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Research paper

Analytical Modeling of Oscillations of Movable Operating Elements of a Roll-Drum Activator

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Abstract

The article investigates the equipment for mechanical activation of building mixtures and the effect of the resulting oscillations on the activation efficiency and on the stability of the equipment operation. The activation of the entire building mixture with filler and wetted by finite amount of water can solve many problems that determine the quality of bricks or materials. Carried out researches of harmful oscillations of working of roll-drum activator (RDA) and their effects on the quality of processing. The own frequency of free oscillations arising in the RDA during its operation was determined experimentally. The possibility to stabilize the machine by installing shock absorbers has been examined. The influence of unwanted oscillations on the quality of products has been examined; the most optimal stabilization of the movement of "lever – roll" by using of additional external oscillations on the system has been defined. The analytical model of law of motion of the system with external oscillations and the parameters of its oscillations has been obtained. The parameters of oscillations of the roll has been determined; it was confirmed that the external oscillations have a positive effect on the efficiency of the activation and product quality.

Keywords: *Law of motion; Oscillation; Own frequency; Roll; Roll-drum activator.*

1. Introduction

The idea of activating materials to change their properties for use in different areas of production is popular [1-4]. However, activation of the entire mixture with filler and damp finite amount of water can solve many problems that determine the quality of building bricks or building materials: quality of blending of components, activation of the filler to the achievement of a more effective form, correction of grain size, etc. [5].

Staff of the department of mechanization of construction processes of Kharkiv National University of Construction and Architecture (KNUCA) designed roll-drum machine (Fig. 1), which is proposed to be used as a mixer-activator [6].

A significant problem of a roll-drum machine is the near-resonance oscillation that occurs during the machine's operation. Similar problems arise in many high-speed drum machines. For example, the eccentricity of centrifuges (that is caused by uneven load of the material and rotational frequency approaching the frequency of oscillations) makes the centrifuge operation resonant [7]. But studies based on the analysis of the unstable vibrational characteristics of the centrifuges make it possible to change the mode of operation to prevent oscillations of centrifuges. Traditionally, these control methods were passive. Recently active control of oscillations of centrifuges has appeared. This approach is the main way to prevent oscillations of centrifuges. There are two basic methods of the active control. The first one is to change the own frequency of the rotor centrifuges. There are many ways to change that frequency; a typical and effective method is to change the stiffness of an axle support of the rotor [7]. The second basic method is the use of an active control of the vibrational force.

We consider it most effective to create conditions for preventing formation of near-resonance oscillations.

2. Purpose and Tasks of the Research

The aim is to find a way of reducing oscillation arising during processing in a drum machine and propose appropriate measures to reduce the amplitude of oscillation.

To achieve the goal, the following tasks were set:

1. to know the mechanism and causes of the oscillation;
2. to determine the parameters of the oscillation and its influence on the efficiency of processing;
3. to propose ways to eliminate a harmful effect of high-frequency oscillation;
4. to create an analytical model of the impact of forced high-frequency oscillations on movable operating elements and processed material.

3. Basic Material and Results

Roll-drum activator (RDA, Fig. 1) of continuous action consists of a cylindrical drum 1, which is mounted on rollers 2. The drum 1 is rotating with supercritical velocity. A special drive is rotating it. A roll 3, which is able to roll along the inner surface of the drum 1, is located inside the drum 1. The roll 3 is pressed against the drum 1, for example by a lever and a pneumatic system 4. A knife 6 is pressed against the top of the drum. Loading and unloading of materials are carried out by trays 5 and 7. Pressing also can be carried out by a lever and a load.

Material (building mixture) is being loaded into the drum through tray 5 and is being rolled between the roll 3 and the drum 1. The material is undergoing mechanical compression, goes up along with the drum, is being cut off by a knife 6, and is falling again on the roll 3. This cycle is being repeated many times. The material is being moved along the drum by pressure and then getting unloaded through the tray 7. Pressure in the material under the roll 3 is significantly less than pressure in mills of similar design. The highest pressure in the material can be 1MPa.
BRA of periodic action is distinguished by the fact that one of the ends of the drum is closed by a wall. The drive is connected to this end coaxially (Fig. 2, 5, 6).

