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Substantiation of consolidated inertial parameters of vibrating bunker feeder

***Abstract**. The article describes the method of installing the inertial parameters of a vibration bunker feeder. The emphasis is on the fact that for the complete calculation of the vibration bunker with propeller-shaped movement of bodies, implemented on the basis of hyperboloid torsion, it is necessary to know precisely the consolidated mass or the sum of the moment of inertia, which in the future are used to determine the coefficient of stiffness of the elastic node. For this purpouse we propose a method for establishing analytic dependencies for combined inertial parameters. The results were verified on a pilot sample of a vibration bunker feeder.*

***Streszczenie.** W artykule opisano sposób ustalania parametrów bezwładnościowych wibracyjnego podajnika bunkrowego. Nacisk położono na fakt, że do pełnych obliczeń bunkra wibracyjnego z ruchem ciał w kształcie śmigła, realizowanym w oparciu o skręcanie hiperboloidalne, konieczne jest dokładne poznanie masy skonsolidowanej lub sumy momentu bezwładności, które służą do wyznaczania współczynnika sztywności węzła elastycznego. Wyniki zweryfikowano na próbce pilotażowej wibracyjnego podajnika bunkrowego. (**Weryfikacja skonsolidowanych parametrów inercyjnych wibracyjnego podajnika bunkrowego**).*

Keywords: vibration bunker feeder, hyperboloid torsion, consolidated mass, combined moment of Interia.
Słowa kluczowe: wibracyjny podajnik bunkrowy, skręcanie hiperboloidalne, masa skonsolidowana, łączny moment bezwładności.

Introduction

Vibrating bunker feeders are commonly used to automate production processes. They are widely manufactured by different world enterprises: Spirol International Corporation, Moorfeed Corporation A Division of Executive Automation Systems Inc., NTN Corporation. [1-5]. Two main types of vibration bunker feeders can be distinguished: with directed (fig. 1) and independent (fig. 2) oscillations.

In vibration bunker feeders with independent (elliptical) oscillations, the direction of throwing is provided by force disturbance at two independent coordinates. Therefore, for determining the coefficients of stiffness of elastic nodes, we use well-known expressions for the consolidated mass at rectilinear oscillations or the consolidated moment of inertia of the system at angular oscillations [6-10].

a)

b)

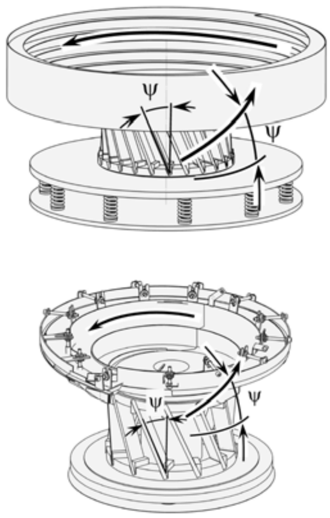


Fig. 1. Vibrating bunker feeders with guided (screw-like) oscillations in which the bowl is made cylinder-like (a) and a conical-like (b)

In vibration bunker feeders with propeller oscillations, the direction of throwing is ensured due to sloping elastic elements. In such structures, the propeller movement involves two: rectilinear and angular. Therefore, in order to determine the stiffness coefficient of an elastic node on the basis of a hyperboloid torsion, it is necessary to know the consolidated mass or the summed moment of inertia at propeller-shaped oscillations, which themselves contain, respectively, the proportion of the consolidated moment of inertia of the system at angular oscillations or the consolidated mass in rectilinear oscillations [11,12,16].

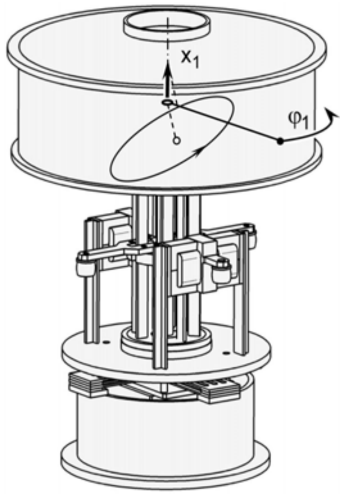


Fig. 2. Vibrating bunker feeder with independent (elliptical) oscillations

In the technical literature you can find methods for calculating vibration bunker feeders parameters. One of the defining parameters is the consolidated mass or the summed moment of inertia of the oscillatory system, the motion of which is carried out on a propellant trajectory. It is impossible to calculate the vibration bunker parameters precisely, without knowing these parameters [13,14,17].

