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On the Dynamics of an Enhanced Coaxial Inertial Exciter for Vibratory Machines

Volodymyr Gurskyi [1], Vitaliy Korendiy [1], Pavlo Krot [2,*], Radosław Zimroz [2], Oleksandr Kachur [1] and Nadiia Maherus [1]

[1] Institute of Mechanical Engineering and Transport, Lviv Polytechnic National University, 79013 Lviv, Ukraine
[2] Faculty of Geoengineering, Mining and Geology, Wroclaw University of Science and Technology, 50-370 Wroclaw, Poland
* Correspondence: pavlo.krot@pwr.edu.pl

Abstract: Theoretical investigations into the capabilities of a coaxial inertial drive with various operating modes for vibratory conveyors and screens are conducted in the paper. The coaxial inertial exciter is designed with one asynchronous electric motor and the kinematically synchronized rotation of two unbalanced masses. Three variants of angular speeds ratios, namely $\omega_2/\omega_1 = 1$, $\omega_2/\omega_1 = -1$, and $\omega_2/\omega_1 = 2$, are considered. Based on these relations, the circular, elliptical, and complex motion trajectories of the working members are implemented. In the first two cases, single-frequency harmonic oscillations take place. In the latter case, the double-frequency periodic oscillations are excited. The dynamic behavior of the motor's shaft during its running-up and running-out is considered. The influence of the inertial parameters of the unbalanced rotors and the relative phase shift angle between them on the elliptical trajectories of the vibratory system's mass center motion is investigated. The use of forced kinematic synchronization provides the motion stability of the vibratory system for all considered working regimes.

Keywords: vibratory machines; dynamical model; coaxial inertial drive; asynchronous electric motor; double-frequency oscillations; elliptical trajectories; kinematic synchronization

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1. Introduction

Inertial vibration exciters are the most commonly used drives of vibratory screens and conveyors. Depending on the granulation (pelletization) degree and the physicomechanical characteristics of the medium being treated, the rectilinear, circular, or elliptical vibrations of the working member must be provided. These trajectories can be generated due to the specific installation of a single vibration exciter relative to the vibratory system's mass center or by implementing two or more exciters [1]. It is expedient to mount the inertial exciter on the line that passes through the mass centers of the oscillating members of the vibratory screen. If this is not provided at the design stage, the motion trajectories of different points over the whole working surface will be different [2], which can have a negative effect on the efficiency and output of various technological and manufacturing processes. In practice, these negative effects can be observed during the process of discharging the material from a vibrating sieve when this material is intensively hopping (jumping) over the sieving surface. This effect is caused by the additional torque of the inertial vibration exciter, which generates parasitic angular oscillations of the sieve about its center of mass.

Among the most efficient vibration exciters are the ones that allow for changing the amplitude and frequency of oscillations and the trajectory of the working member motion. The specific mounting of the industrial vibration exciters on the rotating (turning) flanges provides the possibility of changing the vibrations' amplitude and the motion trajectories of the oscillating bodies [3]. As such, the synchronous rotation of two unbalanced rotors, which are symmetrically around the center of mass installed on the screen's body, can be provided using additional systems of frequency control [4].

Another factor affecting the amplitude–phase–frequency characteristics of the vibratory system and the motion trajectory of the working member is changing the stiffness parameters of the main and additional supporting springs. Such springs can be installed between two tiers (decks) of the sieves [5]. A novel approach to implementing various elliptical trajectories of the vibratory system is based on controlling the characteristics and parameters of its dynamic model. This technique can be provided due to applying special controllable magnetorheological dampers characterized by changeable stiffness and dissipation characteristics [6]. These devices allow for transforming the traditional circular trajectory to the elliptical or rectilinear one. However, the industrial prototypes of such dampers are currently not widespread, because further investigations on their reliability and longevity are to be carried out.

The complex trajectories of the vibratory system motion can be generated by applying a larger number of independently driven unbalanced rotors. In such systems, the basic motion parameters (acceleration, velocity, displacement amplitude, and orbit shape) can be controlled by changing the rotation direction, frequency, and initial phase of each unbalanced rotor. The implementation of several independent exciters provides the necessity of developing unique theoretical approaches and applying the specialized control systems to ensure their synchronous rotation [7]. The synchronization conditions are defined by the design of the structural elements of the vibratory system, which can be significantly complicated by the exciter's specific parameters, with free bodies additionally performing rotary motion [8]. For the kinematically synchronized vibration exciters, the problems of providing motion stability are very important from the viewpoint of implementing the directed (rectilinear) oscillations of the working member [9]. As such, the application of the double-frequency working regimes can be an effective way of improving simple single-frequency vibratory machines by using the technical capabilities of the existent drive [10] or by changing its design parameters [11].

A large number of scientific and practical investigations are dedicated to the possibilities of implementing two and more independent vibration exciters. As a result of the counter-rotation of two unbalanced masses at different angular speed ratios, it is possible to generate directed (rectilinear), circular, elliptical, and more complex trajectories of the working member, e.g., similar to Lissajous figures [12]. For the systems with three vibration exciters, significantly larger opportunities to affect their kinematic characteristics are opened [13]. In such systems, the influence of the initial phases of separate unbalanced rotors and the conditions of their dynamic self-synchronization is determinative [14].

The implementation of three independent vibration exciters (unbalanced rotors) improves the operational efficiency and output of technological machines and vibratory systems with asynchronous electric drives (Figure 1). The stable motion conditions can be reached at the corresponding phase-frequency characteristics [15] and can be estimated by analyzing the motion stability criteria [16]. Better opportunities for controlling and holding stable working regimes are typical for synchronous electric drives [17]. However, it is necessary to mention that dynamic processes occurring in machines equipped with several drives are much more complicated and require a broader range of problems that should be analyzed [18]. Most commonly, the problems are caused by the resonance phenomena occurring in vibratory equipment during its multi-frequency excitation. In such a case, the problem of providing dynamic rigidity and strength is of the most urgent ones, and can be solved at the stages of designing and frequency analysis of the corresponding vibratory system [19].

The process of providing the stable motion of the double-frequency systems with multiple frequencies of the unbalanced rotors is significantly more complicated [20]. The stability problem, which consists of ensuring the constant phase shift of rotors, can be solved by applying the corresponding control system and by monitoring (estimating) the operation parameters of the electric motors [21,22]. The stability must be provided in the specified range of technological conditions and loading factors [23].

Figure 1. Vibration conveyor with three inertial drives ("GEA Group AG").

In general, the modern tendencies of the development in vibratory technologies and systems require generating more complex motion trajectories with a broader spectrum of harmonics. This can be reached by implementing the compound (combined) motions of the working members of the vibration exciters, particularly the planetary-type ones [24,25]. The enhanced excitation provides complex trajectories of the working members, which are characterized by the additional oscillatory processes and the increased number of harmonics. As such, the working sieving surfaces must be appropriately designed [26]. Unambiguously, the increase in the number of harmonics improves the efficiency of the technological processes in conjunction with reducing the power consumption of the drive due to generating smaller vibration amplitudes. Estimation of technological benefits is very important while implementing novel designs of vibratory machines and while repairing and modernizing the existing ones [27]. As such, the qualitative dynamic analysis of the efficiency of the technological processes and operating conditions of vibratory systems is carried out with the help of traditional numerical methods (e.g., FEM) and by applying novel techniques (e.g., DEM) [28,29].

In order to improve the efficiency and energy savings of various vibratory systems, the inertial-type exciter designed in the form of the doubled unbalanced rotor is proposed in [30,31]. The characteristic feature of such a drive is generating two non-stationary disturbing forces in two mutually perpendicular directions and, correspondingly, two fundamental harmonics of the higher (upper) and lower order with respect to the fundamental frequency of the motor's shaft. Solving the corresponding parametric synthesis problems allowed for implementing two designs of the double-frequency vibration exciter characterized by an equal range of the disturbing force [30,31]. In such a case, one exciter generates double-frequency oscillations with an emphasis on the first harmonic, while another exciter is focused on the second harmonic. The proposed double-frequency vibration exciter allows for implementing the reversibility effects, which are very important for vibratory conveyors [32]. The vibratory devices and their drives must be designed while taking into account the necessity of providing maximal compactness and the highest indicators of operational efficiency [33].