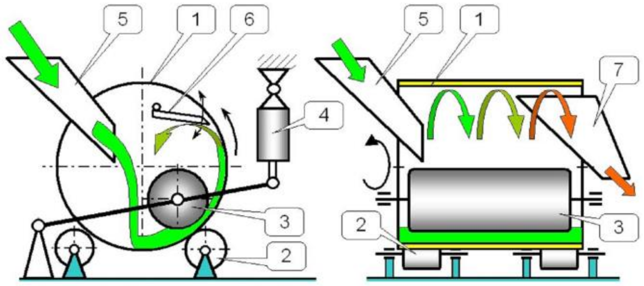

Fig. 1: Roll-drum activator (RDA) of continuous action with pneumatic pressing system: 1 - drum, 2 - rollers, 3 - roll, 4 - device for pressing the roll to inner surface of the drum, 5 - loading tray, 6 - knife, 7 - unloading tray.

Experiments were conducted on laboratory sample of RDA of periodic action. During experiments, it was observed that the main reason of lever oscillation is the roll, which is being rolled on layer of the material, compacting it. But at the same time it repeats all irregularities of the layer, creating obstacles on its way.
The main reason for this is that the material is unstable on the working surface of the drum due to differences in the adhesive properties of the drum surface. Also, the reason is the elastic properties of the material. This breaks the integrity of the layer of the material; the material partially falls under the roll before it meets the knife.
To measure the frequency and the amplitude of the oscillations of the system, a measuring system (Fig. 2) has been developed. It consists of a tenzobeam of rectangular cross section, strain gauge station, analog-to-digital converter [8] and computer.

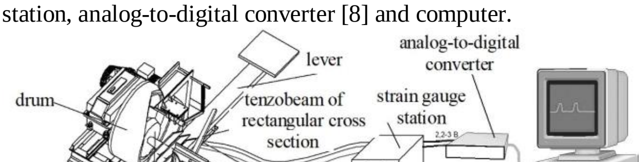

Fig. 2: Measuring system for the registration of oscillations of the system.

The oscillation period has been determined, and own frequency of the system has been calculated (own frequency is 2.78 oscillations per second, ω = 0.44 rad / s). The amplitude of the oscillations decreases during processing. At the same time, the frequency of the oscillations does not change significantly and roughly corresponds to own frequency of the system. The transition to the mode without oscillations does not occur.
One of the options to reduce the amplitude of the oscillations of the system "lever – roll" is the use of a hydraulic shock absorber in the pressing system. The objective was to find a possibility of using such system and to compare it in terms of the efficiency of activation with one of the pressing systems without absorber. The experimental technique includes the preparation of RDA to work by activating portions (that are slag mixture moistened up to 12 ÷ 14%) for 4-5 minutes with specific pressing force P = 80 kN/m.
A fragment of diagrams of lever oscillations are shown in Fig. 3. The "middle line" corresponds to the position of the roll in case of a uniform layer and absence of oscillations. The results of the measurements of the compression strength of the samples R_c, MPa, are listed in Table 1.

Table 1: Compressive strength of samples R_c,MPa, after 7 day hardening

time of processing, minutes	with shock absorber		without shock absorber	
	0.5min	1.5min	0.5min	1.5min
strength of brick sample, MPa	11.3	15	15	17.3

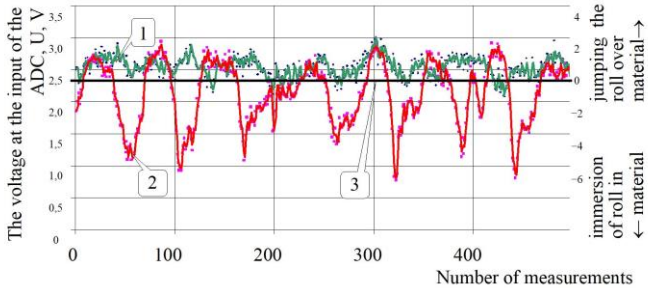

Fig. 3: Diagrams of lever oscillations: 1 - with a shock absorber; 2 - without a shock absorber; 3 – "middle line" (the position of the roll in case of a uniform layer and absence of oscillations).