Analysis of literary sources and problem statement

Thus, in the literature [6-10], analytical dependences are provided for establishing a consolidated mass or moment of inertia of a vibrational system, the movement – of which is carried out on a screw-driven trajectory. According to [6,18], the consolidated mass in such a motion is defined as

(1) $$m_{rht} = m_r \cos(\chi)\ \tan(\psi) + \frac{J_r}{z^2},$$

where:

(2) $$m_r = \frac{m_1 m_2}{m_1 + m_2}$$

is a consolidated mass during straight-line motion;

(3) $$J_r = \frac{J_1 J_2}{J_1 + J_2}$$

is a combined moment of inertia during angular oscillations m_1, m_2 and J_1, J_2 - respectively, the masses and moments of inertia of the first and second oscillatory bodies; z- resonant debugging; ψ - angle of tilt of elastic elements relative to the vertical plane; χ - angle, which is set according to the dependence

(4) $$\chi = \arcsin\left(\frac{l \sin\psi}{2R}\right),$$

where R – the radius of the location of the elastic elements of the hyperboloid torsion; l - the length of the elastic element.

According to [7], the mass is defined as

(5) $$m_{rht} = \sqrt{(m_r \sin\psi)^2 + \left(\frac{J_r}{R^2}\cos\psi\right)^2}.$$

In [8] the following expression is given

(6) $$m_{rht} = m_r\left(1 + \frac{r^2}{R^2 \tan^2\psi}\right),$$

where: r – the mass inertia radius m_1 (bowl).

According to [9]

(7) $$m_{rht} = m_r \sin^2\psi + \frac{J_r}{R^2}\cos^2\psi.$$

Therefore, there is no uniqueness in how to establish a consolidated mass of oscillatory system, the movement of which is carried out on a propellant trajectory.

Purpose and tasks of research

As you can see, there are inaccuracies in the analytical expressions for the consolidated mass and the consolidated moment of inertia of the oscillatory system in propeller-shaped motions, which makes it impossible to establish its parameters. In addition, in literary sources, there are practically no examples of calculations that would clearly demonstrate the application of the presented techniques, the use of which requires both experience and relevant knowledge [15,18,20,21].

Therefore, for the final establishment of analytical expressions, we will deduce their output. It requires:

1. To consider the basic schemes of two-mass oscillation systems based on an elastic node in the form of a hyperboloid torsion.
2. To make mathematical models of two-mass oscillation systems.
3. To establish by mathematical models analytical dependences for the consolidated mass and the consolidated moment of inertia of the oscillatory system, carrying out the propellant motion.
4. To confirm experimentally the obtained results.

Methods of establishing analytical dependences for the consolidated mass and moment of interia of the oscillatory system carrying the propellant motion

The formation of analytical dependences for the establishment of parameters of oscillatory systems is carried out by solving a system of linear differential equations provided that the transient processes have passed and the motion is established. For the compilation of differential equations of mathematical models of oscillatory systems, we use the energy balance method and the Lagrange equation of the second type. Experimental confirmation of the results is carried out in a natural way in laboratory conditions with the use of measuring equipment [17-19].

Method of determination of the consolidated mass and the moment of interia of the vibrational system, the movement of which is carried out on a screw-shaped trajectory

It is known that the coefficient of rigidity during tension (compression) through the consolidated mass for rectilinear motion (fig. 3) and the coefficient of rigidity for a system that performs angular (torsional) oscillations is determined based on expressions (2) and (3), namely:

(8) $$c = m_r\left(\frac{\Omega}{z}\right)^2 = \left(\frac{m_1 m_2}{m_1 + m_2}\right)\left(\frac{\Omega}{z}\right)^2;$$

(9) $$c_t = J_r\left(\frac{\Omega}{z}\right)^2 = \left(\frac{J_1 J_2}{J_1 + J_2}\right)\left(\frac{\Omega}{z}\right)^2,$$

where: Ω - the peak frequency of forced oscillations in the established operating modes. We note that the establishment of the coefficient rigidity value is possible due to the fact that we know the analytical expressions for the consolidated mass m_r and the consolidated moment of inertia J_r.

