Study of the frequency multiplication of the oscillation of the milling machine blowout exciter

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Abstract. The article provides a theoretical discussion of frequency multiplication of oscillations in milling machines. Following the theoretical discussion, a power balance approximation for the vibration exciter of milling machines is presented. Expressions are determined for the oscillation amplitude under weak elastic and dissipative constraints imposed on the working organs of milling machines. The results of a theoretical experiment are analyzed and presented with sequences of numbers. The phase shift angle of the pendulum oscillation relative to the phase of the unbalance rotation is discussed. The spring stiffness coefficient and the damper resistance coefficient of milling machines are provided to maintain its equilibrium in stable operation.

INTRODUCTION

Vibration technology encompasses, first and foremost, machines, stands, devices, instruments, and tools in which deliberately induced vibration performs useful functions. In the following discussion, we will primarily focus on this particular aspect. The study of vibration phenomena is of greatest interest precisely during processing operations [1]. Details regarding the selection of optimal vibration sensor placement on lathes are considered in [2]. Furthermore, work [3] addressed the application of APC (Automated Process Control) systems for solving production tasks related to optimizing machining regimes and monitoring process discipline. It should be noted that the aspects of APC development and implementation discussed above pertain to turning operations. Their application on other types of equipment requires separate investigation. For instance, in the case of milling, the causes of vibration and its propagation through machine tool components have not been sufficiently studied. Work [4] examined issues of vibration monitoring during processing on a milling machine using a vibration sensor mounted on the cutting tool holder. To describe the motion of the shell, classical displacement equations [5, 6, 7] are used.

Researchers globally and in our country have conducted studies to enhance the efficient use of road transport and establish a scientific foundation for improving service quality [8]. Tests were conducted at the optimal temperature for each microorganism. The pH of the medium was maintained at 4.5–5.0 for yeast and 6.8–7.0 for bacteria. Experiments were performed both with and without the addition of a surface-active substance (surfactant), specifically technical sulfoureide, at a concentration of 0.05% [9]. Many industrial sectors, such as power generation (nuclear, hydroelectric, etc.), maritime transport, and aviation, utilize machinery containing very large metallic components. These parts are subject to wear and, over time, require machining to restore specific functional tolerances [10]. The dynamics of mobile machine tools have been studied in on-site environments [11-13]. The authors used substructure decoupling to develop a dynamic model of a complete mobile milling machine system, including its support structure. This model was subsequently used to predict chatter stability. Such models are particularly valuable when machining unique parts without prior process experience. A mobile milling machine has also been compared to a robotic arm for face milling operations [14]. There are many parts and components in engineering electric drive construction machinery, and the layout of structural parts is complex. The cooperation of various parts is the basis for effectively ensuring the stable and reliable operation of engineering electric drive construction machinery. Engineering machinery with electric drive systems comprises numerous complex components. The effective cooperation of these parts is essential for ensuring stable and reliable operation.

Engineering electric-drive construction machinery is the mechanical equipment that converts other forms of energy into electric energy [15]. The common engineering electric drive construction machinery mainly includes hydraulic turbines, steam turbines, diesel engines, etc. All forms of engineering electric-drive construction machinery transmit mechanical energy to engineering electric drive construction machinery through transmission engineering electric drive construction machinery, build the magnetic circuit and circuit of transmission engineering electric drive construction machinery through appropriate magnetic and conductive materials and push the piston downward to do work under the extrusion of the piston to realize energy conversion. Engineering electric drive construction machinery is widely used in industrial and agricultural production, national defense, science and technology, life and other fields that have important basic significance. In industrial production, a large number of fluid machinery, such as compressors, pumps, etc., often have abnormal vibration, which has a great impact on normal production [16]. The fault diagnosis technology widely used in the vibration diagnosis of rotating machinery is mostly based on signal analysis. When facing the abnormal vibration of on-site machines, it depends on the experience of on-site technicians to a great extent. Because there is no obvious and quantitative law between vibration causes and characteristic signals, and due to the limitations of production conditions, many fault sites cannot obtain complete detection data. Therefore, on-site diagnosis is a very difficult problem. The abnormal vibration of engineering electric drive construction machinery is divided into bending vibration, torsional vibration and axial vibration. Through the nonlinear dynamic analysis and abnormal feature extraction of engineering electric drive construction machinery, the state test and detection of engineering electric drive construction machinery are realized [17]. Furthermore, existing research struggles to effectively depict the dynamic evolution of the health status of the target object across various time scales. Consequently, the concept of digital twins has been introduced as a promising approach for predicting and managing the health of construction machinery. However, a fundamental challenge lies in modeling complex systems, which is also an inherent obstacle within the realm of digital twins. Traditional mechanistic models demand an extensive amount of specialized knowledge, making it arduous to encompass all the behaviors and rules of the system, especially for intricate systems featuring global or local unknowns.

