The influence of the vertical cantledge mass on the vibratory drum vibrations during the material compaction

V N Tarasov[1], I V Boyarkina[1], G N Boyarkin[2] and V S Serebrennikov[1]

[1]Siberian State Automobile and Highway University (SibADI), 5, Mira Pr., Omsk, 644080, Russia

[2]Omsk State Technical University, 11, Mira Pr., Omsk 644050, Russia

Abstract. The road roller vibratory drum is considered as a three-mass vibration system consisting of the eccentric weight mass m_1, vibratory drum mass m_2, additional mass m_3, which is transmitted to the vibratory drum axis through the roller frame. Differential equations of vertical vibrations were formulated, they were solved, analytical expressions were obtained of the amplitude A_1 of the vibrator drum vertical vibrations and of the amplitude A_2 of the additional mass m_3 vibrations, rigidly connected with the road roller frame. The dependences of the vertical vibration amplitudes A_1 and A_2 on the vibration frequency, mass ratio $(m_1 + m_2)$ and m_3, stiffness factors and viscous friction factors of the soil and rubber-metal shock absorbers have been established. The additional mass m_3 is considered as an additional shock absorber that protects the roller frame from vibration. The differential equation of the additional mass m_3 vertical vibrations has been obtained, which are excited by the harmonic inertia force. The analytical dependence conclusion for the determination of the dynamic factor K_d as the ratio of the dynamic and static vibration amplitudes of the additional mass m_3 of the vibratory drum have been specified. The specified dependence of the driving force transfer factor K determination on mass m_3 to the roller frame has been obtained. The diagrams of the dependence of K_d and K factors on the frequency ratio of forced and natural vibrations of the mass m_3 are constructed.
Key-words: vibratory drum, dynamic processes, differential equations, amplitudes of vertical vibrations, vibration protection.

1. Introduction

A modern vibration roller is an example of the beneficial use of vibration in technological processes of the material and soil compaction. The vibratory drum vertical vibration is combined with complex and energy-consuming dynamic processes of the mechanical energy transmission from the vibratory drum to the compacted material, in this case from the vibration from the vibratory drum is transmitted through rubber-metal shock absorbers to the roller frame and power units. The vibration roller studies take place in areas of improving the design, increasing quality and productivity of the compaction process, creating the intelligent control computer systems of the material and soil compaction modes by vibratory rollers etc. Fundamental studies of the vibratory roller interaction with deformable soil were performed by Russian scientists in the 1930's-60's.

In modern studies [1] the authors consider the influence of the soil mass, which is involved in the vibration process during its compaction, the vibratory drum and vibratory drum frame masses are taken into account.

In paper [2] the authors propose the compaction optimization method from the point of view of the process energy efficiency and productivity increase. The idea of the work consists in the taking into account the interaction of the compacted material layer with the vibratory drum.

A large number of studies is devoted to the compaction mode control, creation of computer intelligent control systems based on the analysis of a large number of factors characterizing dynamic processes [3–5].
Despite the huge number of publications devoted to vibration roller [6-11], there are no studies relative to the influence of the vibratory drum on the material compaction working process and roller frame protection from vibration.
In the reviewed papers the relationships of the vibratory drum vibration amplitudes and cantledge additional mass with the vibratory drum frame are not studied. There are no studies of the vibration roller frame vibration from the vibratory roller vibrations.

2. Problem statement

At present the vibratory drum vibration theory cannot explain many of the dynamic processes accompanying the vibratory drum working process.
The road roller vibratory drum of a medium type possessing the mass of 5-10 tons, performs vertical vibrations of the limited amplitude A_1= 1÷3 mm with a frequency of up to p= 80 Hz. In the technical documentation companies – manufacturer indicate the total mass of the road roller and distribution of this mass along the axes. However, the mass falling on the front vibratory drum includes the own mass of the vibratory drum, consisting of the shell mass, mass of shafts and hydraulic motors located inside the vibratory drum and additional mass transmitted vertically from the roller frame, where mechanisms and power unit are installed. The additional mass m_3 transmitted to the vibratory drum is connected by rubber-metal shock absorbers with the roller frame. In this paper the task of studying the influence of the additional mass, which is transmitted through the roller frame to the vibratory drum during the material and soil compaction.