The motivation of the present research is the development of a theoretical basis for the implementation of the single-frequency and double-frequency operating regimes of the vibratory systems with different trajectories of the working member motion using the coaxial vibration exciter driven by a single asynchronous electric motor, which actuates two kinematically synchronized unbalanced rotors. The forced synchronization allows for providing the exact prescribed ratio between the rotors' frequencies and phase shift angle under different technological conditions. The use of a single vibration exciter of the compact design solves the problems of ensuring motion stability and dynamic synchronization, which take place during the multi-drive systems' operation. Additionally, the conducted research is necessary to consider the operational modes of the electromechanical drive to assess the loading during transient processes. Specific attention must be paid to the progress of the running-up (speeding-up) processes when the effects of "sticking" ("jamming") of the electric motor's shaft rotation at the natural frequencies of the vibratory system can occur [34–36]. In multi-mass oscillatory systems, several working regimes can be implemented [37]. In the near-resonance operating conditions of the double-mass systems, the Sommerfeld effect is desirable. However, in this case, the control possibilities are limited to some extent [38]. This fact is related to the possibility of a sudden change in the system's operating conditions and going into the far-over-resonance working regimes with undesired phases (modes) of vibrating bodies, distorting the required trajectory of motion.

The purpose of this research paper is to investigate the different variants of kinematic synchronization of the novel coaxial inertial exciters and complex trajectories of the working members' motion. In order to study the transient regimes of the system, the double-coordinate model of the asynchronous electric motor is used. As a result, the influence of the inertial parameters of the exciters and the relative phase shift between them on the elliptical trajectories of the vibratory system's mass center motion is determined.

2. Design of Vibrator and Methods of Research

2.1. Dynamical Model of the Inertial Vibration Exciter with a Single Asynchronous Electric Motor

The kinematic diagram of the vibration exciter (Figure 2) consists of the internal 1 and external 2 unbalanced rotors, which are coaxially installed in a single body 3 with the help of bearings 4. The actuation of the unbalanced rotors is performed through the belt transmissions 5 and 6 from a single asynchronous electric motor 7.

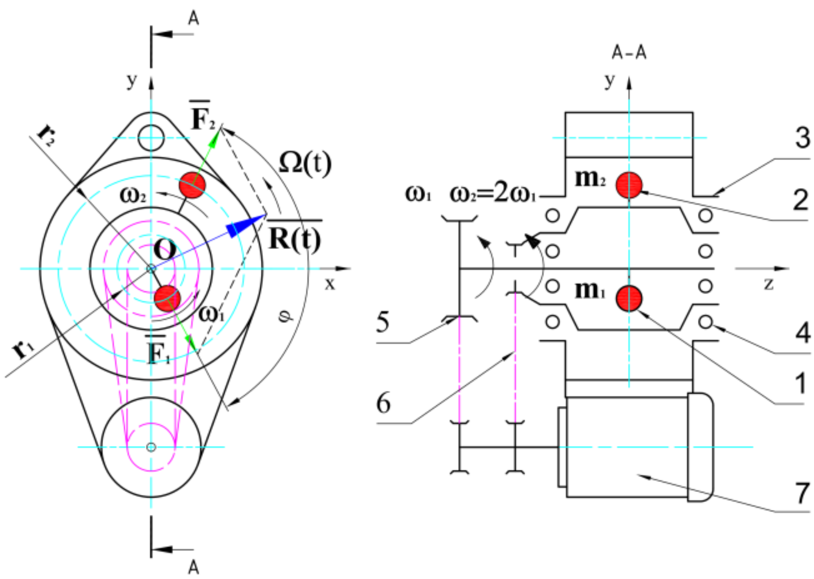


Figure 2. Kinematic diagram of the inertial vibration exciter with a single asynchronous electric motor, as follows: 1, 2—internal and external unbalanced rotors, respectively; 3—exciter's body; 4—bearings; 5, 6—belt transmissions; 7—electric motor.

Due to the use of such a diagram, the vibration exciter resultant force $R(t)$, which is transmitted to the machine's body, changes according to the following dependence:

$$R(t) = \sqrt{F_1{}^2 + F_2{}^2 + 2F_1F_2\cos(\omega_2 t - \omega_1 t + \varphi)} \tag{1}$$

where $F_1 = m_1 r_1 \omega_1{}^2$, $F_2 = m_2 r_2 \omega_2{}^2$ are the magnitudes of excitation (disturbing) forces of the internal and external unbalanced rotors, respectively; $m_1 r_1$, $m_2 r_2$ are the static moments of the internal and external unbalanced rotors, respectively; ω_1, ω_2 are the angular speeds of the internal and external unbalanced rotors, respectively; $\varphi = \varphi_2 - \varphi_1$ is the relative phase shift angle between the internal and external unbalanced rotors. This formula corresponds to the parallelogram law of two vectors' addition of instant forces $\vec{F}_1(t)$ and $\vec{F}_2(t)$ with the constant relative phase shift angle φ between them. The magnitude of the resultant vector $|\vec{F}_1(t) + \vec{F}_1(t)|$ is denoted as $R(t)$.

Based on the previous investigations [10], the following synchronization conditions are substantiated as the most desirable ones from the viewpoint of practical implementation: $\omega_2 = \omega_1$, $\omega_2 = -\omega_1$, and $\omega_2 = 2\omega_1$, under the specific values of the relative phase shift angle φ between the unbalanced rotors. In order to provide these relations, two belt transmissions 1 and 2 are used for actuating two coaxially installed unbalanced rotors from a single electric motor (Figure 3).

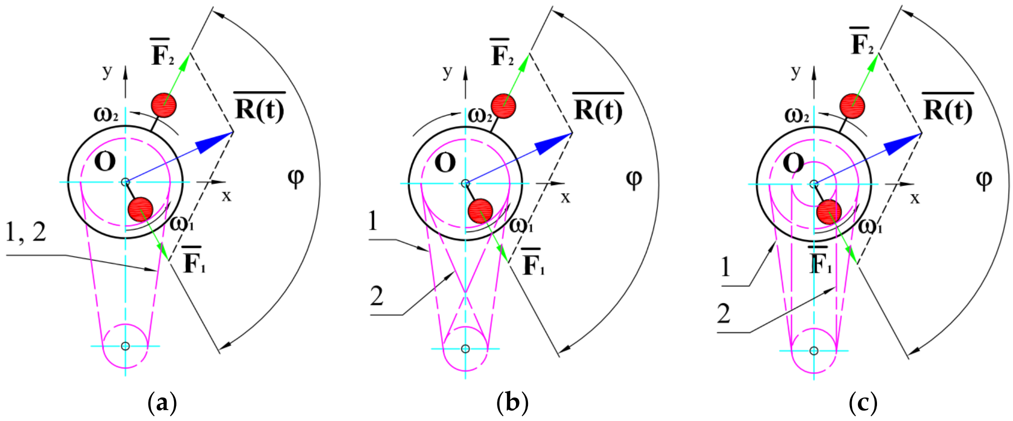


Figure 3. Schemes of kinematic synchronization of the unbalanced rotors of the inertial vibration exciter. (**a**) $\omega_2 = \omega_1$; (**b**) $\omega_2 = -\omega_1$; (**c**) $\omega_2 = 2\omega_1$.

The drives with asynchronous electric motors are the most commonly used in vibratory systems and require relatively simple control systems for changing their shafts' angular speed. The use of a single electric motor for actuating two unbalanced rotors allows for saving energy and increasing operating efficiency. In order to analyze the drive operation, the most widespread double-coordinate electromechanical model of the asynchronous electric motor is used [39,40]. The electromagnetic torque generated by the motor's shaft is determined using the following formula [40]:

$$T_d(t) = \sqrt{3} \cdot p \cdot L_m \cdot \left(i_{s\beta}(t) \cdot i_{r\alpha}(t) - i_{s\alpha}(t) \cdot i_{r\beta}(t)\right) \tag{2}$$

where p is the number of poles; L_m is the mutual inductance; $i_{s\alpha}$, $i_{s\beta}$, $i_{r\alpha}$, and $i_{r\beta}$ are the projections of the stator's and rotor's currents on the coordinate axes $\alpha - \beta$, respectively.

