To the question of non-harmonic force oscillations and moment by the vibration generator

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Abstract In the article, a vibration exciter is considered, containing four unbalances rotating around parallel axes, the axis of rotation of which are located on a common base, rigidly connected with the working body. The imbalances rotate uniformly and in pairs have the same angular velocity in the opposite direction. The conditions are formulated to ensure excitation of a rectilinearly oscillating force and rotation conditions by the vibration exciter. The purpose of the created dibaleans is to inform the working body of the machine of non-harmonic oscillations. The dependence of the force excited by the vibration exciter on the angle of rotation of the imbalance is determined.

As a result of the studies, it was found that a centrifugal vibration exciter with four unbalances rotating in pairs with the same angular velocities can be used depending on the mutual phasing of the unbalances to excite non-harmonic oscillations of either force or moment.

1. Introduction
We known the way of transferring vibration technological and transport machines of out-in motions of an axifugal vibration generator. But this way is used quite rarely. The main reason is inefficient knowledge of characteristics of the axifugal vibration generator with 4 deballances, illumination of the power factor (power or moment) with parameters according to the law of factor motion.

2. Materials and methods
The vibration generator has 4 deballances (figure 1), which turns around a parallel shaft. The rotational shafts of deballances are on the main base, and connects with tool. Deballances turn slowly and mutually and have the identical angular speed of reverse direction.

Furthermore, the angular speed of one pair of deballances is twice smaller than another pair of deballances, so one pair turns with the speed \( \omega \), and another - \( 2\omega \). Deballances which turn with equal angular speed have equal deballances, so there are equal products of unbalanced mass \( m \) on its eccentricity \( r \) in relation to rotational shaft.

For the identification of deballances the numeration is given. It is regarded that the first pair of deballances, turns with frequency \( \omega_1 = \omega \), second pair \( \omega_2 = 2\omega \). In accordance with numeration mass and eccentricity of every deballance of first pair are \(- m_1 \) and \( r_1 \), second pair are \( m_2 \) and \( r_2 \). For simplicity similar-named deballances turn with equal angular speed. The line which joins two similar-named deballances is common perpendicular and internal between two shafts, has common distance of
similar-named debalances. Debalances which turn with low angular speed are low-speed and with high speed – fast speed.

There are some aspects of the fluctuating force vibration generator in-line of rotational shafts of debalances have to be situated with interaxle lines of first and second debalances which are parallel; rotational shafts of similar-named debalances are symmetrical to the same plumb to interaxle lines.

The rotation of debalances must be simultaneous and according to phase. With the use of gear or belt gear the rotation excludes friction of leader and driven elements. On the scheme of the vibration generator (figure 1) simultaneous rape and phase of rotation debalances are not shown.

Phasing of debalances provides the aspect: any period of time with conjoined stop of debalances, the similar-named debalances must be symmetrical according to the centre line of rotational shafts of debalances.

It should be noticed that the main purpose of such generator is the way of sign non-harmonic actions of tool. In such case non-harmonic of law of actions is expressed in inequality of positive meaning of speeding-up of the module for major speeding-up.

It is known that such non-harmonic law of actions for tools in comparison with harmonic law of actions with equal conditions provides the increasing of medium speed of vibration transport of center body (or separate center bodies or flowing medium). In such case, the central speed of vibration transport depends on state of models of major meaning of speeding – up the work force in line and back direction.

Let us have a look at the principle of such generator with 4 debalances. The deviation of vibration force from angular speed of debalances is identified.

The easiest configuration of debalances have two configurations. The first-configuration of force persistence of debalances is perpendicular to line which connects shafts and its rotation. The second -centripetal of force persistence of similar-named debalances directs to separate sight of straight line which connects debalances turn.

In figure 1, debalances is in described configurations. Thus centripetal force of persistence of first and second debalances pairs directs in one sight.