3. Theory

The vibratory drum shown in Figure 1 consists of vibration exciter *1* mounted on the shaft inside vibratory drum *2*. The front frame is supported at the vibratory drum axis through which an additional vertical cantledge is transmitted on the vibratory drum from the roller mass in the form of conditional body *3* with the mass m_3, which is connected to the drum by rubber-metal shock absorbers.

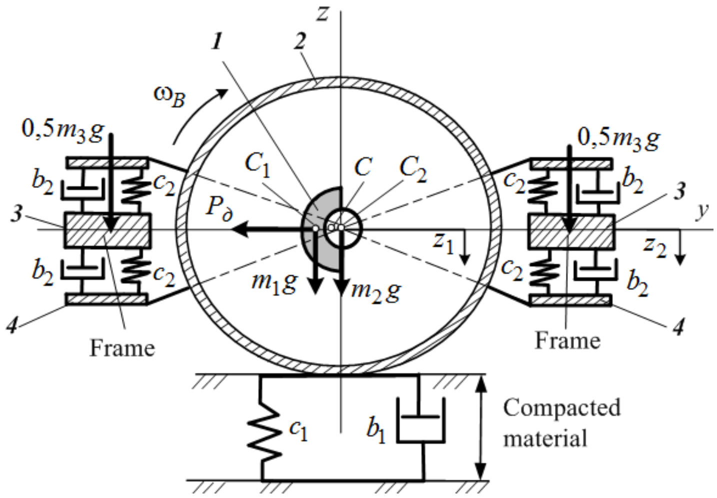

Figure 1. Simulation model of the vibratory drum interaction with the soil and roller frame.

The compacted medium is also an element of the mechanical system, which is characterized by the stiffness and viscous friction factor. The masses considered in Figure 1 perform vertical vibrations and ensure the material compaction over which the vibratory drum rolls.
With a discrepancy of less than 1% the sum of the masses (m_1 + m_2) can be considered as a single mass with the generalized coordinate z_1 and mass m_3 having the generalized coordinate z_2 in the coordinate system Oyz (see Figure 1).

The mechanical properties of the material to be compacted are characterized by the stiffness factor c_1 and viscous friction factor b_1, rubber-metal shock absorbers are characterized by parameters c_2 and b_2 respectively.
The mechanical system is affected by the radial harmonic force of the vibration exciter, which creates the vertical driving harmonic force on the vibratory drum determined according to the formula

$$P_{\partial z} = P_\partial \sin pt = m_1 r_1 p^2 \sin pt, \tag{1}$$

where m_1 – is unbalanced mass of the eccentric weight; r_1 – is the eccentricity of the eccentric weight; p – is the angular velocity of the vibration exciter rotation.
The equations of the linear mechanical system movement with harmonic forcing effects are known in the vibration theory [12, 13, 14].
The movement of the mechanical system in the coordinate system *Oyz* for the vibratory drum can be described by two differential equations

$$(m_1 + m_2)\ddot{z}_1 + (b_1 + b_2)\dot{z}_1 - b_2\dot{z}_2 + (c_1 + c_2)z_1 - c_2 z_2 = P_\partial \sin pt\,. \tag{2}$$

$$m_3\ddot{z}_2 + b_2(\dot{z}_2 - \dot{z}_1 + c_2(z_2 - z_1) = 0\,. \tag{3}$$

The solution of the system of differential equations (2), (3) can be performed in a complex form, i.e. to be presented as [15]

$$z_1 = A_1 e^{ipt}\,; \quad z_2 = A_2 e^{ipt}\,. \tag{4}$$

In equations (4) i – is an imaginary unit.
Inserting solution (4) into the differential equations (2), (3), we obtain the algebraic equations of the vibration amplitudes, containing the real and imaginary components