The diagrams (see Fig.3) shows that the deviation above the "middle line" is much greater in the case of a shock absorber than without a shock absorber. Accordingly, contact time is less when using a shock absorber. This obviously leads to a decrease in the strength of the samples from the activated mixture when using a shock absorber.
Use of a hydraulic shock absorber significantly (three times) reduces the amplitude of the oscillation of the system (see Fig. 3) but significantly (by 13-25%) reduces the quality of the activation.
A promising way of stabilizing the motion of the system *"lever – roll"* is an imposition of an additional external oscillations on the system. In addition, these oscillations can also improve the efficiency of activation.
The main reason for the oscillations of the lever is the roll, which rolls along the layer of the material, pressing it. But at the same time, it repeats all irregularities of the layer, partially and periodically displaces the mixture in front of itself, creating an obstacle in its path.
A promising way of stabilizing the movement of the "lever – roll" system was the imposition of additional external oscillations on this system. In this case, such oscillations can provide an increase in activation efficiency. The mechanism of the action of high-frequency oscillations of the operating element on an uneven layer of the material is illustrated in the figure (Fig. 4).
Due to external oscillations, a part of the material is being destroyed; harmful oscillations are being eliminated.

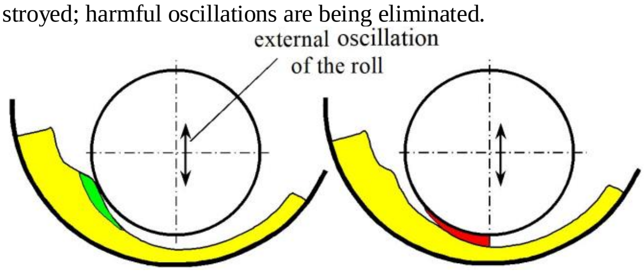

Fig. 4: Illustration of the "destruction" of a part of the material using external oscillation of the roll.

The purpose of the analytical research is to create an analytical model of the impact of forced high-frequency oscillations on movable operating elements and processed material (Fig. 5).
"S" is a coordinate defining the position of the roll 1 (see Fig. 5). An exciting force (a vibrator with an asynchronous motor) is being attached to the load 3 (Fig. 5). The system "*lever 2 – roll 1 – vibrator 3*" oscillates during operation of the machine about the axis 4. Drum 5 is mounted on rollers 6. The drum 5 is rotating with supercritical velocity from a special drive.

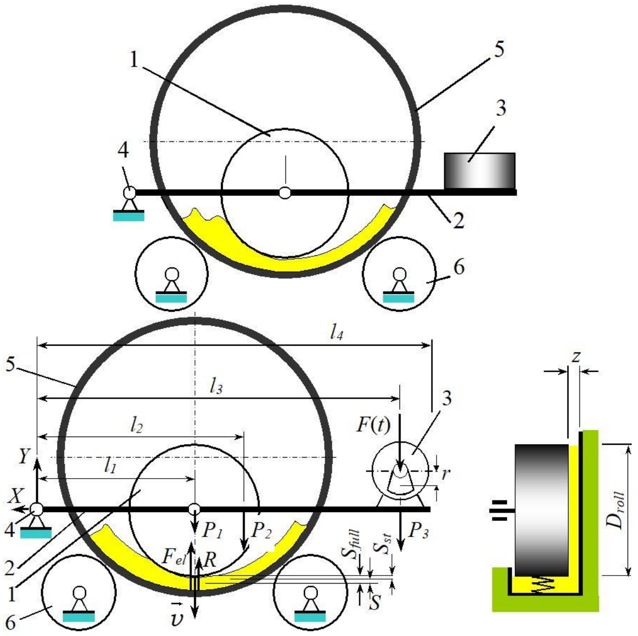

Fig. 5 The system "*lever 2 – roll 1 – vibrator 3*" with oscillates about the axis 4.

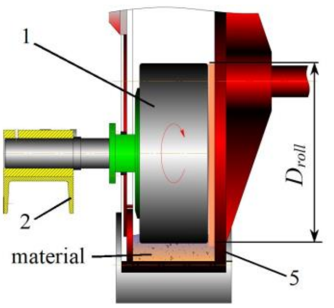

Fig. 6: Drum from inside.

