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Research paper

Research of Working Process of a Vibration Machine with Controlled Parameters of Motion

Ivan Nazarenko[1*], Oleg Dedov[2], Mykola Ruchynskyi[3], Anatoly Sviderskyi[4]

[1]*Kyiv National University of Construction and Architecture, Ukraine*
[2]*Kyiv National University of Construction and Architecture, Ukraine*
[3]*Kyiv National University of Construction and Architecture, Ukraine*
[4]*Kyiv National University of Construction and Architecture, Ukraine*
**Corresponding author E-mail: i_nazar@i.ua*

Abstract

The constructive scheme of the vibration unit is developed. Presented results of researches of a rammers on determination of parameters, which will provide high efficiency of vibration action on the processed environment for ensuring the quality of the compaction process, which is based on the idea of direct transfer of energy from the working organ to the processed environment. Revealed zones of effective operation of the parameters and characteristics of the system "machine-environment" to ensure high quality of the compaction process, which is based on the idea of the purposeful use of elastic characteristics of the general vibration system, which, under all the same conditions, also reduces the energy costs for some machines to 50%.

Keywords: *compaction plate, frequency and form of oscillation, spatial vibrations, vibrator, vibration rammer*

1. Introduction

Vibration action machines are widely used for grinding, sorting, mixing, sealing and other technological processes. As a rule, the implementation of the technological process is realized on the basis of the usual harmonic modes of motion of vibration machines, in which as vibrators are used vibration blocks of centrifugal action. The hypothesis about the possibility of direct transfer of energy from the working organ to the processed environment is proposed, which is expected to increase the efficiency of the work process. The implementation of such energy transfer is realized in a number of machines: vibrating soil and road rollers, rammers, machines for surface compaction of concrete mixtures, etc. Thus, the basis of the study of complex dynamic systems should be based on a mathematical model for the implementation of which can be used calculation complexes of general purpose, based on numerical calculations and basic laws of the theory of elasticity, plasticity, etc. Recently, such settlement systems have been widely used in various industries, including engineering.

The results of researches of rammers on determination of parameters, which will provide high efficiency of vibration action on the processed environment (soil, concrete mixes), are presented in the work.

2. Literature review

Ensuring the high technological level of vibration rammers, as well as other vibration machines for compaction, depends to a large extent on the physical and mathematical models of the vibration system "Rammer - Ground" which adequately meets the real conditions of the work process. The greatest difficulties arise when choosing a soil model due to the lack of a commonly accepted approach to determining its characteristics and functional dependencies on vibration parameters. There are two fundamental approaches to soil modeling (which is subject to the vibration process): the modeling of inertia-elastic and dissipative properties that are inherent to any vibration system, discrete or distributed parameters.

Many studies [1, 2, 3, 4 and others] devoted to the study of shock-vibration systems are represented, which are the analysis and synthesis of mechanical shock-vibration systems. Concerning works devoted to the study of shock-vibration machines for compaction concrete mixtures, it is possible to note [7, 8, 9, 10 and others]. In these works the parameters of motion of such systems are determined, based on certain assumptions. The most common are the assumptions about the model of the "vibrating machine - processed environment" system, which seems discrete. In work [10] is proposed the refinement of the model was proposed, where a method of transition from discrete continuum systems (discrete - machine, continuum - environment) to purely discrete ones taking into account wave phenomena in a concrete mixture.

3. Purpose of work

The purpose of the article is to investigate and establish the parameters of vibratory machines, which will ensure the high efficiency of vibration action on the processed environment and assessment of the energy characteristics for the development of new progressive machines.

4. Main body

For theoretical researches of the system "vibration rammer - soil" adopted scheme on fig.1, which includes a compacted environment expressed by resistance $b_г$ and stiffness $c_г$ coefficients, vibration rammer, which consists of three masses m_1, m_2, m_3, which are interconnected by elastic elements of rigidity c_1, c_2, c_3, between masses m_1 and m_2 is possible a punch(shock) through the buffer with the stiffness $c_б$ and the coefficient of resistance $b_б$.

The vibration rammer is used as a hinged equipment to the excavator, the connection of the vibration rammer to the arrow of the excavator m_4 is carried out by the equipment which has the stiffness $c_в$ and the resistance coefficient $b_в$.

The compaction of the building mixture may be carried out under the action of two components: the dynamic, which creates oscillations with a given frequency and amplitude, and static, which arises as a result of the weight force of the rammer and the hydraulic force $F_{пр}$ of the arrow of excavator.

Thus, the adjustment of the compaction process modes can be accomplished both by changing the dynamic action (frequency and amplitude of oscillations) and by changing the static pressure on the compaction environment. As a result of the combination of such actions, it is expected to obtain optimal modes of operation at all stages of compaction.