For identification of relation of force of vibration generator the violent configuration shows, which occupies after slewing out of original configuration upon the expiration of violent configuration at time of $t$ (figure 2). In this case low-speed debalances turn on corner $\delta_1 = \omega_1 t = \alpha x = \delta$, but fast-rotating -on corner $\delta_2 = \omega_2 t = 2\alpha x = 2\delta$.

With the straight line of debalances turning the centrigugal force persistence is developed. The centripetal force persistence is developed where first and second debalances were alike $P_1 = m_1 r_1 \omega^2$ and $P_2 = 4m_2 r_2 \omega^2$.

We can expand the force persistence of debalance on two perpendicular components: upstanding and acclinal (figure 3). Upstanding and aclinal components of force persistence are alike $P_1^B = m_1 r_1 \omega^2 \cos \delta$ and $P_2^T = m_2 r_2 \omega^2 \sin \delta$. In figure 3, aclinal components of force persistence of similar-named debalances compose one another. Upstanding components of force persistence of debalances combine and produce composite force:

$$P_x = 2m_1 r_1 \omega^2 \cos \delta + 8m_2 r_2 \omega^2 \cos 2\delta \quad (1)$$

Thus, the force which changes to non-harmonic law is directed along the straight line, was the central line of rotational shafts.

It should be noted that such force is a base and tool, which is connected with the straight line of non-harmonic action in such case if this force gets in the centre of mass of a wavering system.

The equation 1 shows that stirring force of vibration generator is a sum of two forces: first force-persistence forces of first pair of debalances, second force – persistence forces of second pair of debalances. Each of these forces is a product of two values – constant and variable. Constant value is equal to double meaning of centrifugal force persistence of relevant debalance. Variable value is a cosine function of turning angle of relevant debalances.
For convenience it is necessary to change to non-dimensional expression of the duration vibration generator force from turning angle of debalances.

The deviation of vibration force in non-dimensional expression is obtained by dividing of two parts equation (1) on monomial factor of constant efficient summand at right part of equation or on max constant efficient summand, or on min constant efficient summand. It should be noted that the last two ways of changing to non-dimensional expression of duration force of cosine turning angle of one pair of debalances is the unit, but over the cosine function angle another pair of debalances is smaller than unit or longer than unit. In the equation (1) constant values of summand of first part have common multiple \(2mr\). Then the deviation of active force in non-dimensional is alike:

\[
P_L = \frac{P_L}{2mr\omega^2} = a \cos \delta + b \cos 2\delta
\]  

where

\[
a = \frac{m_1r_1}{mr}
\]

\[
b = \frac{4m_2r_2}{mr}
\]

It will be noted that summand of the right part of equation has common multiple \(2\omega^2\). If this multiplier is singular, then after dividing two parts of equation on \(2\omega^2\) the obtaining deviation will be deviation in non-dimensional on the basis \(mr = 1\).

Let us consider the way of forcing action with non-harmonic law of the vibration generator.

It is known the way of forcing actions, which is founded on using vibration generator with 4 debalances, turning road parallel shafts. Rotation shafts are on the base. Debalances turn equally and by pairs have equal in size and duration turning speeds and equal disbalances (figure 3). At that, the first pair of debalances turns with angular speed \(\omega_1 = \omega\), and second – with angular speed \(\omega_2 = 2\omega\). Turning of debalances has to be synchronize and according to phase in such way that similar-named debalances occupy position in which centrifugal force persistence are parallel and are directed in side opposite. Therefore, centrifugal force persistence of similar-named debalances create a pair of forces, moment which is changeable in value and direction. The value and direction of the moment of force are depends on configuration of debalances. The driven moment of vibration generator is equal to algebraic sum of moments, which are created by centrifugal forces persistence of the first and second pair of debalances.

For the following discussions for deviation we determined the moment for turning angle of debalances which are taken like heading.