$$-(m_1 + m_2)p^2 A_1 + (b_1 + b_2)ipA_1 - b_2 ipA_2 + (c_1 + c_2)A_1 - c_2 A_2 = P_\partial\,. \tag{5}$$

$$-m_3 p^2 A_2 + b_2(ipA_2 - ipA_1) + c_2(A_2 - A_1) = 0\,. \tag{6}$$

In equations (5), (6) the components containing imaginary values i can be neglected, ase for real technical systems b_1 and b_2 are small values in these equations.
Therefore, equations (5), (6) with taking into account expression (1) allow to determine expressions for the amplitude A_1 of the vibratory drum vertical vibrations and amplitude A_2 of the additional mass by means of using the real components in equations (5), (6)

$$A_1 = \frac{m_1 r_1 p^2 (c_2 - m_3 p^2)}{((c_1 + c_2) - (m_1 + m_2)p^2)(c_2 - m_3 p^2) - c_2^2}\,. \tag{7}$$

$$A_2 = \frac{m_1 r_1 p^2 c_2}{((c_1 + c_2) - (m_1 + m_2)p^2)(c_2 - m_3 p^2) - c_2^2}\,. \tag{8}$$

4. Results discussion

Numerical experiments have been performed for the road roller with the total mass m=10320 kg, with the mass distribution on the vibratory drum (m_1 + m_2 + m_3)=5050 kg; eccentricity of the vibration exciter eccentric weight r_1 = 0.0655 m; eccentric weight mass m_1= 64.89 kg; declared eccentric weight rotation frequency p = 36.7 Hz, p = 230.59 rad/s.
In Figure 2 for the soil with the stiffness factor c_1 = 8890 kN/m the dependences of the vibratory drum vertical vibration amplitude A_1 on the frequency p of the forced vibrations with different mass ratios (m_1 + m_2) and m_3 are presented. For four dependencies the vibratory drum total mass constant value has been maintained, i.e. (m_1 + m_2 + m_3) = 5050 kg, at the same time the mass m_3 of the vertical eccentric weight and own mass (m_1 + m_2) of the vibratory drum was changing.

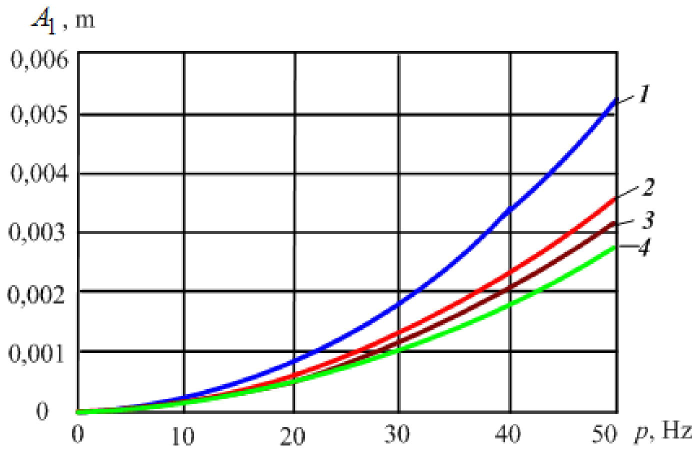

Figure 2. Dependence of the amplitude A_1 of the vibratory drum vertical vibrations on the forced vibration frequency p for different ratios of the masses of the vertical eccentric weight m_3 and vibratory drum mass $(m_1 + m_2)$:
1 – $(m_1 + m_2)$=2500 kg, m_3=2550 kg; *2* – $(m_1 + m_2)$=3550 kg, m_3=1500 kg;
3 – $(m_1 + m_2)$=4000 kg, m_3=1050 kg; *4* – $(m_1 + m_2)$ =4500 kg, m_3=550 kg.