Characteristics of the investigated system (Fig. 5):
m1, m2, m3 – masses of bodies of the mechanical system (P_1, P_2, P_3– weights of each of these bodies (the roll, the lever and the mounted vibrator)), l_1, l_2, l_3, l_4 – geometrical characteristics of the mechanism; *c* – stiffness of the layer of the material. The viscosity of the material being processed $\eta \approx (100-800)\text{Pa}\cdot\text{s}\cdot$

$$\mu = \eta \cdot \frac{\text{square}_{roll}}{z} = \eta \cdot \frac{\pi \cdot D_{roll}^2}{4 \cdot z}$$ – viscous resistance coefficient.

z – clearance between the end of the roll and the end of the drum;
p=300 rad/s – angular velocity of the forced oscillations;
$F_0 = m_{imbalance} \cdot p^2 \cdot r$ – exciting force of the vibrator.
The second fundamental theorem of dynamics of mechanical system was used (the work–energy theorem) [9, 10].
$\vec{P}_1, \vec{P}_2, \vec{P}_3$ – weights of the roll 1, the lever 2 and the vibrator 3, respectively (on the Fig. 5);
$\vec{F}_{el}$ – elastic force of the material, $\vec{X}, \vec{Y}$ – reaction of the bearing of the hinge of the lever, $\vec{R} = -\mu \cdot \vec{\upsilon}$ – force of viscous resistance, $\vec{F}(t)$ – exciting force of the vibrator.
The system has one degree of freedom. Position of an element of the system is determined by the coordinate "*S*" (Fig. 5).
The work–energy theorem can be written in the form:

$$\frac{dT}{dt} = \sum N^{e} + \sum N^{i}, \tag{1}$$

where *T* – kinetic energy of the system, $\sum N^{e}$ – sum of powers of external forces, $\sum N^{i}$ – sum of powers of internal forces.
The time derivative of the kinetic energy of the mechanical system is equal to algebraic sum of the internal and the external forces.
The kinetic energy of the system as the sum of the kinetic energies of the bodies (*lever, drum, roll, vibrator, rollers*):

$$T = T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7 = \frac{1}{2} J_{C1} \cdot \omega_2^2 + \frac{1}{2} J_{C2} \cdot \omega_2^2 + \frac{1}{2} J_{C3} \cdot \omega_2^2 +$$
$$+ \frac{1}{2} J_{drum} \cdot \omega_{drum}^2 + \frac{1}{2} J_{roll} \cdot \omega_{roll}^2 + \frac{1}{2} J_{vibrator} \cdot \omega_{vibrator}^2 + 2\frac{1}{2} J_{roller} \cdot \omega_{roller}^2 =$$
$$= \frac{1}{2}\omega_2^2 (J_{C1} + J_{C2} + J_{C3}) +$$
$$+ \frac{1}{2}\left(J_{drum} \cdot \omega_{drum}^2 + J_{roll} \cdot \omega_{roll}^2 + J_{vibrator} \cdot \omega_{vibrator}^2 + 2 \cdot J_{roller} \cdot \omega_{roller}^2\right) \tag{2}$$

where ω_2 – angular velocity of oscillations of the lever 2 (Fig. 5) about the axis 4;
J_{C1}, J_{C2}, J_{C3} – moments of inertia of the roll 1, the lever 2 and the vibrator 3 about the axis 4, respectively (Fig. 5);
ω_{drum} – angular velocity of rotation of the drum 5 around its axis;
ω_{roll} – angular velocity of rotation of the roll 1 around its axis;
$\omega_{vibrator}$ – angular velocity of the shaft of the vibrator 3 around its axis;
ω_{roller} – angular velocity of rotation of one of the rollers 6 around its axis;
J_{drum} – moment of inertia of the drum 5 around its axis;
J_{roll} – moment of inertia of the roll 1 around its axis;
$J_{vibrator}$ – moment of inertia of the shaft of the vibrator 3 around its axis;
J_{roller} – moment of inertia of one of the rollers 6 around its axis.
Expression for speed of the roll 1 around axis 4:

$$\omega_2 = \frac{\upsilon}{l_1} = \frac{1}{l_1}\upsilon \; ; \; \omega_2^2 = \frac{1}{l_1^2}\upsilon^2 \cdot \tag{3}$$

Substituting (3) into (2) gives:

$$T = \frac{1}{2} m_{pr} \upsilon^2 = \frac{1}{2} m_{pr} \dot{S}^2 , \tag{4}$$

where $m_{pr} = \dfrac{J_{C1} + J_{C2} + J_{C3}}{l_1^2}$.