Additionally, these mechanistic models face difficulties keeping up with changing system states or incurring prohibitively high update costs. Consequently, methods grounded in mechanistic models exhibit poor scalability and pose challenges in terms of verification [18]. Ke et al.'s groundbreaking proposal of a novel gear wear prediction scheme, designed for forecasting the remaining service life of gear transmission systems, serves as a strong research foundation for advancing detection methods in the field of engineering machinery. Simultaneously, Ma et al. pioneered the development of a digital twin model for a bearing test bench, leveraging multidisciplinary simulation. They adeptly identified closely spaced modal parameters through the application of a modal decomposition algorithm, enabling comprehensive bearing fault analysis across diverse domains. In the process of abnormal vibration of engineering electrical transmission construction machinery and equipment, the famous bathtub curve law is basically followed and the whole process includes a running-in period, a normal probation period and a wear and tear period. Through the necessary measurement and fault diagnosis of mechanical equipment, we can find the stage of the equipment at a certain point in time to avoid the equipment entering the loss fault in advance. Mechanical abnormal vibration diagnosis technology for mechanical equipment refers to the use of detection devices to detect the state information of mechanical equipment in operation or under relatively static conditions under a certain working environment, judge whether the mechanical equipment is in a normal operation state by analyzing the operation state information of mechanical equipment, and qualitatively and quantitatively judge the real-time operation state of mechanical equipment and its parts in combination with the fault mechanism and historical operation state of the diagnostic object.

Second, vibration technology encompasses equipment and devices for measuring, monitoring, and controlling vibration. Only thirdly does vibration technology include apparatus for preventing, suppressing, damping, or isolating harmful vibration. The development, creation, and refinement of vibration machines for various purposes have been made possible through the work of numerous researchers, mechanical engineers, designers, and technicians. Responding to technologists' demands for increased productivity, designers have addressed this by enlarging machine dimensions, increasing installed power, and raising the static moment and angular velocity of the unbalances in centrifugal vibration exciters.

METHODOLOGY

The study of milling machine dynamics is an important task in the design, modernization, and operation of metalworking equipment. In this context, the term "dynamics" refers to processes related to vibrations, oscillations, the motion of working parts (tools/workpieces), as well as the machine's stability and operational accuracy under various working conditions. For the analysis of these processes, three main groups of methods are used: theoretical, numerical, and experimental. Theoretical methods are based on the mathematical description of mechanical processes. These methods include the use of Lagrange equations, which enable the construction of motion models for systems with multiple degrees of freedom; harmonic analysis for studying frequency characteristics; and the development of single-mass and multi-mass models to assess stability and the nature of vibrations. These methods allow for the prediction of potential dynamic issues and the optimization of design parameters during the early stages of development. The entire variety of processes involving changes in a scalar quantity in mechanics can be divided into two classes: oscillatory processes and non-oscillatory processes. Generally speaking, the magnitude, spatial orientation, and temporal character of the forcing forces and moments generated by centrifugal vibration exciters depend on the motion of the machine's vibrational working parts, which are set into oscillation by these very exciters.

RESULTS

The task of creating a superharmonic vibration drive requires the implementation of a series of design measures. To clarify the approaches to this, we will examine several schemes with progressively increasing complexity. Initially, we will consider the scheme presented in Fig. 1.