The currents projections on the axes $\alpha - \beta$ are determined by the following system of differential equations [17]:

$$\left.\begin{array}{r} R_s \cdot i_{s\alpha}(t) + \frac{d}{dt}\left(L_s \cdot i_{s\alpha}(t) + L_m \cdot i_{r\alpha}(t)\right) = u_{s\alpha}(t) \\ R_s \cdot i_{s\beta}(t) + \frac{d}{dt}\left(L_s \cdot i_{s\beta}(t) + L_m \cdot i_{r\beta}(t)\right) = u_{s\beta}(t) \\ R_r \cdot i_{r\alpha}(t) + \frac{d}{dt}\varphi_1(t) \cdot \left(L_r \cdot i_{r\beta}(t) + L_m \cdot i_{s\beta}(t)\right) + \frac{d}{dt}\left(L_r \cdot i_{r\alpha}(t) + L_m \cdot i_{s\alpha}(t)\right) = 0 \\ R_r \cdot i_{r\beta}(t) - \frac{d}{dt}\varphi_1(t) \cdot \left(L_r \cdot i_{r\alpha}(t) + L_m \cdot i_{s\alpha}(t)\right) + \frac{d}{dt}\left(L_r \cdot i_{r\beta}(t) + L_m \cdot i_{s\beta}(t)\right) = 0 \end{array}\right\} \tag{3}$$

where $u_{s\alpha}(t) = U_0 \sin(\omega t)$, $u_{s\beta}(t) = U_0 \sin\left(\omega t - \frac{2\pi}{3}\right)$ are the laws of changing the voltage on the stator's winding; U_0 is the nominal voltage value; ω is the nominal cyclic (angular) frequency of the voltage change.

The system of differential Equations (3) is to be solved with respect to the current at the following zero-value initial conditions:

$$i_{r\alpha}(0) = 0,\ i_{s\alpha}(0) = 0,\ i_{r\beta}(0) = 0,\ i_{s\beta}(0) = 0 \tag{4}$$

In order to perform the numerical modeling and analysis of the electric motor running-up (speeding-up) conditions, the electromechanical model of the vibratory system [35] is used, and the characteristics of the asynchronous electric motor of 1.1 kW power [41] are presented in Table 1.

Table 1. Technical characteristics of the asynchronous electric motor [41].

Parameters	Symbol	Values
Electric power	P	1.1 kW
Nominal voltage	U_0	230 V
Nominal speed	n	1420 rpm
Number of poles	p	2
Stator resistance	R_s	7.6 Ω
Rotor resistance	R_r	3.6 Ω
Stator inductance	L_s	0.6015 H
Rotor inductance	L_r	0.6015 H
Mutual inductance	L_m	0.58 H
Moment of inertia	J	0.005 kg·m^2

2.2. Dynamical Model of the Vibratory System with Kinematically Synchronized Unbalanced Rotors

The dynamical model of the vibratory screen or conveyor can be presented in the form of a single-mass oscillatory system [42] with two degrees of freedom, which is subjected to the unsteady disturbance generated by the considered vibration exciter. This model is correct enough in the case when the exciter is installed in the mass center of the working member. The following motion equations describe the double-coordinate spatial oscillations of the working member and the rotary motion of the electric motor's shaft about its longitudinal axis:

$$\left.\begin{array}{c} m\cdot\frac{d^2}{dt^2}x(t)+k_x\cdot x(t)+c_x\cdot\frac{d}{dt}x(t)=R_x(t)\\ m\cdot\frac{d^2}{dt^2}y(t)+k_y\cdot x(t)+c_y\cdot\frac{d}{dt}y(t)=R_y(t)\\ I\cdot\frac{d^2}{dt^2}\varphi_i(t)+M_r(t)+V(t)=T_d(t)\end{array}\right\} \tag{5}$$

where m is the total mass of the working member and the vibration exciter; k_x, k_y are the stiffness coefficients of the supporting springs; c_x, c_y are the damping (viscous friction) coefficients of the springs; $I=\sum_{i=1}^{2}\left(J+m_i{r_i}^2\right)$ is the total inertial moment of the rotating masses about the longitudinal axis of the motor's shaft; J is the moment of inertia of the motor's shaft about its longitudinal axis; $M_r(t)$ is the moment of the viscous friction forces acting in the motor's shaft bearing, as follows:

$$M_r(t)=\sum_{i=1}^{2}\left(\gamma\cdot\left|\frac{d}{dt}\varphi_i(t)\right|+f\cdot m_i r_i\cdot\frac{d}{dt}\varphi_i(t)^2\cdot\frac{d_{0i}}{2}\right) \tag{6}$$

while $V(t)$ is the vibration moment acting upon the motor's shaft, which is generated due to the plane oscillations of the vibratory system, as follows:

$$V(t)=\sum_{i=1}^{2}m_i r_i\cdot\left(\frac{d^2}{dt^2}y(t)\cdot\cos(\varphi_i(t))-\frac{d^2}{dt^2}x(t)\cdot\sin(\varphi_i(t))\right),\ i=1,\ 2 \tag{7}$$

Additionally, $R_x(t)$, $R_y(t)$ are the projections of the alternating disturbing force of the exciter on the coordinate axes x and y, as follows:

$$\begin{array}{l} R_x(t)=m_1r_1\cdot\left(\left(\frac{d}{dt}\varphi_1(t)\right)^2\cdot\cos(\varphi_1(t))+\frac{d^2}{dt^2}\varphi_1(t)\cdot\sin(\varphi_1(t))\right)+\\ +m_2r_2\cdot\left(\left(\frac{d}{dt}\varphi_2(t)\right)^2\cdot\cos(\varphi_2(t)+\varphi)+\frac{d^2}{dt^2}\varphi_2(t)\cdot\sin(\varphi_2(t))\right),\\ R_y(t)=m_1r_1\cdot\left(\left(\frac{d}{dt}\varphi_1(t)\right)^2\cdot\sin(\varphi_1(t))-\frac{d^2}{dt^2}\varphi_1(t)\cdot\cos(\varphi_1(t))\right)+\\ +m_2r_2\cdot\left(\left(\frac{d}{dt}\varphi_2(t)\right)^2\cdot\sin(\varphi_2(t)+\varphi)-\frac{d^2}{dt^2}\varphi_2(t)\cdot\cos(\varphi_2(t))\right)\end{array} \tag{8}$$

All parameters of the vibratory system needed to perform numerical modeling are presented in Table 2.

Table 2. Parameters of the vibratory system.

Parameters	Symbol	Values
Total mass of exciter	m	100 kg
Unbalanced mass 1	m_1	5 kg
Unbalanced mass 2	m_2	5 kg
Springs' stiffness in horizontal and vertical directions	k_x, k_y	3.944×10^5 N/m
Coefficient of viscous damping in horizontal and vertical directions	c_x, c_y	2512 N·s/m
Coefficient of viscous friction in bearings	γ	0.01 N m s/rad
Static moment of the internal unbalanced rotor	$m_1 r_1$	0.15 kg·m
Static moment of the external unbalanced rotor	$m_2 r_2$	0.06 kg·m
Total inertial moment of rotating masses	I	0.015 kg·m^2
Bearing inner diameters	d_{01} d_{02}	0.050 m 0.120 m
Friction coefficient in bearings	f	0.004

Let us adopt zero-value initial conditions for solving the system of differential Equations (5), as follows:

$$\varphi_i(t) = 0,\ \frac{d}{dt}\varphi_i(t) = 0, x(t) = 0, \frac{d}{dt}x(t) = 0, y(t) = 0, \frac{d}{dt}y(t) = 0 \tag{9}$$

The systems of differential equations describing the motion conditions of the vibratory machine (5) and the electric motor's shaft (3) are considered simultaneously with the general synchronization condition of the unbalanced rotors, as follows:

$$\varphi_2(t) = \frac{\omega_2}{\omega_1} \cdot \varphi_1(t) \tag{10}$$

The input voltage control of the electric motor, which is provided by the frequency converter, is assigned as a piecewise linear function, as follows:

$$U_0(\omega) = \begin{cases} \frac{230}{148.7}\omega\ [V],\ \ if\ 0 \le \omega \le 148.7\left[\frac{rad}{s}\right] \\ 230\ [V],\ \ \ if\ \ 148.7\left[\frac{rad}{s}\right] < \omega \le 200\left[\frac{rad}{s}\right] \end{cases} \tag{11}$$

Correspondingly, the instantaneous values of alternate voltages are assumed to be as follows:

$$u_{s\alpha}(\omega, t) = U_0(\omega)\sin(\omega t); u_{s\beta}(\omega, t) = U_0(\omega)\sin\left(\omega t - \frac{2\pi}{3}\right) \tag{12}$$

In order to mimic real practical conditions as closely as possible, an electromechanical model of a vibration system with the equations of motion of an asynchronous electric motor (2)–(4), (11), (12), taking into account its real parameters (see Table 1), was used.