The value $m_{pr} = const$ is called «***pr**esent mass*» (reduced mass) [9, 10, 11].
Also,
$\left(J_{drum} \cdot \omega_{drum}^2 + J_{roll} \cdot \omega_{roll}^2 + J_{vibrator} \cdot \omega_{vibrator}^2 + 2 \cdot J_{roller} \cdot \omega_{roller}^2\right) = const$,
since all components of this expression are determined constructively, they do not change during the oscillations of the system around axis 4. The time derivative of this expression is equal to zero.
The time derivative of kinetic energy [9, 10, 11]:

$$\frac{dT}{dt} = \frac{1}{2} \cdot 2 \cdot m_{pr} \dot{S}\ddot{S} = m_{pr} \ddot{S} \cdot \dot{S} , \tag{5}$$

where $m_{pr}\ddot{S}$ – force, $\ddot{S}$ – acceleration, $\dot{S}$ – speed.
Consider the right-hand side of the equation (1). Expression for power:

$$N = \vec{F} \cdot \vec{\upsilon} = F \cdot \upsilon \cdot \cos\left(\vec{F}, \vec{\upsilon}\right) . \tag{6}$$

Put the mechanical system does not change. That is, a body that enters the system, doesn't deform and speeds of its points with respect to each other are equal to zero. Then the sum of powers of all internal forces is equal to zero:

$$\sum N^i = 0 \cdot \qquad (7)$$

If velocity of the points is equal to zero, then the sum of powers of external forces at these points is also equal to zero.
The sum of powers of other forces:

$$\sum N^в = -\vec{F}_{el} \cdot \vec{\upsilon} - \vec{R} \cdot \vec{\upsilon} + \vec{P}_1 \cdot \vec{\upsilon} + \vec{P}_2 \cdot \vec{\upsilon} \cdot \frac{l_2}{l_1} + \vec{P}_3 \cdot \vec{\upsilon} \cdot \frac{l_3}{l_1}; \qquad (8)$$

or $\sum N^в = F_{pr} \cdot \upsilon$, $\sum N^в = F_{pr} \cdot \dot{S}$, (9)

where $F_{pr} = -F_{el} - R + P_1 + P_2 \frac{l_2}{l_1} + P_3 \frac{l_3}{l_1} + F(t) \frac{l_3}{l_1} \cdot$ (10)

Call the value F_{pr} a «*pr*esent force» (reduced force) [9, 10, 11].
Elastic force (in the expression (10)) is proportional to elastic deformation of the material. Full deformation of layer S_{full} is equal to the sum of static S_{st} and dynamic S deformations:

$$S_{full} = S_{st} + S, \qquad (11)$$

The elastic force is equal to:

$$F_{el} = c \cdot S_{full} = c(S_{st} + S). \qquad (12)$$

Force of viscous resistance: $R = \mu \cdot \upsilon = \mu \cdot \dot{S}$.
Present force is equal to zero in the quiescent. Given that $S = 0;\ \dot{S} = 0; F(t) = 0$, we obtain an equation of state of the system. The equation determines the static deformation of the material:

$$S_{st} = \frac{1}{c}\left(P_1 + P_2 \frac{l_2}{l_1} + P_3 \frac{l_3}{l_1}\right). \qquad (13)$$

Substituting the expression for the time derivative of kinetic energy (5) and the expression for the sum of powers of forces (9) into equation (13) gives differential equation of the system. Divide all terms of the equation by m_{pr}:

$$\ddot{S} + \frac{c}{m_{pr}} \cdot S + \frac{\mu}{m_{pr}} \cdot \dot{S} = \frac{1}{m_{pr}} F(t) \frac{l_3}{l_1} \cdot \qquad (14)$$

$k = \sqrt{\frac{c}{m_{pr}}}$ – circular angular velocity of own oscillations,

$2 \cdot n = \frac{\mu}{m_{pr}}$ – index of attenuation of oscillations.

The initial conditions for motion:

$$t = 0| \quad S = S_0, \quad \dot{S} = \dot{S}_0 \cdot \qquad (15)$$

The differential equation (14) has been integrated. Force of the vibrator 3 (see Fig. 5) varies harmonically:

$$F = F_0 \sin(pt), \qquad (16)$$

where F_0 – amplitude of the exciting forces;
p – angular velocity of oscillations.