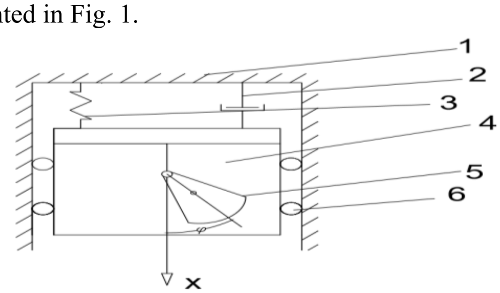

FIGURE 1. The balancing assembly of milling machines, featuring an adjustable unbalance mechanism

Here, the working organ 4 is connected to the fixed frame 1 via a spring 3 and a damper 2. The working organ, constrained by the ideal links 6, is set into oscillatory motion by the unbalance 5.

Let us take as generalized coordinates the displacement xx of the working organ from its equilibrium position and the rotation angle φφ of the unbalance. We then write the differential equation (neglecting gravity, which is insignificant since our further interest lies in the problem of amplifying the third harmonic of the oscillation in the working organ of milling machines):

$$\begin{cases} (m_1+m_2)\ddot{x} + b\dot{x} + cx - m_0 r(\ddot{\varphi}\sin\varphi + \dot{\varphi}^2\cos\varphi) = 0, \\ J\ddot{\varphi} - m_0 r\ddot{x}\sin\varphi = M; \end{cases} \quad (1)$$

b − is the damper resistance coefficient 2;

c − is the spring stiffness coefficient 3;

M − is the constant moment on the unbalance shaft;

Let's introduce the following dimensionless quantities:

$$\tau = \omega t;\ \xi = \frac{m_1+m_2}{m_0 r}x;\ \mu = \frac{M}{J\omega^2}; \quad (2)$$

$$\mathcal{H} = \sqrt{\frac{c}{(m_1+m_0)\omega^2}};\quad \beta = \frac{b}{2\gamma(m_1+m_2)\omega};\ \alpha = \frac{m_0 r}{\sqrt{J}(m_1+m_2)}; \quad (3)$$

here

$$\omega = \frac{2\pi}{T}$$

It represents the mean angular velocity of the unbalance rotation (where TT is the period of one revolution). Denoting further differentiation with respect to ττ by dots over ξξ and φφ, and substituting the quantities from (2) into equation (1), we obtain:

$$\ddot{\xi} + 2\beta\gamma\dot{\xi} + \gamma^2\xi - \ddot{\varphi}sin\varphi - \dot{\varphi}^2cos\varphi = 0 \quad (4)$$

$$\ddot{\varphi} - \alpha^2\ddot{\xi}\sin\varphi = 0 \quad (5)$$

Let us represent the angle φφ as the sum of a linear part and an oscillatory component:

$$\varphi = \tau + \psi; \quad (6)$$

From this it follows: $\dot{\varphi} = 1 + \dot{\psi},\ \ddot{\varphi} = \ddot{\psi}$ (7)

Substituting equalities (6) and (7) into equations (5),

$$\ddot{\xi} + 2\beta\gamma\dot{\xi} + \gamma^2\xi = \ddot{\psi}\sin(\tau + \psi) + (1 + \dot{\psi})^2\cos(\tau + \psi),$$

$$\ddot{\psi} - \alpha^2\ddot{\xi}\sin(\tau + \psi) = \mu \quad (8)$$

Next, we will assume that the angle ψ and its derivatives ψ, ψ', as well as the parameter α^2, are small compared to unity. Let us rewrite equations (8), retaining terms only up to the first order of smallness:

$$\begin{cases} \ddot{\xi} + 2\beta\gamma\dot{\xi} + \gamma^2\xi = \cos\tau + (\ddot{\psi} - \psi)\sin\tau + 2\dot{\psi}\cos\tau \\ \ddot{\psi} = \alpha^2\ddot{\xi}\sin\tau + \mu \end{cases} \quad (9)$$