If we consider a two-mass system based on a hyperboloid torsion (fig. 4), an unknown parameter for us is the coefficient of stiffness of the elastic node. So, if the hyperboloid torsion is driven by the vertical force of diturbunce F, we do not know how to determine the stiffness c_{12} of compression (tension); if it twists with twisting moment M_t - unknown torsion stiffness c_{12t}.

How can we explaine it? The fact that in the first case it is not enough to determine the coefficient of rigidity only through the consolidated mass m_r, and in the second one only through a rotational moment of inertia J_r, since the oscillating masses are in the propellant motion. The formation of the value of the consolidated mass is influenced by the values of moments of mass inertia, and the combined moment of inertia of such a vibrational system includes the mass.

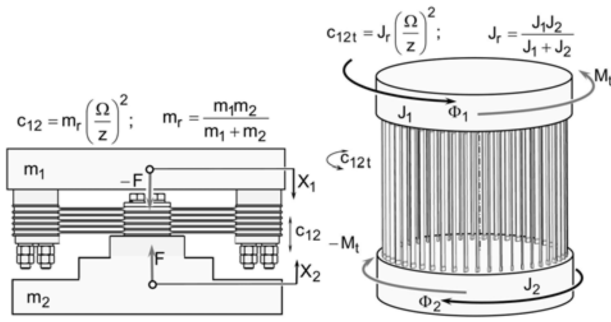
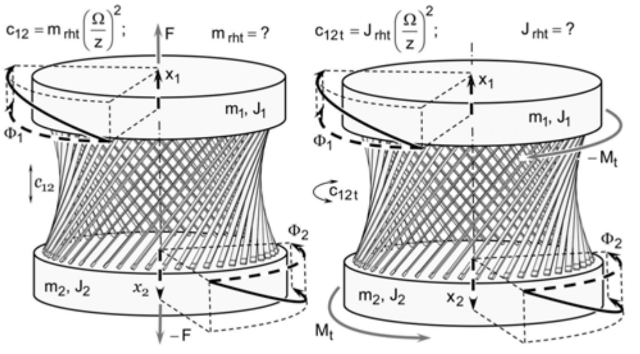


Fig. 3. Oscillation systems with rectilinear (a) and angular (b) movement

We analyze the two-mass vibration system of a vibrating bunker feeder which mass is driven by the vertical force of disturbance F (fig. 4a). For such a scheme, we need to know the expression for the consolidated mass m_{rht} so that according to the formula

(10) $$c_{12} = m_{rht}\left(\frac{\Omega}{z}\right)^2 ,$$

similar to (8), to establish the coefficient of hardness for compression (stretching) unknown to us.



a) b)

Fig. 4. An oscillation system based on a hyperboloid torsion, driven by the force of disturbance (a) and torque (b), and carries a propeller motion, where X_i and Φ_i , respectively, the amplitudes of bodies in linear and angular oscillations

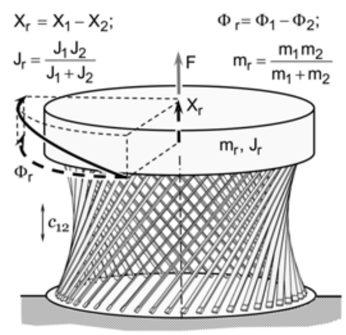


Fig. 5. An oscillation system based on a hyperboloid torsion, reduced to single weight

To simplify the presentation of the material, we reduce the two-mass system of the vibration bunker feeder to the single-mass (fig. 5), tightly gripping its lower mass, taking into account its inertia in the parameters of the upper mass. In essence, in motion there is one vibrational body which inertial parameters are determined according to (2) and (3).