Since we consider the possibilities of operation of the vibrator itself, we omit the equations of rotary oscillations, that is, we assume that translational movement is carried out relative to two axes. In general, if a specific vibration machine is to be considered, then accordingly this movement must be taken into account. That is, for modelling, a generalized dynamical model was simplified, which is described by four coordinates, and we adopted three coordinates instead.

The systems of differential equations of motion of the electric motor (3) and vibration system (5) are solved jointly by the Rosenbrock numerical method using the Maple program and the *dsolve* function, the parameters of which are described in the software package manuals.

The variable input parameters are the phase shift angle between the unbalanced masses, their static moments (see Table 2) and the ratio of the angular speeds of rotation (10).

This made it possible to assess the influence of these parameters on the trajectories and kinematic characteristics of the movement of the center of mass. The nature of transient processes in the form of the start-up and run-out of the electric motor was also studied. With this, we checked the possibility of providing different modes based on one electric motor and the sufficiency of its power for system start-up.

3. Results

While performing the analysis of the system vibrations, the transient and steady-state working regimes of the exciter are considered at different ratios between the angular speeds of the unbalanced rotors. This is shown in Figures 4–8. The considered ratios have been chosen based on the practical needs for implementing elliptical ($\omega_2 = -\omega_1$) and circular trajectories ($\omega_2 = \omega_1$) on standard vibratory machines. Other ratios will produce distinct trajectories, so they have not been considered in this paper. In practice, for some applications, other ratios of angular speeds, which lead to deviation from the elliptical or circular trajectories, may take place and be appropriate. However, for this, it would be more appropriate to use two independent asynchronous motors. Since we currently use one electric motor and kinematically synchronized rotations in order not to complicate the design of the vibrator, we used the most common synchronization conditions. The ratio $\omega_2 = 2\omega_1$ was substantiated in our previous studies and was taken for comparison.

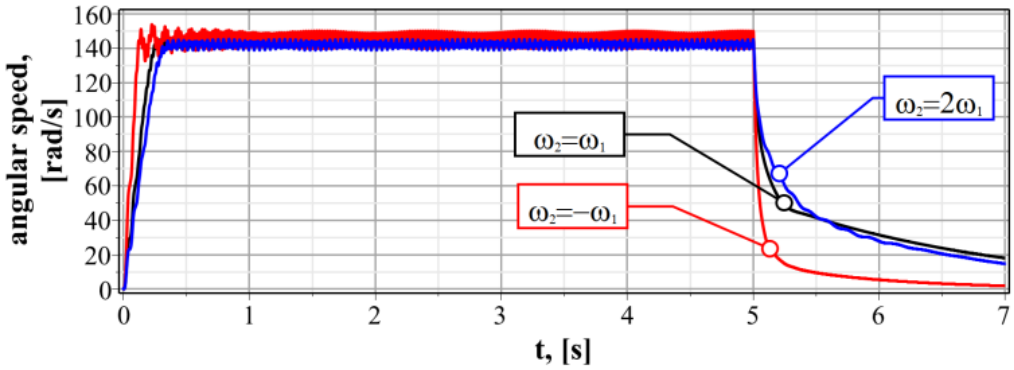


Figure 4. Time series of the electric motor's shaft angular speeds under different synchronization conditions of the unbalanced rotors.

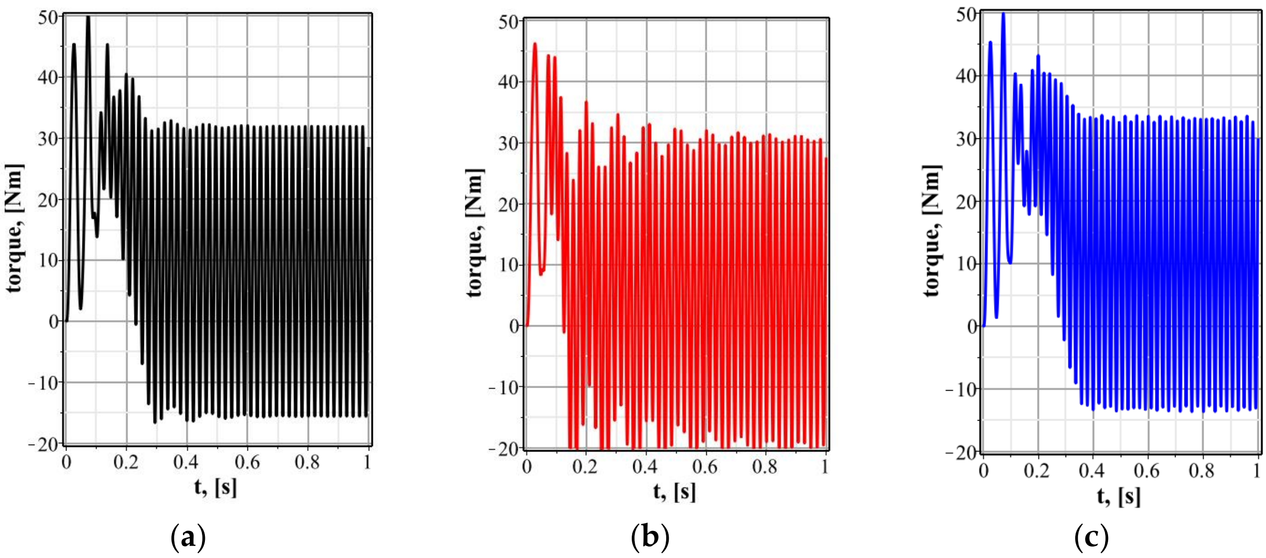


Figure 5. Time dependence of the electric motor's shaft torque at different synchronization conditions of the unbalanced rotors. (**a**) $\omega_2 = \omega_1$; (**b**) $\omega_2 = -\omega_1$; (**c**) $\omega_2 = 2\omega_1$

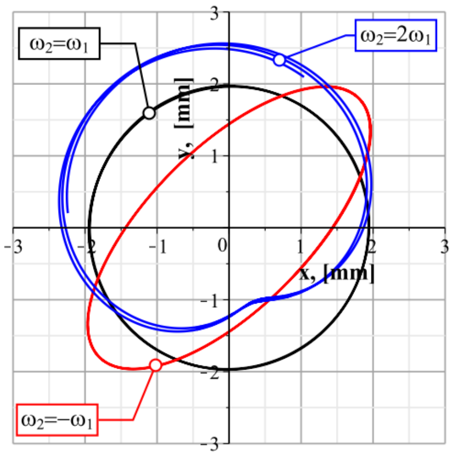


Figure 6. Trajectories of the mass center motion at different ratios between the angular speeds of the unbalanced rotors.

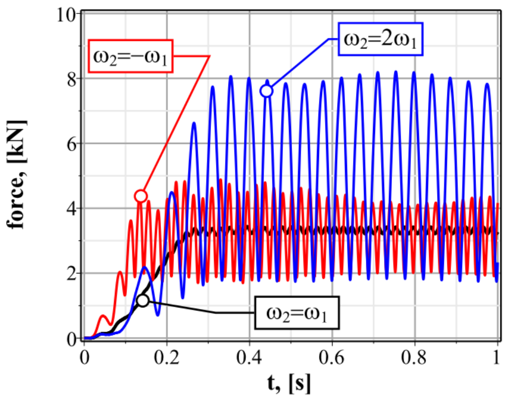


Figure 7. Time dependencies of the disturbing force of the inertial vibration exciter at different ratios between the angular speeds of the unbalanced rotors.