The amplitude A_1 of the vibratory drum vertical vibrations depends on the driving force R_d, according to formula (1) raised to the second power on the eccentric weight rotation frequency. For different soils (see Figure 2) the amplitude A_1 of the vibratory drum vertical vibrations varies within the limits A_1 = 0 ÷ 0.0052 m. For Figure 2 we can make the conclusion that the amplitude A_1 of the vibratory drum vertical vibrations increases with increasing the frequency p and with increasing the vibratory drum mass $(m_1 + m_2)$.
In Figure 3 for the soil with the stiffness factor c_1 = 8890 kN/m, the dependences of the amplitude A_1 of the vibratory drum vibrations on the frequency of forced vibrations for the constant mass of the vibratory drum $(m_1 + m_2)$ = 3000 kg are presented. A numerical experiment has been performed to vary the mass values m_3 of the eccentric weight from the roller frame: m_3 = 150 kg; m_3 = 1000 kg; m_3 = 1500 kg; m_3 = 2050 kg.

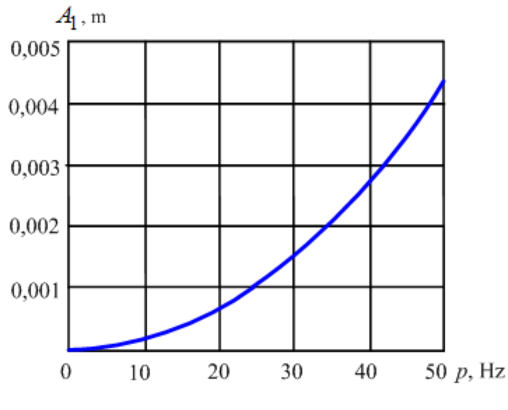

Figure 3. Dependence of the amplitude A_1 of the vibratory drum vertical vibrations on the the forced vibration frequency p.

At the same time it has been established that for different values of the masses m_3 these four dependences coincidence is observed. Consequently, the amplitude A_1 in the specified range of the mass changes doesn't depend on the mass m_3 of the vertical eccentric weight.

Figure 4 shows the dependences of the amplitudes A_1 and A_2 on the soil stiffness factor c_1, which varies within the range c_1 = 8890 ÷ 97222 kN/m. Initial data: frequency of forced vibrations p = 36.7 Hz; vibratory drum mass $(m_1 + m_2)$ = 4000 kg; vertical cantledge mass m_3 = 1050 kg.

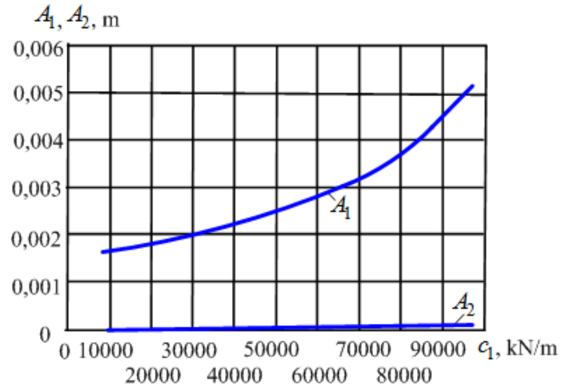

Figure 4. Dependence of the vibration amplitudes A_1 and A_2 on the soil stiffness factor c_1.

The amplitude A_1 of the vibratory drum vertical vibrations during this experiment varies within the range A_1 = 0.0017 ÷ 0.0052 m, at the same time the amplitude A_2 of the additional mass vertical vibrations remains a small value for the entire range of the soil strength change.

For two types of the soil Figure 5 shows the dependences of the vibration amplitudes A_1 and A_2 on the stiffness factor c_2 of rubber-metal shock absorbers. During the determination of the amplitude A_1, curve *1* corresponds to heavy soil with the stiffness factor c_1 = 97222 kN/m and curve *2* for light soil with the factor c_1 = 8890 kN/m.