Coordinates of relative motion x_r and ϕ_r are interconnected. One coordinate is set through another, according to the ratio

(11) $$x_r = \phi_r R \cos\alpha \tan\psi ,$$

that is, a generalized coordinate can only be either x_r and ϕ_r. To establish the consolidated mass in the propellant motion, we formulate an energy balance equation for a generalized coordinate x_r , reducing the rotational motion ϕ_r by linear displacement.

In the process of oscillations, the balance of kinetic K and potential Π energy, which can be represented by equality $\Pi = \mathrm{K}$, is always maintained. The potential energy under the deformation of the hyperboloid torsion by linear x_r will be

(12) $$\Pi = \frac{c_{12}\, x_r^2}{2} ,$$

and the kinetic energy will consist of motions in coordinates and will be

(13) $$\mathrm{K} = \frac{m_r\, \dot{x}_r^2}{2} + \frac{J_r\, \dot{\phi}_r^2}{2} .$$

Using (11) the angular velocity in coordinate φ_r is $\dot{\phi}_r = \dot{x}_r / (R\cos\alpha\ \mathrm{tg}\,\psi)$. Given this and equating (12) and (13), we obtain

(14) $$\frac{c_r\, x_r^2}{2} = \frac{m_r\, \dot{x}_r^2}{2} + \frac{J_r\, \dot{x}_r^2}{2R^2\cos^2(\chi)\tan^2(\psi)} .$$

Equation (14) in the amplitude values of displacement X_r and velocity ΩX_r will be written as

(15) $$c_{12}\, X_r^2 = m_r\, \Omega^2 X_r^2 + \frac{J_r\, \Omega^2 X_r^2}{R^2\cos^2(\chi)\tan^2(\psi)} ,$$

from where, taking into account resonance debugging z

(16) $$c_{12} = \left(m_r + \frac{J_r}{R^2\cos^2(\chi)\,\mathrm{tg}^2(\psi)}\right)\left(\frac{\Omega}{z}\right)^2 .$$

By comparing (10) and (16), we determine the consolidated mass of the oscillatory system based on the hyperboloid torsion, which works on compression under the action of the vertical force of disturbance F (diagram in fig. 4, a)

(17) $$m_{rht} = m_r + \frac{J_r}{R^2\cos^2(\chi)\,\mathrm{tg}^2(\psi)} = \\ = \frac{m_1 m_2}{m_1 + m_2} + \frac{J_1 J_2}{J_1 + J_2}\,\frac{1}{R^2\cos^2(\chi)\tan^2(\psi)} .$$

Using (16) and (17), the intrinsic frequency of the system based on the hyperboloid torsion, given through the consolidated mass will be

(18) $$\Omega_0 = \sqrt{\frac{c_{зг}}{m_{rht}}} = \sqrt{\frac{c_{12}}{\dfrac{m_1 m_2}{m_1 + m_2} + \dfrac{J_1 J_2}{J_1 + J_2}\,\dfrac{1}{R^2\cos^2(\chi)\tan^2(\psi)}}} .$$

For a vibrational system based on a hyperboloid torsion, driven by a torque M_t (scheme in fig. 4, b), the energy balance equation will look like

(19) $$\frac{c_{12r}\, \phi_r^2}{2} = \frac{J_r\, \dot{\phi}_r^2}{2} + \frac{m_r\, \dot{\phi}_r^2 R^2\cos^2(\chi)\tan^2(\psi)}{2} ,$$

where, carrying out the transformations are identical to the above, the combined moment of inertia

(20) $$J_{rht} = J_r + m_r R^2\cos^2(\chi)\tan^2(\psi) = \\ = \frac{J_1 J_2}{J_1 + J_2} + \frac{m_1 m_2}{m_1 + m_2} R^2\cos^2(\chi)\tan^2(\psi) .$$

So, the value of torsion c_{12r} at torsion moment is

(21) $$c_{12t} = (J_r + m_r R^2\cos^2(\chi)\tan^2(\psi))\left(\frac{\Omega}{z}\right)^2 ,$$

and the actual frequency of the system based on the hyperboloid torsion, fed through the summed moment of inertia is defined as

(22) $$\Omega_0 = \sqrt{\frac{c_{12t}}{J_{hrt}}} = \sqrt{\frac{c_{12t}}{J_r + m_r R^2\cos^2(\chi)\tan^2(\psi)}} .$$

We specified expressions for setting of the consolidated mass and moment of inertia. The combined mass for the hyperboloid torsion can also be established considering the two-tone design, without reducing it to single-mass (fig. 4a). To do this, we construct differential equations of free oscillations of the system by generalized linear coordinates x_1 and x_2 .

