 to the hitch 5 which is in relative equilibrium, and due to the possibility of different deviations of the pendulums 3, 4, and for independent sections 6, 7 with elastic-damping elements, this is their different deformation and force interaction, such an action of the frame of the rod 1 is leveled, i.e. almost completely «absorbed» by independent elastic-damping elements.

It follows that with any kind of oscillations of the hitch rod in the transverse vertical plane, the developed structure of the sprayer boom suspension is able to dampen them, minimizing the negative impact on the design of the hitch rod. This effect will provide high vibration protection of the sprayer boom and as a result – it will improve the quality of application of the working preparation on the plant surface, increase the durability of the elements of the hitch and its suspension.

Kinematic connections between the sections of such a suspension are described in paper [11], and the application of chemically modified composite materials in the elements of movable joints that increase the life of such kinematic pairs is shown in investigations [12].

Therefore, based on the accepted laws of sprayer motion, causing kinematic perturbation of the mass of the rod, which is fixed to its suspension, we represent the differential equations of motion of the rod mass in order to determine the ability of the suspension to dampen such oscillations.

We assume that the mass of the rod, which is mounted on the suspension make vertical and angular oscillations in the transverse-vertical plane. Differential equations of its motion are as follows:

$$m_h \ddot{z}_h + 2c\left[\left(z_h - z_p \sin(\omega t)\right)\begin{pmatrix} az_p^2 \sin(\omega t)^2 - 2az_p z_h \sin(\omega t) - \\ - bz_p \sin(\omega t) + az_h{}^2 + bz_h + f \end{pmatrix}\right] +$$
$$+ 2\mu\left[\begin{matrix}\left[\dot{z}_h + \omega z_p\left(2\sin\left(\frac{\omega t}{2}\right)^2 - 1\right)\right]\times \\ \times\begin{pmatrix} 3az_p^2 \sin(\omega t)^2 - 6az_p z_h \sin(\omega t) - \\ - 2bz_p \sin(\omega t) + 3az_h^2 + 2bz_h + f \end{pmatrix}\end{matrix}\right] = 0; \quad (4)$$

$$I_h \ddot{\varphi}_h + 2cd\left[\begin{matrix} d\left(\varphi_h - \psi_p \cos(\omega t)\right)\times \\ \times\begin{pmatrix} ad^2\psi_p^2 \cos(\omega t)^2 - 2ad^2\psi_p\varphi_h \cos(\omega t) + ad^2\varphi_h^2 - \\ - bd\psi_p \cos(\omega t) + bd\varphi_h + f \end{pmatrix}\end{matrix}\right] +$$
$$+ 2\mu d \cdot \left[d\left(\dot{\varphi}_h + \omega\psi_p \sin(\omega t)\right)\left[\begin{matrix} 6a\left(2\sin\left(\frac{\omega t}{2}\right)^2 - 1\right)d^2\psi_p\varphi_h - \\ - 3a\left(\sin(\omega t)^2 - 1\right)d^2\psi_p^2 + 3ad^2\varphi_h^2 + \\ + 2b\left(2\sin\left(\frac{\omega t}{2}\right)^2 - 1\right)d\psi_p + 2bd\varphi_h + f \end{matrix}\right]\right] = 0, \quad (5)$$

here: m_h is the weight of the bar;

I_h is the moment of inertia of the rod in the transverse-vertical plane;

z_h is vertical movement of the rod;

φ_h is angular movement of the rod in the transverse-vertical plane;
c is stiffness coefficient of elastic elements of PDE;
μ is coefficient of viscous resistance of PDE;
d is the distance from the center of gravity of the concentrated mass of the rod on the suspension to the hinge of its suspension;
a, b, f are coefficients of the polynomial agreement function in the form

$$\Delta_z = az_h^3 + bz_h^2 + fz_h, \quad (6)$$

where – movement in PDE from the actual movement of the rod.

The obtained systems of equations describe the oscillations of the suspension, taking into account the nonlinear relationship between the movement of the mass of the rod and the deformation of the PDE.

The analytical solution of equations (4), (5) is too cumbersome and difficult to apply in practice, so we perform this procedure numerically using Mathcad software package.

Before proceeding with the solution, it is necessary to determine the coefficients of the matching function (6), taking into account the geometric and kinematic parameters of the suspension in question. Accordingly, we get: $a = 75{,}99$; $b = -8{,}37$; $f = 1{,}21$.

