Results of research to improve efficiency of vibrating machines

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Abstract. The paper presents the results of solving the problem of creating an area oscillator, the design of which allows direct oscillations to be obtained in one housing. Moreover, the direction of the action of the driving force can be changed within 360°, during installation. The paper presents a methodology and a tool for the calculation and design of vibration devices with an asymmetric forcing force, which allows you to send a large part of the generated oscillation energy to perform useful work and reduce its consumption during the idle stage. The novelty of the work is that it presents some completed results of theoretical studies and practical implementation of them, relating to the second and third level of enhancement of vibration equipment. The presented scientific and practical results expand the capabilities of engineers of designers and creators of vibration equipment in the field of its improvement and expansion of the field of application. The article outlines the conceptual issues of the formation of the idea and directions of research on the development of the theoretical foundations for the design of vibration modules for technological processes using vibrational technologies. The main results obtained in the development of experimental industrial models of innovative vibration mechanisms and in the development of methods for their design are given.

1. Introduction
In the production activity of man, technological processes that relate to mechanical processes, such as: grinding, sorting, mixing, compaction, shaping and immersion of structural elements, for example piles and dowels into the ground, play an important role. All these processes are to some extent related to the use of forced oscillation of the working bodies of mechanical equipment and machines. Vibration processes used in engineering and in technological processes have long been separated from the independent direction of research and have been widely used in various industries.

On the one hand, all the vibration processes have common basic parameters: amplitude, frequency, oscillation phase, mass of the oscillating body, forcing force, the point of application of the driving force, the shape and direction of the oscillations.

On the other hand, the schemes of excitation of vibration, as a process aimed at performing useful work, have such a wide range and principle of organization that turns individual vibrational mechanisms into separate classes of machines: universal, special or unique.

2. Main part
The most widespread in engineering, obviously, are universal unbalanced area and depth vibrators,
which create a circular driving force, which in turn forms circular or elliptical oscillations of the system. To perform a number of technological operations, oscillations that occur along a straight line are necessary. These oscillations are referred to as directed oscillations. For such processes, the most common scheme is a vibrating mechanism consisting of two paired vibrators with a circular driving force. In such vibrational mechanisms, quenching of the components of the driving forces $P$ in one, for example, horizontal, plane and doubling, addition of $Q = 2P$, of these forces in a perpendicular, vertical, plane is provided in Figure 1.

![Figure 1. Diagram of a double vibrator with directional oscillations](image1.png)

$P$ – driving force of each vibrator, $Q$ – driving force of the directional oscillation, $r$ – eccentricity of the unbalanced unbalance mass, $\omega$ – rotational velocity of the unbalances, $\varphi$ – current value of the angle of rotation of the driving force.

Vibration mechanisms with directional oscillations have received a certain distribution. However, the characteristic features of these mechanisms do not allow them wider dissemination. These features include: the need to synchronize the rotation of unbalanced shafts; the need to increase the overall dimensions of vibration mechanisms in comparison with single-shaft vibrators; The presence of a single point of application of the total forcing force and the only direction of its action.

To eliminate the shortcomings of dual directional vibrators, an experimental industrial sample was developed, and a single-shaft vibrator of directional oscillations of planetary type was tested.

Creation of a single-shaft vibrator of directed oscillations of planetary type. Construction of the planetary mechanism with a gear ratio of internal gearing equal to 2 is used as a basis. In this case, the point of engagement of the pitch circle of the gear rotates along a rectilinear trajectory during rotation. If a center of unbalanced mass is placed at a point located at a distance of the radius of the dividing circle, then such mechanism generates oscillations along a straight line, i.e. directed oscillations. Implementation of this idea is provided by theoretical research and practical implementation of the vibrator. A single-shaft vibrator of directional oscillations (Figure 2) provides directional oscillations along a straight line. Using screw fixers on the body, Figure 2a, a cylindrical vibration vibro-cassette, Figure 2b, any position within 360 degrees can be easily set and the direction of action of the driving force is changed [1,2].