To compile the equations of the system motion, we use the Hamilton principle [2, 3, 10], according to which the motion of masses of the system can be carried out in two cases:

- motion without impact between masses m_1 i m_2:

$$\begin{aligned}
&m_1\ddot{x}_1 + b_1(\dot{x}_1 - \dot{x}_3) + c_1(x_1 - x_3) + b_г\dot{x}_1 + c_г x_г = 0,\\
&m_2\ddot{x}_2 + b_2(\dot{x}_2 - \dot{x}_3) + c_2(x_2 - x_3) = F_{гід},\\
&m_3\ddot{x}_3 + b_2(\dot{x}_3 - \dot{x}_2) + c_2(x_3 - x_2) + b_1(\dot{x}_3 - \dot{x}_1) +\\
&+ c_1(x_3 - x_1) + b_в(\dot{x}_3 - \dot{x}_4) + c_в(x_3 - x_4) = -F_{гід},\\
&m_4\ddot{x}_4 + b_в(\dot{x}_4 - \dot{x}_3) + c_в(x_4 - x_3) = -F_{пр}.
\end{aligned} \qquad (1)$$

- motion with impact between masses m_1 i m_2:

$$\begin{aligned}
&m_1\ddot{x}_1 + b_б(\dot{x}_1 - \dot{x}_2) + c_б(x_1 - x_2) +\\
&+ b_1(\dot{x}_1 - \dot{x}_3) + c_1(x_1 - x_3) + b_г\dot{x}_1 + c_г x_г = 0,\\
&m_2\ddot{x}_2 + b_2(\dot{x}_2 - \dot{x}_3) + c_2(x_2 - x_3) +\\
&+ b_б(\dot{x}_2 - \dot{x}_1) + c_б(x_2 - x_1) = F_{гід},\\
&m_3\ddot{x}_3 + b_2(\dot{x}_3 - \dot{x}_2) + c_2(x_3 - x_2) +\\
&+ b_1(\dot{x}_3 - \dot{x}_1) + c_1(x_3 - x_1) + b_в(\dot{x}_3 - \dot{x}_4) +\\
&+ c_в(x_3 - x_4) = -F_{гід},\\
&m_4\ddot{x}_4 + b_в(\dot{x}_4 - \dot{x}_3) + c_в(x_4 - x_3) = -F_{пр}.
\end{aligned} \qquad (2)$$

where $F_{гід} = f(t)$ - force of vibrations; $F_{пр} = f(t)$ - squeezing force on the arrow of the excavator.

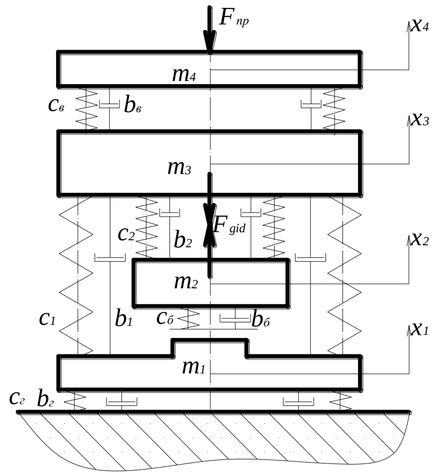

Fig. 1: The calculation scheme of the system "vibration rammer - the ground"

The complexity of the processes occurring during the compaction of building mixtures undoubtedly makes it difficult to solve the problems of choice and justification of compaction models.

5. Results of research

Let's consider how the disturbing hydraulic force F changes, considering that the drive of a vibration rammer is carried out from a hydro system of the base machine and at the same time as a vibrator a design of a distributor with a rotary spool is used(Fig.2) and executing mechanism - hydraulic cylinder, which allow us to consider the system being investigated as a hydraulic operated mechanism of a pulsator type with a hydraulic inverted coupling.

The hydrosystem of the base machine is characterized by two values [26, 84]: p_H - nominal pressure and Q_H - nominal flow of the working fluid.

A rotary type hydraulic distributor with guaranteed overlapping of work windows can have two positions:

- windows of the supply and drain are closed;
- windows of the supply and drain are open.

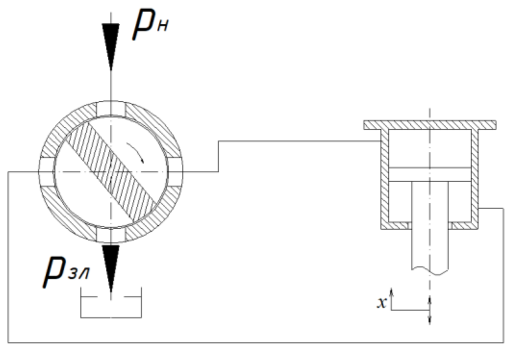

Fig. 2: Scheme of the hydraulic executing mechanism of the vibration rammer.