The general solution S of the inhomogeneous differential equation (14) is the general solution of the homogeneous equation S_{hom} and a particular solution of the inhomogeneous equation: $S = S_{hom} + S_{part}$. Homogeneous differential equation corresponding to a given inhomogeneous equation is equal to:

$$\ddot{S} + 2 \cdot n \cdot \dot{S} + k^2 \cdot S = 0. \qquad (17)$$

Solution of this equation is equal to:

$$S = A \cdot e^{\lambda t}, \qquad (18)$$

where A and λ – undefined constants.
Substituting (18) into (17) gives:

$$\lambda^2 + 2 \cdot n \cdot \lambda + k^2 = 0. \qquad (19)$$

The equation (19) is a characteristic equation of the differential equation (17); it has two roots:

$$\lambda_{1,2} = -n \pm \sqrt{n^2 - k^2}. \qquad (20)$$

The roots of the characteristic equation may be complex conjugate or real.
In the first case ($n < k$), the general solution of the equation (17) is equal to:

$$S = \left(A_1 \cdot e^{ik_1 t} + A_2 \cdot e^{-ik_1 t}\right) \cdot e^{-nt}, \qquad (21)$$

where A_1, A_2 – constants of integration,

$$k_1 = \sqrt{k^2 - n^2}. \qquad (22)$$

Using the well-known Euler's formula, the *hom*ogeneous equation gives the solution of the equation (21):

$$S_{hom} = a \cdot e^{-nt} \cdot \sin(k_1 t + \beta), \qquad (23)$$

where a, β – constants of integration.
In the second case ($n = k$), the general solution is equal to:

$$S = e^{-nt} \cdot (A_1 + A_2 \cdot t). \qquad (24)$$

In the third case ($n > k$), the general solution is equal to:

$$S = e^{-nt} \cdot \left(A_1 \cdot e^{ik_2 t} + A_2 \cdot e^{-ik_2 t}\right). \qquad (25)$$

where $k_2 = \sqrt{n^2 - k^2}$.
The circular angular velocity of own oscillations and the index of attenuation of oscillations have been calculated.
Determine, which value (n or k) is greater.
Calculate the particular solution of the inhomogeneous differential equation:

$$\ddot{S} + 2 \cdot n \cdot \dot{S} + k^2 \cdot S = \frac{F_0 \sin(p \cdot t)}{m_{pr}} \cdot \frac{l_3}{l_1}. \qquad (26)$$

The *part*icular solution is equal to:

$$S_{part} = A \cdot \sin(p \cdot t) + B \cdot \cos(p \cdot t). \qquad (27)$$

Substituting (27) into (26) and subsequent transformation of the system of equations in order to determine the constants A and B, gives:

$$A=\frac{\left(k^2-p^2\right)}{\left(k^2-p^2\right)^2+4\cdot n^2\cdot p^2}\cdot\frac{F_0}{m_{pr}}\cdot\frac{l_3}{l_1},$$
$$B=\frac{2\cdot n\cdot p}{\left(k^2-p^2\right)^2+4\cdot n^2\cdot p^2}\cdot\frac{F_0}{m_{pr}}\cdot\frac{l_3}{l_1}. \quad (28)$$

Summing (23) and (27) gives the general solution of the inhomogeneous equation (26):

$$S=a\cdot e^{-nt}\cdot\sin(k_1t+\beta)+A\cdot\sin(pt)+B\cdot\cos(pt). \quad (29)$$

The constants a and β can be determined from the initial conditions (15). In order to do this, a system of equations for the unknown constants has been obtained by equating the time derivative of (29) to initial conditions:

$$S_0=a\cdot\sin(\beta)+B,$$

$$\dot{S}_0=a\cdot[-n\cdot\sin\beta+k_1\cdot\cos\beta]+A\cdot p.$$

Solving this system gives:

$$a=\sqrt{(S_0-B)^2+\frac{1}{k_1^2}\cdot\left(\dot{S}_0+n\cdot S_0-n\cdot B-A\cdot p\right)^2},$$
$$tg\beta=\frac{k_1\cdot(S_0-B)}{\dot{S}_0+n\cdot S_0-n\cdot B-A\cdot p}. \quad (30)$$

Substituting angle β into (29) gives the law of motion of the mechanism.