![Figure 2. Single-shaft directional vibrator of planetary type: a – vibrator assembly, b – vibration cassette, c – test bench](image2.png)

Construction of vibrators of directional oscillations of planetary type can be successfully used both in road rollers, Figure 2c, and in bolting machine.

Along with the problem of organizing effective directional oscillations, there is one more important
area for enhancement vibration machines and mechanisms. The problem is that for the realization of a number of technological processes, it is necessary to have not only directional oscillations, but such oscillations, when the driving force acts directly in the direction of performing useful work. Such processes include, for example, compaction of soil and asphalt mix by road rollers, throwing sorted material on the deck plate of the bolting machine and some others.

The presence of a simple driving force and directed oscillations for such processes requires compensation, damping, its component on the “non-working”, idle side of the driving force action. Compensation or blanking of the idle component of the driving force, as a rule, is carried out by attaching to the vibrating system an additional mass, which carries the name – cantledge[3].

Given the presence of such technical problems, the authors of this article conduct research, theoretical substantiation and the creation of more perfect vibration mechanisms and machines.

The basis for the theory of creating a vibrational mechanism with an asymmetric forcing force was the provision on the addition of harmonic oscillations. This task is theoretically inverse to the task of the expansion of an oscillatory process by means of a Fourier series. It is known that a blow is the most effective way of doing work. If the function of a single systematic blow, Figure 3, is expanded in a Fourier series, then we get a series of harmonic oscillation components that, when added, practically simulate the impact effect [4].

![Figure 3. Scheme of the working cycle of the vibrator with the presence asymmetry of the driving force](image)

Attempts to use this provision were periodically made in a number of research schools. Proceeding from this proposition, a method and arrangement for exciting asymmetric oscillations was proposed. It is known that the efficiency of asymmetric oscillations is estimated by the coefficient of asymmetry of the driving force, which can be called the dynamic coefficient:

$$k_d = \frac{F_w}{F_i}$$

where $F_w$ – component of the driving force in the direction of action when performing useful work, $F_i$ – component of the driving force in the direction of action when performing idling.

To evaluate the dynamic coefficient of a multistage vibrational mechanism with asymmetric oscillations, a vibration stand with asymmetric oscillations and a method for determining the dynamic coefficient were developed.

The paper provides recommendations for designing a vibration mechanism with asymmetric oscillations, consisting of four stages. We recommend the following ratio of the static moments of the unbalance 100: 18.72: 5.8: 1.38. With a multiple ratio of the rotation speed of the unbalanced shafts: 1: 2: 3: 4.

Using the suggested initial data, Table 1, we have the dynamic coefficient $K_d = 3.8213$. 
Table 1. Initial data for determining the dynamic coefficient

| № of vibrator | 1   | 2     | 3   | 4   |
|---------------|-----|-------|-----|-----|
| Weight (kg)   | 10  | 1.872 | 0.58| 0.138|
| Radius (cm)   | 1   | 1     | 1   | 1   |
| Beginning phase (degree) | 0   | 0     | 0   | 0   |
| Degree of rotation of the unbalance (rpm) | 500 | 1000  | 1500| 2000|

When certain values of the unbalance mass and the static moment of unbalance are changed, Table 2, we obtain a higher dynamic coefficient $K_d = 3.997$.

Table 2. Initial data for determining the dynamic coefficient with changed parameters

| № of vibrator | 1   | 2     | 3   | 4   |
|---------------|-----|-------|-----|-----|
| Weight (kg)   | 10  | 1.873 | 0.58| 0.17|
| Radius (cm)   | 1   | 1     | 1   | 1   |
| Beginning phase (degree) | 0   | 0     | 0   | 0   |
| Degree of rotation of the unbalance (rpm) | 500 | 1000  | 1500| 2000|

Graphically, the sum of the harmonic oscillations is shown in Figure 4.