In the first case, the flow of working fluid to the cylinder can be considered absent, so the distributor can be regarded as a throttle nozzle-damper (Fig. 3). For such a throttle, ignoring the pressure of the drainage line, the flow of the working fluid is determined by the formula [14, 22]:

$$Q = \mu \cdot \pi \cdot d_в \cdot z\sqrt{2 \cdot g\,\frac{p_н}{\gamma}}, \qquad (3)$$

where μ – cost factor; $d_в$ – diameter of the window; z – gap between the valve and the housing of the distributor; γ – volume mass of the liquid.

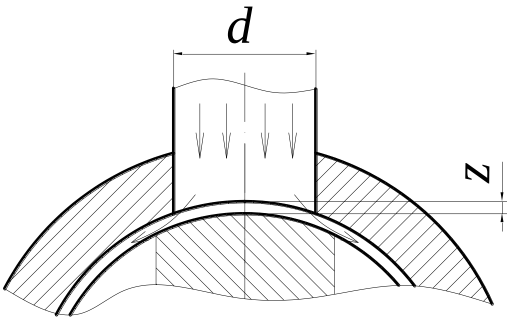

Fig. 3: Distributor scheme with closed window

Equation (3) makes possible to determine the losses due to the flow of the working fluid in the distributor.
Let's consider the case when the windows of the supply and drain are open (Fig. 4).
The area of the open window will be determined through the area of the segment of a circle in radius r. Then the flow of liquid on the distributor:

$$Q = \mu \cdot S_w \sqrt{2 \cdot g \frac{\Delta p_{roz}}{\gamma}}, \tag{4}$$

$$S_w = \frac{r^2}{2}(\psi - sin\psi),$$

$\psi = 2\,arccos\left(\frac{r-l}{r}\right)$, where Δp_{roz} - pressure drop on the dis-

$$l = R\left(\varphi - \frac{\pi}{2}\right).$$

tributor;- the area of the window of the distributor.

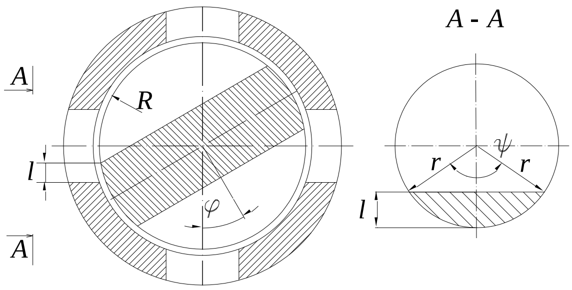

Fig. 4: Distributor scheme with open window

It is quite obvious that the opening cycle - the closure of windows is repeated through the angle $\varphi = \pi / 2$ of the valve, so the value of l will vary from zero to $2r$ with a period $\pi/2$:

$$l = R\left(\varphi - \frac{n\pi}{2}\right), \quad n = 0,1,2..k \tag{5}$$

Since the distributor has a guaranteed window overlap, for some time the work windows are closed *(l = 0)* and open *(l = 2r)*. The character of the area of the window of the distributor is shown in Fig. 5.

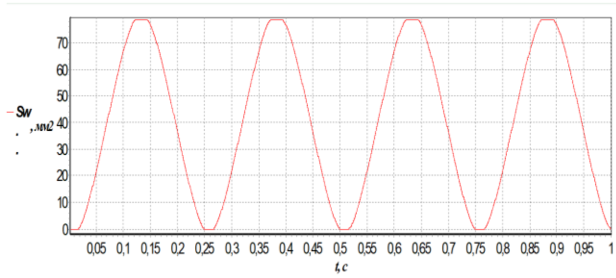

Fig. 5: The character of changes area of the window of distributor in time

Angle of valve turn $\varphi = \omega_{roz} \cdot t$ defines the position of the valve and has a connection with the frequency ω_{roz}, which is an input function for controlling the process of the vibration rammer.
The pressure drop on a distributor during an open window can be determined by the following formula:

$$\Delta p_{roz} = p_н - p_{зл} - \Delta p_{цил} \tag{6}$$

To determine $\Delta p_{цил}$ consider the movement of the piston of the hydraulic cylinder under the action of the forces of the system (Fig. 6). To simplify mathematical equations we assume that one end of the cylinder is rigidly fixed, and the disturbing force $F_{гід}$ of the system operates on the piston (Fig. 6).