In Figure 4, the running-up (speeding-up) and running-out (stopping) processes take less time under the conditions of synchronous counter-rotation of the unbalanced rotors ($\omega_2 = -\omega_1$) and transient oscillations with a frequency of about 10 Hz at the beginning. At multiple angular speeds ($\omega_2 = 2\omega_1$), the transient processes of the motor's shaft rotation are slightly longer. As such, the decrease in the shaft angular speed with respect to its nominal value is observed within the range of the permissible slip.

In Figure 5, for the steady-state regime (t > 0.4 s), the peak-to-peak amplitudes of torque are almost the same (47–51 Nm) for all three cases. For the second case (Figure 5b), torque has greater negative values (up to −20 Nm), and the average torque is lower. This is due to the action of vibration moment by Formula (7) when two masses rotate in opposite directions. The fluctuation (modulation) of the electric motor's shaft torque is more notable under the conditions of two masses counter-rotation in comparison with the other regimes (Figure 5a,c).

Under the conditions of equal angular speeds of the unbalanced rotors ($\omega_2 = \omega_1$), the traditional single-frequency oscillations are excited and the circular motion trajectories of the system's mass center are described in Figure 6. In this case, the disturbing force is almost constant with its amplitude value of 3.6 kN (Figure 7). Under the counter-rotation conditions of the unbalanced rotors ($\omega_2 = -\omega_1$), the elliptical trajectories are described, and the disturbing force changes its value within 2–4 kN. In the case of double-frequency oscillations ($\omega_2 = 2\omega_1$), the trajectories are similar to the circular ones but are slightly distorted by the second harmonic. Here, the disturbing force is characterized by the largest range of 2–8 kN.

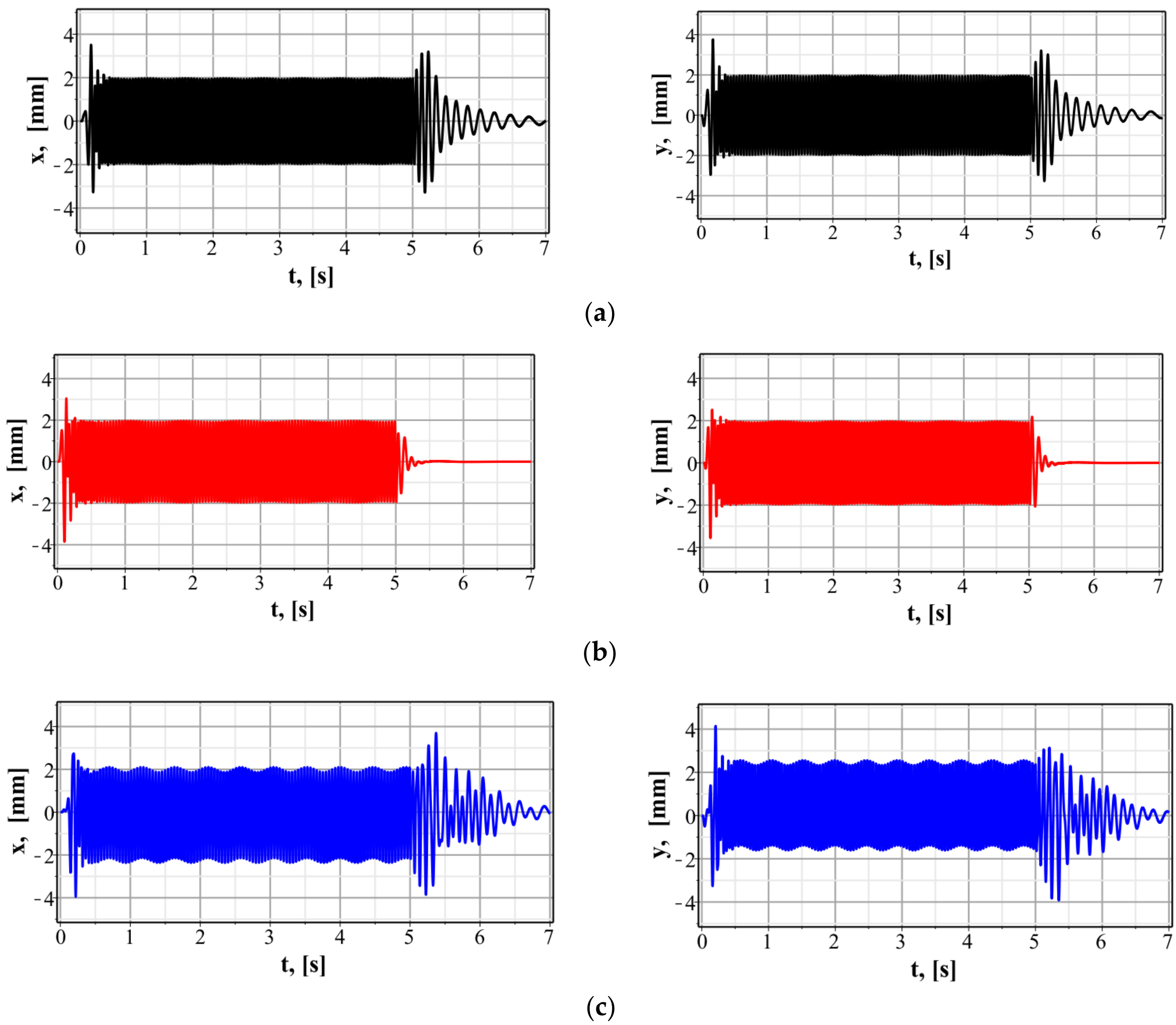


Figure 8. Time dependencies of the working member displacements at different synchronization conditions of the unbalanced rotors. (**a**) $\omega_2 = \omega_1$; (**b**) $\omega_2 = -\omega_1$; (**c**) $\omega_2 = 2\omega_1$

The amplitudes of vibrations in horizontal and vertical directions are within 2 mm under all operating conditions (Figure 8). The transient processes occurring in the vibratory system during its running-out (stopping) are of specific interest. In particular, under the conditions of the unbalanced rotors' counter-rotation ($\omega_2 = -\omega_1$), the running-out (stopping) processes are less intensive, and the oscillations die out (decay) quicker (Figure 8b). At other operating regimes ($\omega_2 = \omega_1$, $\omega_2 = 2\omega_1$), these processes are much longer and more intensive (Figure 8a,c).

In general, the processes of intensively increasing the working member oscillation amplitude during the motor's shaft running-up (speeding-up) and running-out (stopping) can be significantly shortened by implementing special vibration isolators [43]. Based on the thorough analysis of the obtained dynamic parameters, let us present the technical characteristic of the vibratory system in Table 3.

Table 3. Technical characteristics of the vibratory system.

Parameters	Angular Velocities Ratio ω_2/ω_1		
	1	**−1**	**2**
Static moments of the unbalanced rotors 1/2, kg·m		0.15/0.06	
Angular speed of the unbalanced rotors 1/2, rad/s	148.7/148.7	148.7/−148.7	148.7/297.4
Disturbing force, kN	3.6	2–4.5	2–8.2
Total mass, kg		100	
Displacement amplitude, mm		2	
Acceleration, m/s^2	40.7	53.2	89.4
Power, kW		1.1	

The previously performed investigations show that the relative phase shift angle φ is of crucial importance for defining the character of the working member acceleration and the trajectory of the mass center motion. In particular, under double-frequency conditions ($\omega_2 = 2\omega_1$), the acceleration characteristics change both in horizontal and vertical directions. That is why, considering the technological purpose of a vibratory machine, it is necessary to choose the corresponding values of the relative phase shift angle φ, which provides the required character of the acceleration changing in the corresponding direction [31].

The use of the synchronous counter-rotation of the unbalanced rotors ($\omega_2 = -\omega_1$) allows for implementing elliptical trajectories of the working member. The recommended geometrical parameters of the obtained ellipse are determined by the technological requirements aimed at providing maximal efficiency of the corresponding process [44]. During the operation of the proposed vibration exciter and while generating elliptical oscillations of the working member, the position of the ellipse major axis is inclined at the angle $\varphi/2$ with the horizontal axis. Under steady-state operating conditions, the lengths of the ellipse minor and major axes and the amplitudes of horizontal and vertical oscillations of the working member are defined by the static moments of the internal and external unbalanced rotors (Figure 9).