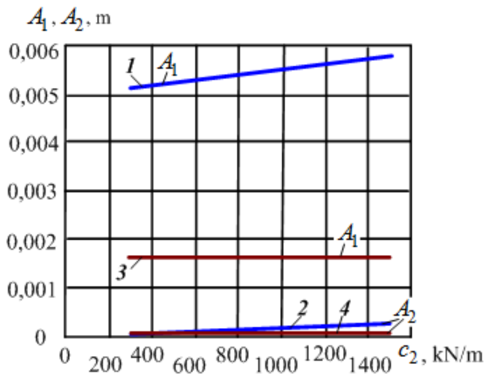

Figure 5. Dependences of the amplitudes A_1 and A_2 of vertical vibrations on the stiffness factor c_2 of rubber-metal shock absorbers for two categories of the soil: *1*, *3* – c_1 = 97222 kN/m; *2*, *4* – c_1 = 8890 kN/m.

The amplitude A_2 of the roller frame mass m_3 vibrations are of a small value for the considered range (see Figure 5, curves *3*, *4*).

The considered method has allowed to establish the dependence of the A_1 amplitude of the vibratory drum vertical vibrations on the vibratory drum mass and its parameters. The independence has been established of the amplitude A_1 of vibrations from the mass m_3 of the vibratory drum additional eccentric weigh. However, mass m_3 in Figure 1 for reasons of design cannot be equal to zero, as it depends on the vibration roller mass distribution across the vibratory drum axles and own mass.

In Figure 1 the vibratory drum is considered as a mechanical system performing vertical vibrations. However, the mass m_3 on this scheme as a structural element plays the role of a damper of mechanical vibrations of the vibratory drum frame. The mass m_3 acts as the roller frame shock absorber, i.e. protects it from the vibratory drum vibration.

With an inertial disturbing effect on the mass m_3 the following forces act: restoring force c_2z_2, viscous friction force $b_2\dot{z}_2$ and d'Alembert inertia force of this mass, brought into the order of target forces.
The equation of the mass m_3 vertical vibrations can be written as the basic equation of Newton dynamics for the material point [12, 13]

$$m_3\ddot{z}_2 = -c_2z_2 - b_2\dot{z}_2 + F(t), \tag{9}$$

where $F(t)$ – is the d'Alembert inertia force converted to the category of active Newtonian forces.
In this case, the inertia force of the additional mass m_3 is the harmonic force

$$F(t) = H\sin pt = -m_3\ddot{z}_2 \sin pt. \tag{10}$$

Equation (9) can be represented as an equation of forced vibrations for a material point.

$$\ddot{z}_2 + 2n_2\dot{z}_2 + \omega_2^2 z_2 = \frac{H}{m_3}\sin pt, \tag{11}$$

where H/m_3 – is the mass unit acceleration.
The values of equation (11) are determined by the formulas

$$n_2 = \frac{b_2}{2m_3}, \quad \omega_2^2 = \frac{c_2}{m_3}.$$

The solution of the differential equation of forced vibrations (11) with a harmonic forcing effect has the following form [12-16]

$$z_2 = \frac{H}{m_3\sqrt{(\omega_2^2 - p^2)^2 + 4n_2^2p^2}}\sin(pt - \delta_2), \tag{12}$$

where δ_2 – is the phase shift of the driving force and displacement

$$\mathrm{tg}\,\delta_2 = \frac{2n_2p}{\omega_2^2 - p^2}. \tag{13}$$

Let's determine the speed of body *2* as the time derivative of expression (12)

$$\dot{z}_2 = \frac{Hp}{m_3\sqrt{(\omega_2^2 - p^2)^2 + 4n_2^2p^2}}\cos(pt - \delta_2). \tag{14}$$

The reaction force R, transmitted by the mass m_3 with the help of rubber-metal connections to the frame is determined by the formula

$$R = -c_2z_2 - b_2\dot{z}_2. \tag{15}$$

Inserting (12) and (14) into equation (15) we'll obtain

$$R = -\frac{c_2H(\sin(pt - \delta_2) + (pb_2/c_2)\cos(pt - \delta_2))}{m_3\sqrt{(\omega_2^2 - p^2)^2 + 4n_2^2p^2}}. \tag{16}$$

From equation (16) we can distinguish the constant

$$K'_d = \frac{H}{m_3\sqrt{(\omega_2^2 - p^2)^2 + 4n_2^2p^2}}. \tag{17}$$

As per its physical essence the factor K'_d – is nothing but the dynamic amplitude A_2 of the m_3 mass vertical vibrations.