4. Example of Implementation of Dependence

An example of a solution for a version of RDA (Fig. 5).
p=300 rad/s; $F_0=m_{imbalance}\cdot p^2\cdot r=1\cdot 300^2\cdot 0{,}07=6300$N;
l_4 = 1,3m; l_3 = 1,2m; l_2 = 0,7m; l_1 = 0,5m;
P_3 = 100N – weight of the vibrator;
P_2 = 200N – weight of the lever;
P_1 = 300N – weight of the roll.

$J_{C1}=7{,}5\text{kg}\cdot\text{m}^2$,

$J_{C2}=11{,}267\text{kg}\cdot\text{m}^2$,

$J_{C3}=14{,}4\text{kg}\cdot\text{m}^2$.

$$m_{pr}=\frac{J_{C1}+J_{C2}+J_{C3}}{l_1^2}=132{,}7\text{kg}.$$

$$c=1\cdot 10^6\,\frac{\text{N}}{\text{m}}.$$

Force of viscous resistance: $R=\mu\cdot\upsilon=\mu\cdot\dot{S}$.

Viscosity $\eta\approx 800\text{Pa}\cdot\text{s}$; $\mu=\eta\cdot\frac{\text{square}_{roll}}{z}=\eta\cdot\frac{\pi\cdot D_{roll}^2}{4\cdot z}$ – viscous resistance coefficient. z – clearance between the end of the roll and the end of the drum (for example, z = 0,01м); D_{roll} = 0,3m.

$$\mu=800\cdot\frac{\pi\cdot 0{,}3^2}{4\cdot 0{,}01}=5655\left(\text{Pa}\cdot\text{s}\cdot\frac{\text{m}^2}{\text{m}}=\frac{\text{N}}{\text{m}^2}\cdot\text{s}\cdot\text{m}=\frac{\text{s}}{\text{m}}\cdot\text{N}\right).$$

$k=\sqrt{\frac{c}{m_{\text{pr}}}}=\sqrt{\frac{10^6}{132{,}7}}=86{,}82\frac{1}{\text{s}}$ – circular angular velocity of own oscillations of the system,

$n=\frac{\mu}{2\cdot m_{pr}}=\frac{5655}{2\cdot 132{,}7}=21{,}3\frac{1}{\text{s}}$ – index of attenuation of oscillations.

In this case, $n<k$.

Coefficients A and B are equal to:

$$A=\frac{\left(k^2-p^2\right)}{\left(k^2-p^2\right)^2+4\cdot n^2\cdot p^2}\cdot\frac{F_0}{m_{pr}}\cdot\frac{l_3}{l_1};\ A=-1{,}35\cdot 10^{-3},$$

$$B=\frac{2\cdot n\cdot p}{\left(k^2-p^2\right)^2+4\cdot n^2\cdot p^2}\cdot\frac{F_0}{m_{pr}}\cdot\frac{l_3}{l_1};\ B=2{,}093\cdot 10^{-4},$$

$$k_1=\sqrt{k^2-n^2}=\sqrt{86{,}82^2-21{,}3^2}=84{,}163\frac{1}{\text{s}}.$$

For initial conditions ($S_0=0;\quad \dot{S}_0=0$), constants a and β are equal to:

$$a=\sqrt{(S_0-B)^2+\frac{1}{k_1^2}\cdot\left(\dot{S}_0+n\cdot S_0-n\cdot B-A\cdot p\right)^2}=4{,}76\cdot 10^{-3},$$

$$tg\beta=\frac{k_1\cdot(S_0-B)}{\dot{S}_0+n\cdot S_0-n\cdot B-A\cdot p};\ \beta=-0{,}044.$$

The law of motion of the mechanism:

$$S=a\cdot e^{-nt}\cdot\sin(k_1t+\beta)+A\cdot\sin(pt)+B\cdot\cos(pt);$$

$$S=4{,}76\cdot 10^{-3}\cdot e^{-21{,}3t}\cdot\sin(84{,}16\cdot t+(-0{,}044))+$$
$$+\left(-1{,}35\cdot 10^{-3}\right)\cdot\sin(300\cdot t)+2{,}09\cdot 10^{-4}\cdot\cos(300\cdot t).$$

Diagrams of the law of motion for this version of RDA are shown in Fig. 7, 8.