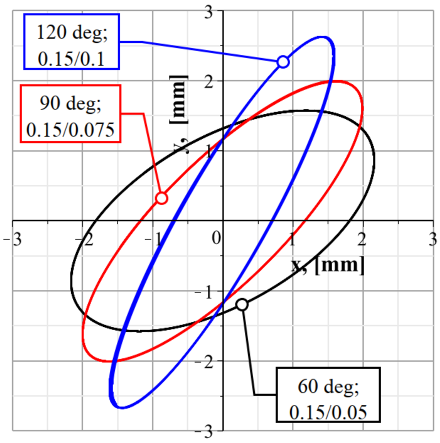
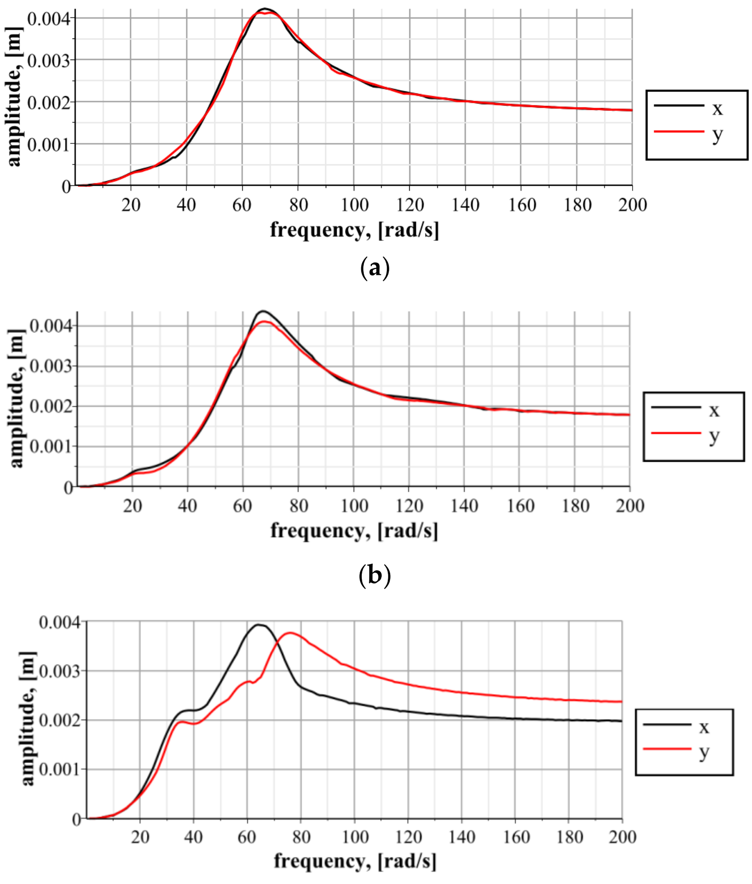


Figure 9. Trajectories of the working member's mass center motion at different values of the phase shift angle φ and static moments of the unbalanced rotors m_1r_1/m_2r_2 under their counter-rotation conditions, as follows: $\omega_2 = -\omega_1$.

A similar principle of the unbalanced rotors' counter-rotation is implemented in the multidrive vibratory systems [45]. In such a case, the forced frequency also influences the system's dynamic characteristics. The determinative characteristic feature of these systems is the possibility of controlling the inclination angle of the elliptical trajectory with the horizontal axis and the lengths of the ellipse minor and major axes by changing the wide range of parameters, e.g., forced frequency, relative phase shift angle, static moments of the unbalanced rotors, etc.

In order to improve the operational possibilities of the considered drive, it is expedient to ensure the fast change in the relative phase shift angle between the unbalanced rotors. This can be implemented in practice by applying controllable electromagnetic and eddy-current clutches, special reversible mechanisms, phase shifters, or electric motors with two shafts, etc. In order to improve the running-up conditions of the vibratory machine, it is expedient to ensure that the phase shift angle $\varphi = 180°$ at the transient working regimes of the electric motor. When the motor starts operating at the steady-state conditions and the angular speed of the shaft reaches the nominal value, the phase shift angle can be changed to the required value. This allows for fast control of the machine's dynamic parameters and the trajectories of the working member.

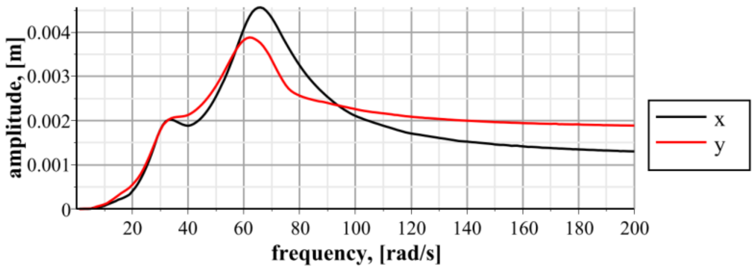
The running-up (speeding-up) processes of the vibratory system at different synchronization conditions can be characterized by the corresponding amplitude–frequency characteristics (frequency response functions) (see Figure 10), which are typical for all single-mass vibratory systems. The frequencies of resonances in vertical (y) and horizontal (x) directions are equal for two variants of a single frequency synchronisation, namely $\omega_2 = \omega_1$ and $\omega_2 = -\omega_1$. In the case of double-frequency synchronization, namely $\omega_2 = 2\omega_1$, a small difference in resonances for the horizontal and vertical coordinates x and y is observed. At the same time, the values of the stiffness in both directions (k_x, k_y) are assigned as equal (3.944×10^5 N/m), as they are similar in real machines. The observed effect is a consequence of the assigned phase-shift angle ($\varphi = 90^{\circ}$) between rotating unbalanced masses and the Sommerfeld effect, which plays a more significant role in the second case.



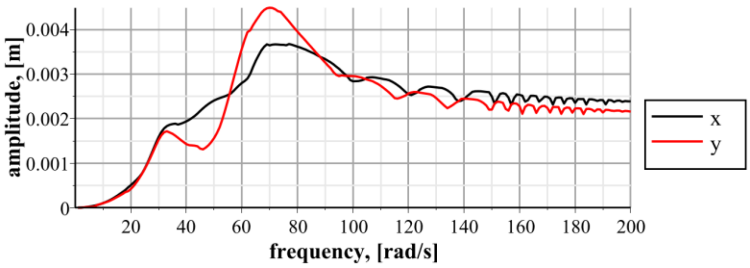
(c)

Figure 10. Amplitude–frequency characteristics of the vibratory system at the phase-shift angle $\varphi = 90°$ and under different synchronization conditions of (**a**) $\omega_2 = \omega_1$; (**b**) $\omega_2 = -\omega_1$; (**c**) $\omega_2 = 2\omega_1$.

The coincidence of natural frequencies in both directions ($\omega_n = 62.8$ rad/s) can increase the machine resonance vibration amplitudes during the running-out period when the machine passes internal resonances at a slow rate of the exciters' rotational speed decreasing. Meanwhile, the divided peaks of machine resonances can provide smoother stopping without excessive loading on the structural elements. In addition, the influence of the initial phase-shift angle on amplitudes of machine vibration under synchronization condition $\omega_2 = 2\omega_1$ is shown in Figure 11. Depending on the phase-shift angle, either the horizontal (x) or vertical (y) direction will have greater amplitude at almost the same frequency of internal transient resonance.



(**a**)



(**b**)

Figure 11. Amplitude–frequency characteristics of the vibratory system for under synchronization condition $\omega_2 = 2\omega_1$ at the phase-shift angles of (**a**) $\varphi = 0°$ and (**b**) $\varphi = 180°$.

4. Discussion

The recently carried out theoretical investigations are focused on substantiating the possibilities of replacing the multi-drive vibratory systems with several inertial exciters by a single vibration exciter with two synchronized coaxially installed unbalanced rotors driven by a single asynchronous electric motor.

The main emphasis is laid on the implementation of circular, elliptical, and complex oscillations of a single-mass vibratory system under the single-frequency and double-frequency excitation conditions. In order to obtain the usually required elliptical trajectories of the working member motion, the synchronous counter-rotation of the unbalanced rotors is applied at $\omega_2 = -\omega_1$. As such, the significant influence on the ellipse geometrical parameters is imposed by the static moments of the unbalanced rotors. Therefore, it is expedient to provide a simple and fast change in the rotors' inertial parameters while designing the vibration exciter. The value of the inclination angle of the ellipse's major axis is equal to half the relative phase shift angle of the unbalanced rotors.