Then (6) is as follows

$$\Delta_z = 75{,}99 z_h^3 - 8{,}37 z_h^2 + 1{,}21 z_h .$$

Initial parameters for calculation are:

$m_h = 353$ kg; $I_h = 9235{,}8$ kg·m^2; $v_m = 12$ km/h (3,33 m/s); $c = 24000$ N/m; $\mu = 1200 \dfrac{Ns}{m}$; $d = 0{,}525$ m; parameters of field inequalities: $a_p = 1{,}66$ m, $z_p = 0{,}05$ m; initial conditions: at $t = 0$, $z_{0h} = 0$, $\dot{z}_{0h} = 0$, $\varphi_{0h} = 0$, $\dot{\varphi}_{0h} = 0$.

Let's present the main results of the solution. The vertical displacements of the mass of the rod mounted on the suspension are shown in Fig. 2.

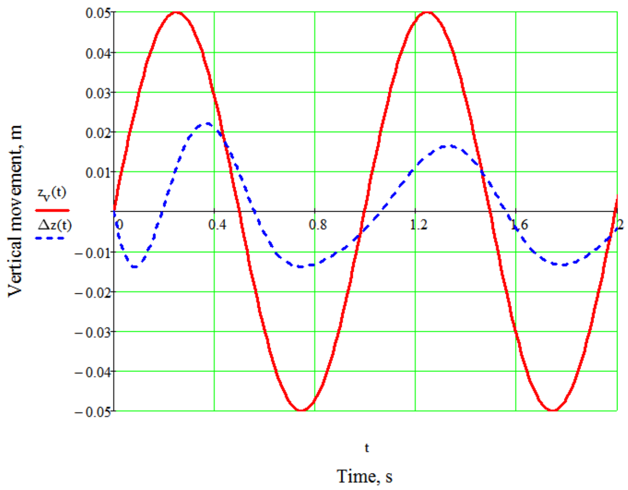

Figure 2. Vertical movement of the rod mass during the sprayer movement
solid line – the curve of movement of the sprayer by field irregularities;
dashed line – the relative movement of the mass of the rod relative to the object of processing

Then, let us consider the angular oscillations of the rod. The maximum angular displacements of the sprayer frame in the numerical implementation of the mathematical model are set to 3^0 (0.052 rad). If we consider that the rod is absolutely rigid and is 24 m long, and is fixed to the suspension rigidly, then for the edge of such beam the linear displacement is 0.612 m, which is quite large, and is unacceptable neither at performance of technological process, nor from the point of view of durability at disturbance of dynamic loadings, etc.

Thus the angular movement of the rod mass on the hitch is as follows, Fig. 3.

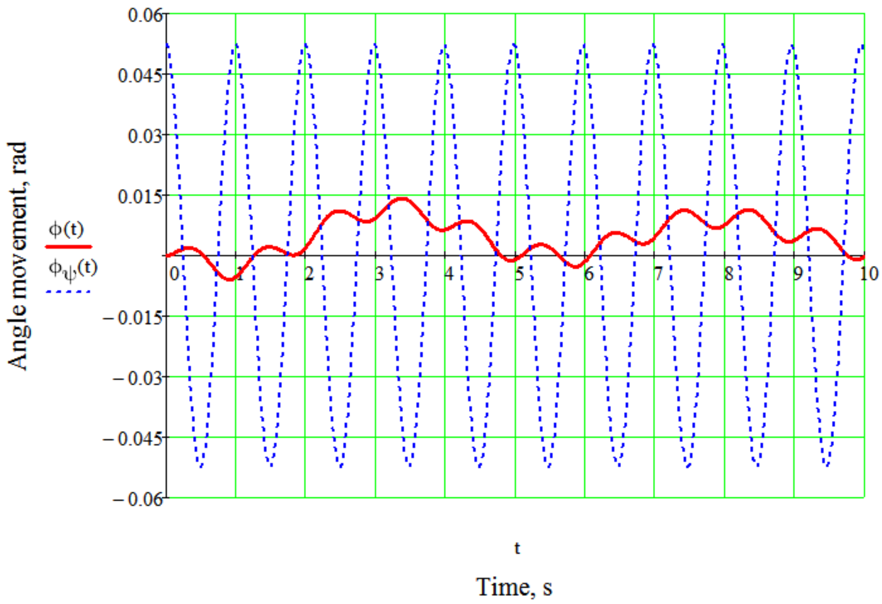

Figure 3. Angular movement of the rod mass on the suspension
solid line – the angular displacement of the rod mass; dotted line – angular displacements of the sprayer frame caused by field irregularities

It should be noted that the laws of sprayer motion on field obstacles are modeled in such a way as to verify the working capacity of the suspension in a fairly rigid operating mode.