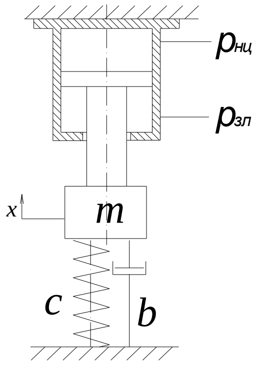

Fig. 6: Scheme to determine the parameters of the motion of the piston

Then the equation of motion of the piston has the form:

$$m\ddot{x} + b\dot{x} + cx = F_{гід}, \tag{7}$$

where m - mass that is attached to the piston; b,c -respectively, the coefficients of rigidity and resistance; $\ddot{x}, \dot{x}, x$ - respectively acceleration, speed and displacement of the piston;
$F_{on} = \Delta p_{цил} \cdot S_{цил}$ - hydraulic resistance force acting on the piston.
For an objective analysis of the mass ratio we use the condition under which the total mass of the rammer, the weight of the compaction plate, the power of the drive remain constant.
The masses m_2 and m_3 change according to the linear law: when mass m_2 increases, the mass m_3 decreases.
The estimation of the effect of the ratio of the mass of the impactor and the upper mass will be made on the basis of comparison of the amplitude of the oscillations of the shock mass at different mass ratios and different frequencies because in the vibration mode the maximum value of the amplitude of the oscillations will correspond to the maximum value of the acceleration of this mass.
Since the weight of the hammer in the vibration shock mode will strike on the compaction plate, then the criterion for maximum acceleration (amplitude of oscillation) is well-grounded.
Thus a number of theoretical experiments were carried out using the above-mentioned conditions. In fig. 7, 8 the dependence of the

oscillation amplitude of mass m_2 on the ratio m_3/m_2 at frequencies 5-100 Hz is shown. In this case, the system had a total weight of 150kg with a mass of compaction plate m_2 10 kg.

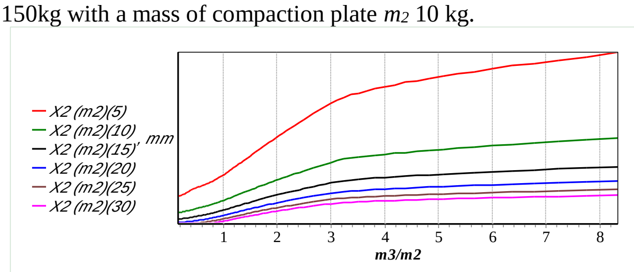

Fig. 7: The dependence of the oscillation amplitude of the mass m_2 on the mass ratio m_3 / m_2 (frequency of oscillations 5-30 Hz)

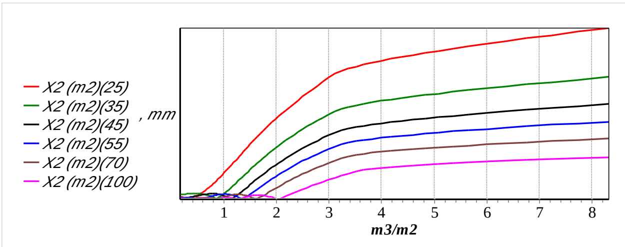

Fig. 8: The dependence of the oscillation amplitude of the mass m_2 on the mass ratio m_3 / m_2 (frequency of oscillations 25-100 Hz)

6. Discussions and results of research of a working process of a vibration rammer with a multi-mode spectrum of oscillations

The choice of physical and mathematical model for the studied system is substantiated. Proven ability of the investigated system with the implementation of variable modes of motion.

The significant influence of constructive and technological parameters on the dynamics of the considered system is established.

The main influence on the movement of the system is determined by the ratio of mass and elastic characteristics (Fig. 7 - 8).

The experiments carried out confirmed the working hypothesis about the possibility of realization of modes, in which the upper mass of the rammer will transmit energy to the processed environment. Obtained dependences of the basic parameters, the use of which ensures the efficiency of the process of compaction of soils. Thus, for the researched system, with the ratio of masses $m_1{:}m_2{:}m_3$ - 1:1:3 and the stiffness coefficients (Fig. 9), an antiphaze motion of masses m_1 and: m_3 is provided, while the amplitude of the oscillations of the compaction plate in the resonance zone is 2.4-3.0 mm.

7. Conclusions

The influence of higher harmonics was noted on the obtained oscillograms when implementing the vibration-shock regime.

So at a frequency of disturbing force intensity equal to 10 Hz, the oscillations of the compaction plate clearly reveal a frequency of 30 Hz, and at a frequency of the disturbing force equal 30 Hz - 60 Hz, respectively. There are three resonant zones on which the vibration rammer can works – 10-15 Hz, 28-32 Hz, 42-46 Hz.Made an estimation of the influence of the weight ratio of the hammer and the upper mass and the choice of this ratio is grounded. On the basis of theoretical studies, the ratio of the weight of the hammer and the upper mass should be taken within 3-4. The influence of the coefficients of stiffness on the overall dynamics of the system is determined. The functional dependence between the coefficients of rigidity c_1 and c_2 and the parameters of motion was obtained, which made it possible to determine the coefficients of stiffness in the vibration and vibration modes of motion.

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