The use of the double-frequency working regimes at $\omega_2 = 2\omega_1$ is characterized by larger values of the working member accelerations compared to the single-frequency conditions. The oscillation amplitudes of the working member are about 2 mm at all operating regimes. The use of forced kinematic synchronization allows for providing the constant value of the relative phase shift angle between two unbalanced rotors and ensuring the motion stability of the oscillating bodies.

The main dynamic characteristics of the vibratory system and the electromechanical parameters of the asynchronous electric motor remain stable during both the transient and steady-state working regimes at all considered synchronization conditions. The differences between the running-up (speeding-up) and running-out (stopping) processes for different operating conditions are negligible (insignificant in practice). Therefore, the working capacity both of the drive and the vibratory machine is provided under all operating conditions.

As the method of cyclic loading reduction on the structural elements of large-scale vibratory machines (sieving screens, heavy feeders), drives of exciters should be regulated from working frequency up to the moment of stopping. Currently, operators usually switch off the power supply and machines pass the main resonance very slowly with high amplitudes and additional stress on the structural elements of machines.

Undoubtedly, the problem of ensuring the stability of rotary motion is decisive for vibratory machines. Especially, as shown in the literature review, it applies to systems with several motors. The use of kinematic coupling between unbalanced masses on the coaxial shafts allowed us to use a single electric motor. As a result, the stability of the movement will now depend on the operation of one electric motor. We calculate the system in such a way that the electric motor operates in a stable mode with a nominal frequency of rotation within the permissible slip limits. The clear periodic functions are obtained for the amplitudes of oscillations in the X and Y directions. We understand that this approach may not cover all theoretically available cases, but the main problem of the transition through resonance has been investigated and the motor reached the nominal frequency of rotation in a stable mode. In general, it can be stated that the developed model is adequate for this version of the coaxial vibrator and proves its stability for chosen ratios of angular speeds, phase shifts, and assumed initial parameters.

The variety of operating conditions, which can be reached with the help of the considered drive, has determinative technological advantages. The implementation of the kinematically synchronized and coaxially installed unbalanced rotors allows for decreasing the manufacturing and maintenance costs of the machine drive and for simplifying the control system. The proposed inertial vibration exciter can be equipped with two independent electromechanical drives for each unbalanced rotor for even more extensive flexibility of control and technological performance, and these issues will be considered in further investigations.

5. Conclusions

The improved coaxial inertial drive is designed for vibratory screens and conveyors, which allows for generating the oscillations of the working member at different ratios between the angular speeds and initial phase shifts of two kinematically synchronized unbalanced rotors for various motion trajectories.

Depending on the technological requirements and the synchronization conditions, the developed drive can provide both the constant disturbing force of about 3.6 kN and the alternating force within 2.0–8.2 kN at a constant power supply of approximately 1.1 kW.

The drive has a simple and compact design and allows for generating the easily controllable circular, elliptical, and complex trajectories of the vibratory machine's working member. The exciter can be effectively implemented in both the resonance and over-resonance vibratory conveyors and screens. The principal advantage of the proposed coaxial scheme is the possibility to use only one drive but generate elliptical trajectories of vibrating working surfaces.

The drive is currently equipped with a single asynchronous electric motor, but the motor type and the number of independently used motors can be changed. These problems will be analyzed in further research on the subject of the paper.

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References

1. Shah, K.P. Construction, Working and Maintenance of Electric Vibrators and Vibrating Screens. Available online: https://practicalmaintenance.net/wp-content/uploads/Construction-Working-and-Maintenance-of-Vibrators-and-Vibrating-Screens.pdf (accessed on 8 December 2022).
2. Nguyen, V.X.; Nguyen, K.L.; Dinh, G.N. Study of the Dynamics and Analysis of the Effect of the Position of the Vibration Motor to the Oscillation of Vibrating Screen. *J. Phys. Conf. Ser.* **2019**, *1384*, 012035. [CrossRef]
3. Despotovic, D.Z.; Pavlovic, A.M.; Radakovic, J. Using Regulated Drive of Vibratory Screens with Unbalanced Motors. *J. Mechatron. Autom. Identif. Technol.* **2017**, *1*, 20–25.
4. Despotović, Ž.V.; Pavlović, A.M.; Ivanić, D. Exciting Force Frequency Control of Unbalanced Vibratory Actuators. In Proceedings of the 2019 20th International Symposium on Power Electronics (Ee), Novi Sad, Serbia, 23–26 October 2019; pp. 1–6.
5. Zhao, G.; Wang, X.; Lin, D.; Xu, N.; Yu, C.; Geng, R. Study of Double-Deck Vibrating Flip-Flow Screen Based on Dynamic Stiffness Characteristics of Shear Springs. *Minerals* **2021**, *11*, 928. [CrossRef]
6. Ogonowski, S.; Krauze, P. Trajectory Control for Vibrating Screen with Magnetorheological Dampers. *Sensors* **2022**, *22*, 4225. [CrossRef] [PubMed]
7. Jia, L.; Zhang, J.; Zhou, L.; Wen, B. Multifrequency-Controlled Synchronization of Three Eccentric Rotors Driven by Induction Motors in the Same Direction. *J. Low Freq. Noise Vib. Act. Control* **2019**, *38*, 615–632. [CrossRef]
8. Zhang, X.; Gu, D.; Yue, H.; Li, M.; Wen, B. Synchronization and Stability of a Far-Resonant Vibrating System with Three Rollers Driven by Two Vibrators. *Appl. Math. Model.* **2021**, *91*, 261–279. [CrossRef]
9. Zhang, X.; Hu, W.; Gao, Z.; Liu, Y.; Wen, B.-C. Composite Synchronization on Two Pairs of Vibrators in a Far Super-Resonant Vibrating System with the Single Rigid Frame. *J. Low Freq. Noise Vib. Act. Control* **2021**, *40*, 2064–2076. [CrossRef]
10. Gursky, V.M.; Kuzio, I.V.; Lanets, O.S.; Kisała, P.; Tolegenova, A.; Syzdykpayeva, A. Implementation of Dual-Frequency Resonant Vibratory Machines with Pulsed Electromagnetic Drive. *Prz. Elektrotechniczny* **2019**, *95*, 41–46. [CrossRef]
11. Filimonikhin, G.; Yatsun, V. Conditions of Replacing a Single-Frequency Vibro-Exciter with a Dual-Frequency One in the Form of Passive Auto-Balancer. *Nauk. Visnyk Natsionalnoho Hirnychoho Universytetu* **2017**, *1*, 61–68.
12. Modrzewski, R.; Obraniak, A.; Rylski, A.; Siczek, K. A Study on the Dynamic Behavior of a Sieve in an Industrial Sifter. *Appl. Sci.* **2022**, *12*, 8590. [CrossRef]
13. Chen, X.; Liu, J.; Li, L. Dynamics of the Vibration System Driven by Three Homodromy Eccentric Rotors Using Control Synchronization. *Appl. Sci.* **2021**, *11*, 7691. [CrossRef]
14. Cieplok, G.; Wójcik, K. Conditions for Self-Synchronization of Inertial Vibrators of Vibratory Conveyors in General Motion. *J. Theor. Appl. Mech.* **2020**, *58*, 513–524. [CrossRef]
15. Zhang, X.; Zhang, W.; Chen, W.; Zhang, X.; Wang, Z.; Wen, B. Theoretical, Numerical and Experimental Studies on Times–Frequency Synchronization of the Three Exciters Based on the Asymptotic Method. *J. Vib. Eng. Technol.* **2022**, *10*, 1091–1109. [CrossRef]
16. Zhang, X.; Zhang, W.; Chen, W.; Hu, W.; Zhang, X.; Wen, B.-C. Synchronization Behaviors of a Vibrating Mechanical System with Adjustable Frequencies and Motion Trajectories. *J. Low Freq. Noise Vib. Act. Control* **2022**, *41*, 945–969. [CrossRef]
17. Fang, P.; Wang, Y.; Hou, Y.; Wu, Y. Synchronous Control of Multi-Motor Coupled with Pendulum in a Vibration System. *IEEE Access* **2020**, *8*, 51964–51975. [CrossRef]
18. Krot, P.V. Dynamical Processes in a Multi-Motor Gear Drive of Heavy Slabbing Mill. *J. Vibroengineering* **2019**, *21*, 2064–2081. [CrossRef]
19. Sedaghati, R.; Suleman, A.; Tabarrok, B. Structural Optimization with Frequency Constraints Using the Finite Element Force Method. *AIAA J.* **2012**, *40*, 382–388. [CrossRef]
20. Zou, M.; Fang, P.; Hou, Y.; Wang, Y.; Hou, D.; Peng, H. Synchronization Analysis of Two Eccentric Rotors with Double-Frequency Excitation Considering Sliding Mode Control. *Commun. Nonlinear Sci. Numer. Simul.* **2021**, *92*, 105458. [CrossRef]
21. Jia, L.; Kong, X.; Zhang, J.; Liu, Y.; Wen, B. Multiple-Frequency Controlled Synchronization of Two Homodromy Eccentric Rotors in a Vibratory System. *Shock. Vib.* **2018**, *2018*, 4941357. [CrossRef]
22. Fang, P.; Shi, S.; Zou, M.; Lu, X.; Wang, D. Self-Synchronization and Control-Synchronization of Dual-Rotor Space Vibration System. *Int. J. Non-Linear Mech.* **2022**, *139*, 103869. [CrossRef]
23. Xiong, G.; Hou, Y.; Fang, P.; Du, M. Stability and Synchronous Characteristics of Dual-Rotors Vibrating System Considering the Material Effects. *J. Vib. Eng. Technol.* **2022**. [CrossRef]
24. Korendiy, V.; Kuzio, I.; Nikipchuk, S.; Kotsiumbas, O.; Dmyterko, P. On the Dynamic Behavior of an Asymmetric Self-Regulated Planetary-Type Vibration Exciter. *Vibroengineering PROCEDIA* **2022**, *42*, 7–13. [CrossRef]
25. Korendiy, V.; Gurey, V.; Borovets, V.; Kotsiumbas, O.; Lozynskyy, V. Generating Various Motion Paths of Single-Mass Vibratory System Equipped with Symmetric Planetary-Type Vibration Exciter. *Vibroengineering PROCEDIA* **2022**, *43*, 7–13. [CrossRef]