The kinetic energy of the system is given by (13)

$$\mathrm{K}=\frac{m_1\dot{x}_1^2}{2}+\frac{J_1\dot{\phi}_1^2}{2}+\frac{m_2\dot{x}_2^2}{2}+\frac{J_2\dot{\phi}_2^2}{2}=$$

(23) $$=\frac{m_1\dot{x}_1^2}{2}+\frac{J_1\dot{x}_1^2}{2R^2\cos^2(\chi)\tan^2(\psi)}+ ,$$
$$+\frac{m_2\dot{x}_2^2}{2}+\frac{J_2\dot{x}_2^2}{2R^2\cos^2(\chi)\tan^2(\psi)}$$

and potential energy

(24) $$\Pi=\frac{c_{12}(x_1-x_2)^2}{2}.$$

Let's introduce the designation $\Xi=R\cos(\chi)\tan(\psi)$. After performing the differentiation (23) and (24) and substituting the results of the well-known Lagrange equation of the second type, we obtain a system of differential equations

(25) $$\begin{cases}\ddot{x}_1\left(m_1+\dfrac{J_1}{\Xi^2}\right)+c_{12}(x_1-x_2)=0;\\ \ddot{x}_2\left(m_2+\dfrac{J_2}{\Xi^2}\right)+c_{12}(x_2-x_1)=0.\end{cases}.$$

The amplitudes of oscillations X_1 and X_2 and system (25) will look like

(26) $$\begin{cases}-X_1\Omega^2\left(m_1+\dfrac{J_1}{\Xi^2}\right)+c_{12}(X_1-X_2)=0;\\ -X_2\Omega^2\left(m_2+\dfrac{J_2}{\Xi^2}\right)+c_{12}(X_2-X_1)=0.\end{cases}.$$

The frequency matrix (matrix of coefficients for unknowns) of the system (26) has the form

(27) $$\begin{bmatrix}c_{12}-\Omega^2\left(m_1+\dfrac{J_1}{\Xi^2}\right) & -c_{12}\\ -c_{12} & c_{12}-\Omega^2\left(m_2+\dfrac{J_2}{\Xi^2}\right)\end{bmatrix}.$$

and its determinant

(28) $$\left(c_{12}-\omega^2\left(m_1+\frac{J_1}{\Xi^2}\right)\right)\left(c_{12}-\omega^2\left(m_2+\frac{J_2}{\Xi^2}\right)\right)-c_{12}^2 .$$

Equating (28) to zero, the actual frequency of the oscillatory system is

(29) $$\Omega_0=\sqrt{\frac{c_{12}}{\dfrac{\left(m_1+\dfrac{J_1}{\Xi^2}\right)\left(m_2+\dfrac{J_2}{\Xi^2}\right)}{\left(m_1+\dfrac{J_1}{\Xi^2}\right)+\left(m_2+\dfrac{J_2}{\Xi^2}\right)}}}=\sqrt{\frac{c_{12}}{m_{hrt}}},$$

and consequently, the consolidate mass in the propellant motion of bodies is

(30) $$m_{rht}=\frac{m_{r1}m_{r2}}{m_{r1}+m_{r2}},$$

where:

(31) $$m_{r1}=m_1+\frac{J_1}{\Xi^2};$$

(32) $$m_{r2}=m_2+\frac{J_2}{\Xi^2}.$$

Similarly, it is possible to deduce the summed moment of inertia for the circuit in fig. 4b. It is equal to

(33) $$J_{rht}=\frac{(m_1\Xi^2+J_1)(m_2\Xi^2+J_2)}{((m_1\Xi^2+J_1)+(m_2\Xi^2+J_2))}.$$

Experimental confirmation

Let`s consider a two-mass vibrating bunker feeder with an electromagnetic actuator, depicted in fig. 6. Its elastic knot is realized in the form of a hyperboloid torsion formed by three elastic elements of a circular cross-section, arranged in a circle in a radius R = 90 mm and at an angle $\psi=20^\circ$ relative to the vertical plane.