![Graph](image)

**Figure 4.** The result of the calculation of the asymmetry coefficient of oscillations, according to the data of Table 2.

$Y_r$ – the magnitude of the component of the driving force in the working direction of the action (+) and in the direction of idling (-), $t$ is the current time of one cycle, 1, 2, 5, Sum – harmonic of the first, second, fourth unbalance and total vibration, respectively.

Performing the selection of the mass of the imbalance for the fifth and sixth steps, Table 3, it is possible to calculate the largest or necessary coefficient of asymmetry of the driving force.
Table 3. Initial data for determining the dynamic factor for a vibrating mechanism with asymmetric oscillations with six vibrational steps

| № of vibrator | 1    | 2    | 3    | 4    | 5    | 6    |
|---------------|------|------|------|------|------|------|
| Weight (kg)   | 9.5  | 1.873| 0.58 | 0.17 | 0.045| 0.0096|
| Radius (cm)   | 1    | 1    | 1    | 1    | 1    | 1    |
| Beginning phase (degree) | 0    | 0    | 0    | 0    | 0    | 0    |
| Degree of rotation of the unbalance (rpm) | 500  | 1000 | 1500 | 2000 | 2500 | 3000 |

The results of the calculation make it possible to obtain the coefficient of asymmetry of the driving force equal to \( K_d = 4.974 \), Figure 5.

Figure 5. The result of the calculation of the coefficient of asymmetry of the oscillations, according to the data of Table 3.

Yr – the magnitude of the component of the driving force in the working direction of the action (+) and in the direction of idling (-), \( t \) is the current time of one cycle, 1, 2, 5, Sum – harmonic of the first, second, fourth unbalance and total vibration, respectively.

No less important for the design of vibration equipment than the design of multi-stage vibrating mechanisms, there is the issue of designing two-stage vibrators with directed asymmetric vibrations. As a rule, such vibration mechanisms are two-frequency ones. Therefore, it is advisable to consider the results of the calculation of such mechanisms[5,6].

Thus, given the parameters of the oscillatory system, we have the following oscillation pattern, Figure 6.

Table 4. Initial data for a two-stage vibrator, version 1

| \( m_1 \) | \( m_2 \) | \( r_1 \) | \( r_2 \) | \( n_1 \) | \( n_2 \) | \( \varphi \) | \( K_d \) |
|---------|---------|---------|---------|---------|---------|---------|---------|
| 10      | 1.39    | 1       | 1       | 500     | 1000    | 0       | 2.048   |
Figure 6. Two-frequency vibrator with φ=00.

Changing the phase shift within 360° leads to a decrease in the value of Kd. However, for φ = 180°, the largest value of the driving force, changing the sign to the opposite one, shifts by half a period.

As the rotational speed is increased, the magnitude of the component of the driving force on the working side of the action increases in the first place. With a multiple increase in the rotational speed of the shafts, the dynamic coefficient retains its value, Figure 7.

| Table 5. Initial data for a two-stage vibrator, version 2 |
|----------------------------------------------------------|
| m1  | m2  | r1  | r2  | n1  | n2  | φ   | Kd  |
| 10  | 1.39| 1   | 1   | 1000| 2000| 180 | 2.05|

Figure 7. Two-frequency vibrator with φ = 00 and increased speed.

3. Conclusion

Thus the authors developed and proved in work the method of estimating the choice of geometric and technological parameters of the unbalances of a multistage vibrational mechanism with asymmetric oscillations in terms of the dynamic factor, the coefficient of asymmetry, and the forcing force. The constructive solution for creating vibration mechanisms with asymmetric oscillations and the method for their rapid estimation by the asymmetry coefficient of the driving force allow one not only to shorten the time for designing the vibrational mechanism, but also to achieve the maximum efficiency of work by obtaining the maximum coefficient of asymmetry of the forcing force.

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