26. Nazarenko, I.; Gaidaichuk, V.; Dedov, O.; Diachenko, O. Investigation of Vibration Machine Movement with a Multimode Oscillation Spectrum. *East.-Eur. J. Enterp. Technol.* **2017**, *6*, 28–36. [CrossRef]
27. Yu, C.; Wang, X.; Pang, K.; Zhao, G.; Sun, W. Dynamic Characteristics of a Vibrating Flip-Flow Screen and Analysis for Screening 3 Mm Iron Ore. *Shock. Vib.* **2020**, *2020*, 1031659. [CrossRef]
28. Moncada, M.M.; Rodríguez, C.G. Dynamic Modeling of a Vibrating Screen Considering the Ore Inertia and Force of the Ore over the Screen Calculated with Discrete Element Method. *Shock. Vib.* **2018**, *2018*, 1714738. [CrossRef]
29. Chen, Z.; Tong, X.; Li, Z. Numerical Investigation on the Sieving Performance of Elliptical Vibrating Screen. *Processes* **2020**, *8*, 1151. [CrossRef]
30. Gursky, V.; Krot, P.; Korendiy, V.; Zimroz, R. Dynamic Analysis of an Enhanced Multi-Frequency Inertial Exciter for Industrial Vibrating Machines. *Machines* **2022**, *10*, 130. [CrossRef]
31. Gursky, V.; Kuzio, I.; Krot, P.; Zimroz, R. Energy-Saving Inertial Drive for Dual-Frequency Excitation of Vibrating Machines. *Energies* **2021**, *14*, 71. [CrossRef]
32. Czubak, P. Vibratory Conveyor of the Controlled Transport Velocity with the Possibility of the Reversal Operations. *J. Vibroengineering* **2016**, *18*, 3539–3547. [CrossRef]
33. Obertyukh, R.; Slabkyi, A.; Petrov, O.; Bakalets, D.; Sukhorukov, S. Substantiation of the Design Calculation Method for the Vibroturning Device. In *Advances in Design, Simulation, Manufacturing*; Ivanov, V., Trojanowska, J., Pavlenko, I., Rauch, E., Peraković, D., Eds.; Springer International Publishing: Cham, Switzerland, 2022; pp. 185–195.
34. Yaroshevich, N.; Puts, V.; Yaroshevich, T.; Herasymchuk, O. Slow Oscillations in Systems with Inertial Vibration Exciters. *Vibroengineering PROCEDIA* **2020**, *32*, 20–25. [CrossRef]
35. Yaroshevich, N.; Yaroshevych, O.; Lyshuk, V. Drive Dynamics of Vibratory Machines with Inertia Excitation. In *Vibration Engineering and Technology of Machinery*; Balthazar, J.M., Ed.; Springer International Publishing: Cham, Switzerland, 2021; pp. 37–47.
36. Yatsun, V.; Filimonikhin, G.; Podoprygora, N.; Pirogov, V. Studying the Excitation of Resonance Oscillations in a Rotor on Isotropic Supports by a Pendulum, a Ball, a Roller. *East.-Eur. J. Enterp. Technol.* **2019**, *6*, 32–43. [CrossRef]
37. Filimonikhin, G.; Yatsun, V.; Kyrychenko, A.; Hrechka, A.; Shcherbyna, K. Synthesizing a Resonance Anti-Phase Two-Mass Vibratory Machine Whose Operation Is Based on the Sommerfeld Effect. *East.-Eur. J. Enterp. Technol.* **2020**, *6*, 42–50. [CrossRef]
38. Liu, Y.; Zhang, X.; Gu, D.; Jia, L.; Wen, B. Synchronization of a Dual-Mass Vibrating System with Two Exciters. *Shock. Vib.* **2020**, *2020*, e9345652. [CrossRef]
39. Jannati, M.; Sutikno, T. Modelling of a 3-Phase Induction Motor under Open-Phase Fault Using Matlab/Simulink. *Int. J. Power Electron. Drive Syst. IJPEDS* **2016**, *7*, 1146–1152. [CrossRef]
40. Chaban, A.; Łukasik, Z.; Popenda, A.; Szafraniec, A. Mathematical Modelling of Transient Processes in an Asynchronous Drive with a Long Shaft Including Cardan Joints. *Energies* **2021**, *14*, 5692. [CrossRef]
41. Sudheer, H.; Kodad, S.F.; Sarvesh, B. Improved Sensorless Direct Torque Control of Induction Motor Using Fuzzy Logic and Neural Network Based Duty Ratio Controller. *IAES Int. J. Artif. Intell. IJ-AI* **2017**, *6*, 79–90. [CrossRef]
42. Yaroshevich, M.P.; Zabrodets, I.P.; Yaroshevich, T.S. Dynamics of Vibrating Machines Starting with Unbalanced Drive in Case of Bearing Body Flat Vibrations. *Nauk. Visnyk Natsionalnoho Hirnychoho Universytetu* **2015**, *3*, 39–45.
43. Wu, M.; Chen, F.; Li, A.; Chen, Z.; Sun, N. A Novel Vibration Isolator for Vibrating Screen Based on Magnetorheological Damper. *J. Mech. Sci. Technol.* **2021**, *35*, 4343–4352. [CrossRef]
44. Krot, P.; Zimroz, R.; Michalak, A.; Wodecki, J.; Ogonowski, S.; Drozda, M.; Jach, M. Development and Verification of the Diagnostic Model of the Sieving Screen. *Shock. Vib.* **2020**, *2020*, e8015465. [CrossRef]
45. Chen, B.; Yan, J.; Yin, Z.; Tamma, K.K. A New Study on Dynamic Adjustment of Vibration Direction Angle for Dual-Motor-Driven Vibrating Screen. *Proc. Inst. Mech. Eng. Part E J. Process Mech. Eng.* **2021**, *235*, 186–196. [CrossRef]