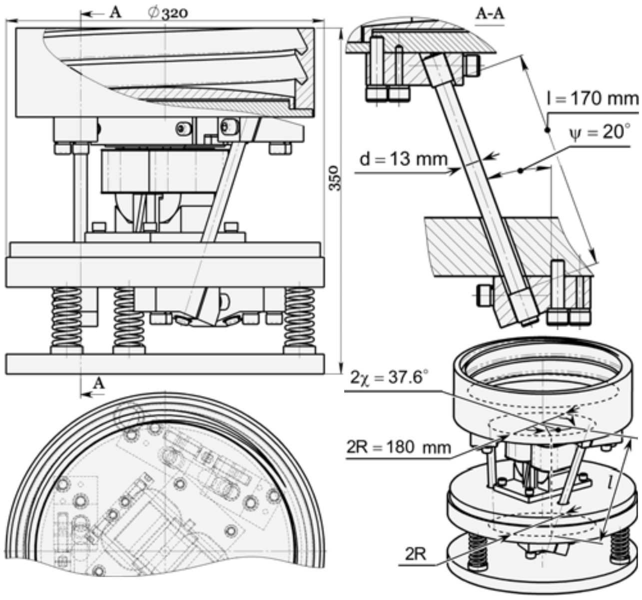


Fig. 6. General view of the vibration bunker Fedder

The movement in the vibrational system is carried out by the power disturbance in the vertical direction from the electromagnetic vibration exciter. In this case, the elastic knife is compressed with the appropriate rotation. Thus, when calculating the inertial parameter, we consider a straight-line motion, in which its degree of inertia is a mass. The inertial values of the bowl of the vibration bunker feeder have been established (fig. 7). For its dynamic equilibrium, a discoid reactive body is used (fig. 8), inertial parameters of which are also defined.

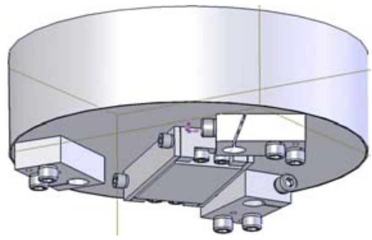


Fig. 7. Bowl, the mass of which is $m_1=22.8$ kg, and the moment of inertia relative to the axis of rotation $J_1=0.29\ \mathrm{kg\cdot m^2}$

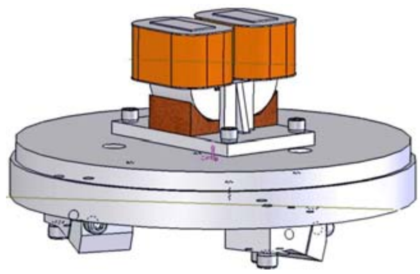


Fig. 8. The reactive mass, the mass of which is $m_2=34.4$ kg, and the moment of inertia relative to the axis of rotation $J_2=0.37\ \mathrm{kg\cdot m^2}$

Having previously assumed that the working length of the elastic elements l = 170 mm (fig. 6), and the average radius of their arrangement in a circle $R=90\ mm$, the angle χ according to (4) is

(34) $$\chi = \arcsin\left(\frac{l \sin\psi}{2R}\right) = \arcsin\left(\frac{0.17 \cdot \sin 20^\circ}{2 \cdot 0.09}\right) = = 0.33\ rad \equiv 18.8^\circ.$$

The combined mass of the oscillatory system based on the hyperboloid torsion is determined according to (30). Thus, the mass of the bowl m_{r1} according to (31)

(35) $$m_{r1} = m_1 + \frac{J_1}{R^2 \cos^2(\chi)\ \tan^2(\psi)} = = 22.8 + \frac{0.29}{0.09^2 \cdot \cos^2 18.8^\circ \cdot \tan^2 20^\circ} = 324.54\ kg.$$

and the mass m_{r2} of the disposable reactive body in accordance with (32)