In the vibration theory it is customary to use the concept – the magnification factor, by which we understand the dynamic amplitude ratio to static amplitude, which is determined by the formula of paper [13]

$$A_{2cm} = H/c_2. \tag{18}$$

Using equation (17) we obtain the magnification factor in the following form

$$K_d = \frac{A_2}{A_{2cm}} = \frac{H}{m_3\sqrt{(\omega_2^2 - p^2)^2 + 4n_2^2 p^2}} \frac{c_2}{H}. \tag{19}$$

Considering that $c_2 = m_3\omega_2^2$, after the conversion of formula (19) we obtain an expression for determining the magnification factor

$$K_d = \frac{1}{\sqrt{\left(1 - \frac{p^2}{\omega_2^2}\right)^2 + \frac{4n_2^2 p^2}{\omega_2^4}}}. \tag{20}$$

By replacing in expression (16) $\frac{pb_2}{c_2}$ to $\frac{2n_2 p}{\omega_2^2}$ we determine the reaction force R in the following form

$$R = -K_d H\left(\sin(pt - \delta_2) + \frac{2n_2 p}{\omega_2^2}\cos(pt - \delta_2)\right). \tag{21}$$

Taking into account that R and H are of the same dimensionality, from equation (21) by appropriate transformations we can obtain the factor K of the transfer of the inertial force H transfer to the roller frame

$$K = \frac{R}{H} = K_d\sqrt{1 + \frac{4n_2^2 p^2}{\omega_2^4}}\sin(pt - \delta_2 + \varepsilon_2), \tag{22}$$

where $\operatorname{tg}\varepsilon_2 = \frac{2n_2 p}{\omega_2^2}$.

The maximum value of the transfer coefficient K is determined by the formula

$$K = K_d\sqrt{1 + \frac{4n_2^2 p^2}{\omega_2^4}}. \tag{23}$$

Formula (23) can be given a final form as a result of the transformation

$$K = \sqrt{\frac{1 + \frac{4n_2^2 p^2}{\omega_2^4}}{\left(1 - \frac{p^2}{\omega_2^2}\right)^2 + \frac{4n_2^2 p^2}{\omega_2^4}}}. \tag{24}$$

5. Consideration of the results

According to expression (20) Figure 6 shows the obtained dependences of the K_d factor on p/ω_2 for different parameters of the vibration roller.

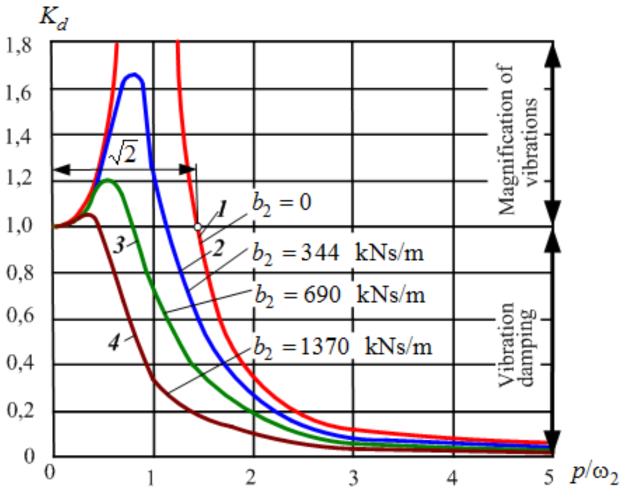

Figure 6. Dependences of the magnification factor K_d on p/ω_2 for different values of the viscous friction factor of rubber-metal shock absorbers.