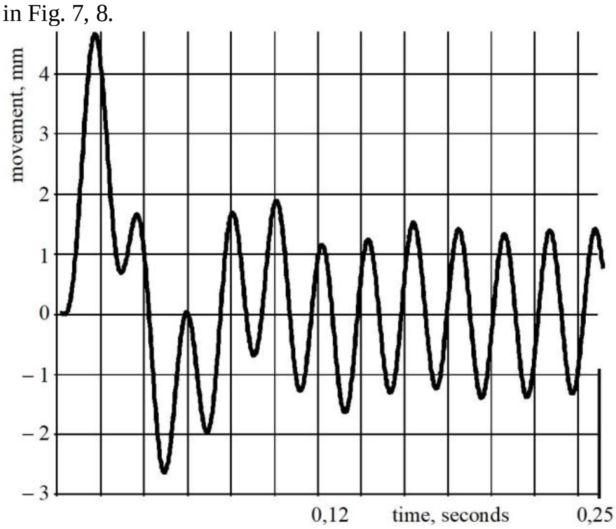

Fig. 7: Diagram of the law of motion of the system "*lever – roll – vibrator*".

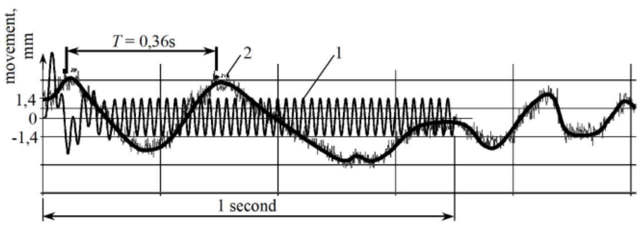

Fig. 8: Diagram of forced oscillations of the system "*lever – roll – vibrator*" (1) and an experimentally determined diagram of the motion of the system without forced oscillations (2).

For this version of RDA, the amplitude of the oscillations of the movable operating element has been determined using the analytical mathematical model of the oscillation system. It is about 1.4 mm at a frequency of oscillation of 3000 oscillations / minute. Such oscillation parameters provide destruction of uneven waves of a material during processing. This allows to stabilize processing of the material, get rid of resonant oscillations, and also increase the efficiency of the activation due to the intensification of influence of the roll on the material.

5. Results and Discussion

In the process of roll-drum activator operation, oscillations of operating elements occur. Their frequency is close to the resonant frequency. The reason for this is the instability of material movement inside the drum and the insufficient rigidity of the system. Ways of preventing the formation of oscillations were not found. Emerging oscillation can be reduced by establishing shock absorbers to suppress oscillations or by using of forced high-frequency oscillations on the system.

The time of effective contact of the material with the roll is reduced (Fig. 3) by using a shock absorber. This, obviously, explains the decrease in the strength of samples when activated with a shock absorber.

Use of a hydraulic shock absorber significantly (three times) reduces the amplitude of the oscillation (Fig. 3), but significantly (by 13-25%) reduces the quality of the activation. Because of this, there is a need for an alternative to such an imperfect system.

A promising way of stabilizing the motion of the system "lever – roll" is imposition of an additional external forced high-frequency oscillations on the system. At the same time, such oscillations can also increase the efficiency of the activation.

The performed oscillation analysis covers experimental and analytical studies that provide sufficient objectivity of the analysis. The analysis is a continuation of investigations carried out at the department of mechanization of construction processes of the Kharkiv National University of Construction and Architecture. The results of the research can be used in a design of a roll-drum actuator.

6. Conclusions

1. Roll-drum machine (activator) with weight pressing down on roll during operation creates oscillations, the frequency of which is close to the own frequency of the system. Those frequencies that are close to resonant oscillations harmfully affect the stability of the machine.

2. It is possible to stabilize the system "lever – roll" by establishing hydraulic shock absorbers. However, such stabilization considerably impairs the efficiency of the activation, so its use is impractical.

3. The most optimal stabilization of the movement of "lever – roll" by using of additional external oscillations on the system has been defined. The analytical mathematical model of the oscillation system in the system of pressing of roll has been created; the parameters of its oscillations have been determined; it was confirmed that the external oscillations has a positive effect on the efficiency of the activation and product quality.

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