(36) $$m_{r2} = m_2 + \frac{J_2}{R^2 \cos^2(\chi)\ \tan^2(\psi)} = = 34.4 + \frac{0.37}{0.09^2 \cdot \cos^2 18.8^\circ \cdot \tan^2 20^\circ} = 419.38\ kg.$$

According to (30), the mass of the oscillating system carrying the propellant motion is

$$m_{rht} = \frac{m_{r1} m_{r2}}{m_{r1} + m_{r2}} = \frac{324.54 \cdot 419.38}{324.54 + 419.38} = 182.96\ kg\ .$$

If we use the expression (17) for calculations, then we obtain

$$m_{rht} = \frac{m_1 m_2}{m_1 + m_2} + \frac{J_1 J_2}{J_1 + J_2} \frac{1}{R^2 \cos^2(\chi)\ \tan^2(\psi)} = = \frac{22.8 \cdot 34.4}{22.8 + 34.4} + \frac{\frac{0.29 \cdot 0.37}{0.29 + 0.37}}{0.09^2 \cdot \cos^2 18.8^\circ \cdot \tan^2 20^\circ} = 182.87\ kg.$$

Expression (17) is given in [10].

Compared to (17), the expression (30) is more precise and gives somewhat more value with the difference to 1%. Expressions (17) and (30) are completely convergent on boundary conditions:

$$\lim_{\psi \to 90^\circ} m_{rht} = \frac{m_1 m_2}{m_1 + m_2};\ \lim_{\psi \to 90^\circ} J_{rht} = \frac{J_1 J_2}{J_1 + J_2};$$
$$\lim_{\psi \to 0^\circ} m_{rht} = \lim_{\psi \to 0^\circ} J_{rht} \to \infty.$$

Let's start calculating the elastic node. As a material of elastic elements, we choose steel, the modulus of elasticity of the 1st and 2nd kind of which respectively are: $E = 2.1 \cdot 10^{11}$ Pa ; $G = 8.1 \cdot 10^{10}\ Pa$; . The coefficient of rigidity c_{12} on the compression (tension) of the hyperboloid torsion, assuming that the frequency of forced oscillations $\Omega = 314\ rad/s$, and the resonant adjustment $z = 0.98$, according to (10), is

$$c_{12} = m_{rht}\left(\frac{\Omega}{z}\right)^2 = 182.9 \cdot \left(\frac{314}{0.98}\right)^2 = 1.88 \cdot 10^7\ N/m.$$

The elastic node is realized on the basis of round elastic elements. Their number is $n = 3$. The diameter of the elastic element, set in accordance with [11], will be $d = = 0.0139$ mm $= 13$ mm .

Having made the necessary calculations of the elastic elements of the hyperboloid torsion for strength, which are not given, since this article is not the subject of this article, a vibrating bunker feeder was made (fig. 9), which provided the appropriate resonance mode of operation [12,14,15].

Fig. 9. Photo of a vibrating bunker feeder

Its own frequency of oscillations is $\Omega_0 = 320$ rad/s , which corresponds to the established resonant debugging $z \approx 0.98$. Therefore, we can state the correctness of the method of establishing the consolidated mass or the moment of inertia of the oscillatory system, the movement of which is carried out on a screw-shaped trajectory [13].

Conclusions

1. The method of establishing a consolidated mass or moment of inertia of a vibrational system, the motion of which is carried out on a propeller trajectory, is sufficiently clear, which allows us visually to follow the stages of the formation of these parameters.
2. Application during the creation of a vibration bunker feeder for the establishment of a consolidated mass of oscillatory system, the movement of which is carried out on a propeller trajectory, has allowed to provide the necessary resonant debugging of the system. It has confirmed the reliability of the presented analytical expressions.
3. Both dependencies for the establishment of the consolidated mass of the oscillatory system (expressions (17) and (30)) or its combined moment of inertia (expressions (20) and (33)), the motion of which is carried out on a propellant trajectory, can be used during engineering calculations, since they give practically the same result.

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