When performing numerical experiments, the parameters of a medium-type vibration roller are used, the total mass of which is m = 10320 kg; vibratory drum has the mass $(m_1 + m_2 + m_3)$ = 5050 kg; eccentricity of the eccentric weight r_1 = 0.0655 m; additional mass of the cantledge m_3 = 2050 kg; frequency of forced vibration p = 36.7 Hz. Studies have been carried out for the following values of the viscous friction factor b_2: b_2 =0; b_2 =6 kNs/m; b_2 =344 kNs/m; b_2 =690 kNs/m; b_2 =1370 kNs/m.
Using Figure 6 there is a visual transformation of the vibration theory for which the magnification factor $K_d > 1$, to the theory of vibration protection for which the dynamic factor $K_d < 1$. The magnification factor K_d when solving problems of the vibration protection has a value less than the unity $K_d << 1$. For the medium-type roller, the magnification factor changes within the range K_d = 0.001 ÷ 0.005, which correspond to p/ω_2 = 27 ÷ 13.5. These values of K_d and p/ω_2 are outside the graph, but they explain the physical nature of the emergence of the vibration theory from the theory of oscillations (see Figure 6).
Figure 7 shows the dependences of the factor K of the dynamic transmission of the effect on the additional mass m_3 of the roller according to expression (24).

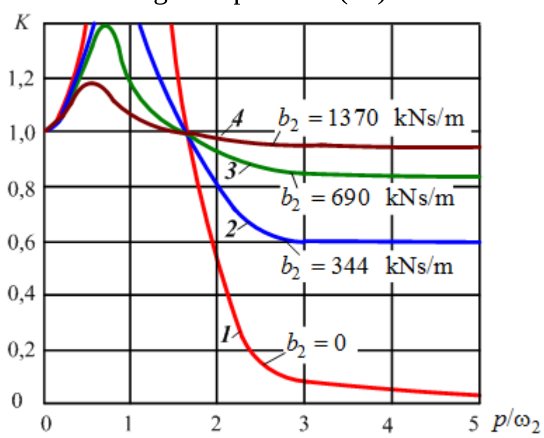

Figure 7. Dependence of the transfer factor K on the p/ω_2 ratio for the vibration roller different parameters.

Similarly Figure 6 and Figure 7 show the dependences of the factor K for different values of the viscous friction factor b_2. In the above resonance zone $p/\omega_2 > 1$ the curves intersect at one point with increasing p/ω_2. Curve *1* on this graph corresponds to the value $b_2 = 0$ and it is some kind of reference against which other options are compared.

In Figure 6 for the magnification factor K_d all the curves of the working area are located in the range of K_d = 0.001 ÷ 0.004 with the frequency p/ω_2 = 15 ÷ 30 Hz.
In Figure 7 the transfer factor curves K have uniformly occupied the area of possible values K=0.01÷1.0 for the corresponding values of the b_2 values of the viscous friction factor.
For a medium-type roller with the viscous friction factor of rubber-metal shock absorbers b_2 = 6 kNs/m the magnification factor K_d = 0.00333, transfer factor K of forcing effects from the mass m_3 to the roller frame is equal to K = 0.01316. This means that the mass m_3 is the vibration damper from the vibratory drum to the roller frame.

6. Conclusion

For the first time the studies of the vibratory drum three-mass mechanical system, containing the vibration exciter mass, vibratory drum mass and additional mass of the cantledge of the roller mass have been performed. As a result of solving the vibratory drum differential equation the dependence of the amplitude A_1 of the vibratory drum vertical vibrations on the soil characteristics and vibratory drum parameters has been established. New knowledge was obtained and expressions were improved for the determination of the magnification factor of an additional mass and transfer factor of the reaction driving force from the additional mass m_3.

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