It was noticed that as heading place of debalances the simple for building up of vibration generator has been already chosen. It is obvious, that for deviation comparison of driven four-debalanced vibration generator of two different force factors, force and moment, it is necessary to choose heading place for debalances in it or another way of force factor comparison, which was satisfied equal conditions. Such conditions state for centrifugal force persistence of similar-named debalances in driven place perpendicular to lines, which connect its turning shafts; centrifugal force persistence of first and second pairs of debalances produces force factors (force or moment) of equal direction. Let us notice and specify driven debalances, the driven configuration, the centrifugal force persistence of similar-named debalances producing max possible force factor.

In figure 4 the violent place of debalances is shown: first pair of debalances with turning angle speed \(\omega_1 = \omega\), on its turning from driven place on angle \(\delta_1 = \delta\), second pair of debalances turns with angle speed \(\omega_2 = 2\omega\), it is turning from driven place on angle \(\delta_2 = 2\delta\). Let us consider the moment which
directs counterclockwise is positive. Like in figure 4, in such place force persistence of first and second pairs of debalances creates a pair of force which moments are positive. The deviation of pair moment which produces of force persistence with low-speed debalances is alike \( M_1 = 2l_1m_1r_1\omega^2 \cos \delta \). The deviation of pair moment which produce of force persistence with fast-speed debalances is alike \( M_2 = 8l_2m_2r_2\omega^2 \cos 2\delta \). So, the deviation of active moment of vibration generator from the turning angle of debalances is alike:

\[
M_\Sigma = 2l_1m_1r_1\omega^2 \cos \delta + 8l_2m_2r_2\omega^2 \cos 2\delta 
\]  

From the equation (5), the deviation of vibration generator moment is non-harmonic.

The right summand of the first part of equation (5) have two summands has a cosine function of turning angle for relevant debalances. The constant summand is criteria of pair moment, which is created by centrifugal force persistence to relevant debalances in starting position.

The deviation of vibration moment in non-dimensional formula is obtained by two ways of transformation of analogue to non-dimensional formula. Let’s these constant summands of the right part of equation have one common summand \( 2lmr\omega^2 \). Then after dividing two parts (5) on \( 2lmr\omega^2 \) the deviation of vibration moment in non-dimensional formula is alike:

\[
M_\Sigma = \frac{M_\Sigma}{2lmr\omega^2} = a_1 \cos \delta + b_1 \cos 2\delta 
\]  

where

\[
a_1 = \frac{l_1m_1r_1}{lmr} \tag{7}
\]

\[
b_1 = \frac{4l_2m_2r_2}{lmr} \tag{8}
\]

It should be noticed that centrifugal forces persistence creates a pair of forces and relevant phasing and turning in one direction. Phasing of debalances has to be provided by one provision. In starting centrifugal forces persistence of similar-named debalances has to direct in opposite sight. Moreover, in starting position centrifugal forces persistence of similar-named debalances can be parallel or along the right line, which connects with two turning shafts.

3. Results and discussion

The conclusions are the following.

The vibration generator with four debalances, by pairs with equal angular speeds, can be used in duration with mutual debalances for vibration non-harmonic actions force or moment. The deviation in non-dimensional formula differs in coefficient \( a_1 \) and \( a \) ahead \( \cos \delta \) function and coefficients \( b \) and \( b_1 \) ahead \( \cos 2\delta \) function. In addition with vibration generator the interacted direction of first and second pair of debalances (\( 2l_1 = 2l_2 \)), than the deviation between force and moment in non-dimensional formula are equal as \( a = a_1 \) and \( b = b_1 \).
Figure 1. Four unbalanced centrifugal exciter circuits.

Figure 2. Scheme of a centrifugal vibration exciter for excitation of rectilinear oscillations of force according to a non-harmonic law.

Figure 3. Scheme of a centrifugal vibration exciter for exciting moment oscillations according to a non-harmonic law.

Figure 4. Arbitrary position of unbalances in a centrifugal vibration exciter to excite moment oscillations in accordance with a non-harmonic law.
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