Conclusions. Analysis (Fig. 2) shows that for this mode of sprayer movement in the first second of the movement on obstacles the relative movement of the rod mass on the suspension does not exceed 0.022 m, then such movement is within 0.018 m, which is quite acceptable for vertical oscillations pouring the working preparation on the treatment surface.

The graphic image (Fig. 3) shows that the sprayer frame disturbs the maximum angular oscillations of 0.052 rad with frequency of 1 Hz, and the rod mass on the hitch deviates by the angle of 0.015 rad during the first 5 s movement of the sprayer field irregularities, and then this amplitude decreases up to 0.011 rad. This means that there is a 3.5-fold and then 4.7-fold absorption of angular oscillations by the suspension. Taking the maximum angular displacement of the rod, for its extreme point, the linear displacement will be 0.175 m, and then – 0.129 m. In addition, there is another important aspect – the oscillation frequency of the fundamental harmonic of the rod is reduced by 5 times and is 0.2 Hz.

Thus, after analyses of the working capacity of the developed suspension of the sprayer boom, its high efficiency, which will significantly improves the quality of the sprayer as a whole is revealed.

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ДОСЛІДЖЕННЯ РОБОТОЗДАТНОСТІ ПІДВІСКИ РОБОЧОГО ОРГАНУ ФУНКЦІОНАЛЬНО-ТРАНСПОРТУЮЧОЇ МАШИНИ

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Резюме. *При вирощуванні практично будь-якої сільськогосподарської культури важко обійтися без її хімічного захисту, внесення рідких добрив тощо. Нові технології вимагають частіших та точніших доглядів. Тобто існує проблема забезпечення технологічного процесу вирощування культур високотехнологічними машинами, робочі органи яких були б надійними в експлуатації та забезпечували відносно точне дозування робочих препаратів. Якщо сказати про хімічний захист рослин, а також підживлення рідкими комплексними добривами, то найпоширенішими машинами, які забезпечують дану технологічну операцію, є штангові обприскувачі. Тут точність дозування робочого препарату та надійність роботи машини значною мірою залежить від штанги, яка виконана у вигляді несучого металевого каркасу та функціонального трубопроводу, де монтуються розпилюючі пристрої. Крім того, вся ця конструкція закріплена на спеціальній підвісці, яка повинна стабілізувати штангу у випадку виникнення різного роду механічних коливань при русі обприскувача польовими нерівностями.*

Тому в роботі досліджено нову конструкцію підвіски штанги обприскувача, яка повинна забезпечити значну стабілізацію штанги і тим самим підвищити якість нанесення робочого препарату на поверхні рослин та, знизивши амплітуди коливання штанги, зменшити динамічні сили на металокаркаси і тим самим збільшити її надійність та ресурс роботи.

Для реалізації поставленої мети виконано аналіз літературних джерел, де на основі отриманих результатів побудовано модель, яка дозволяє з належною точністю відтворити рельєф поверхні поля, яким рухається обприскувач, та створює кінематичне збурення, що викликає коливання каркасу штанги. Отримані залежності введено в загальну динамічну модель, що описує вертикальні та кутові коливання маси штанги, що навішана на конструкцію підвіски, яка розроблена співавторами. Після чисельного розв'язку рівнянь динамічної моделі коливань штанги отримано графічні залежності її кінематичних параметрів та виконано їх аналіз. Результати вказують на високу ефективність розробленої конструкції підвіски штанги, де спостерігається її висока стабілізація. Рівень амплітуд коливань знаходиться в межах відхилень, при яких розпилюючі органи штанги не порушують агротехнічні норми виливу робочого препарату на поверхні рослин.

Ключові слова: *коливання, штанга, обприскувач, динамічна модель, незалежна підвіска, вертикальне переміщення, кутове переміщення, польові нерівності.*

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