Hereafter, we will use the method of successive approximations. For the first approximation,

we $\psi^{\Theta} = 0$ (10)

and, substituting this first approximation for ψ into the first equation of (8) and (9), we obtain the first approximation for ξ:

$$\xi^{\Theta} = \xi_{1a}^{\Theta}\cos(\tau + \tau_1^{\Theta}), \quad (11)$$

Where $\xi_{1a}^{\Theta} = \frac{1}{\sqrt{(\gamma^2-1)^2+4\beta^2\gamma^2}}$;

$$\tau_1^{\Theta} = \text{arctg}\frac{2\beta\gamma}{\gamma^2-1} \quad (12)$$

The subscripts for the amplitude and initial phase here and below denote the harmonic number. The superscript (Θ) denotes the first approximation; the second approximation is written without this subscript.

To obtain the second approximation ψ, we substitute the result (11) into the second equation (9):

$$\ddot{\psi} = -\frac{1}{2}\alpha^2\,\xi_{1a}^{\Theta}\sin(\tau + \tau_1^{\Theta}) - -\frac{1}{2}\alpha^2\,\xi_{1a}^{\Theta}\sin\tau_1^{\Theta} + \mu \quad (13)$$

The conditions for periodicity of solution (13) are the equality

$$\mu^{\Theta} = \frac{1}{2}\alpha^2\,\xi_{1a}^{\Theta}\sin\tau_1^{\Theta} \quad (14)$$

Which, in dimensionless form, yields the power balance, since the value of μ is proportional to the average power required to maintain oscillations.

Based on this condition, equation (13) takes the form

$$\ddot{\psi} = -\frac{1}{2}\alpha^2\,\xi_{1a}^{\Theta}\sin(2\tau + \tau_1^{\Theta}); \quad (15)$$

$$\begin{cases} \dot{\psi} = \frac{1}{4}\alpha^2\,\xi_{1a}^{\Theta}\cos(2\tau - \tau_1^{\Theta}); \\ = \boldsymbol{\frac{1}{8}\alpha^2\,\xi_{1a}^{\Theta}\sin(2\tau - \tau_1^{\Theta})}; \end{cases} \quad (16)$$

To find the second approximation for ξ, we substitute the relations into the first equation of (9), discarding quantities above the first order of smallness with respect to α^2.

$$\ddot{\xi} + 2\beta\gamma\dot{\xi} + \gamma^2\xi = (1 - \frac{\alpha^2\,\xi_{1a}^{\Theta}}{16}\,\tau_1^{\Theta})\cos(\tau - \chi) + \frac{9}{16}\alpha^2\,\xi_{1a}^{\Theta}\cos(3\tau - \tau_1^{\Theta}), \quad (17)$$

Here $\chi = \text{arctg}(\frac{\alpha^2\,\xi_{1a}^{\Theta}}{16}\,\sin\tau_1^{\Theta})$ (18)

The second approximation of ξ is given by the integral of the differential equation (17)

$$\xi = \xi_{1a}\cos(\tau - \tau_1) + \xi_{3a}\cos(3\tau - \tau_3) \quad (19)$$

here $\xi_{1a} = \xi_{1a}^{\Theta}\left(1 - \frac{\alpha^2}{16}\cos\tau_1^{\Theta}\right);\ \tau_1 = \tau_1^{\Theta} + \chi;$ (20)

$$\xi_{3a} = \frac{9\alpha^2\,\xi_{1a}^{\Theta}}{16\sqrt{(\gamma^2-9)^2+36\beta^2\gamma^2}};\ \tau_3 = \tau_1^{\Theta} + \text{arctg}\frac{6\beta\gamma}{\gamma^2-9}\chi; \quad (21)$$

Substituting integral (19) into the second equation of (9), we find the second approximation for the power balance:

$$\mu = \frac{1}{2}\xi_{1a}\sin\tau_1 \quad (22)$$

It is particularly important to note that the displacement of the mean position during oscillations of the working organ is determined by the centrifugal inertia force developed by the pendulum during its swings, which was not accounted for above. The instantaneous value of this centrifugal force is determined from the formulas.

$$\rho = (m_1a + m_0l)\dot{\psi}^2 \quad (23)$$

where, **a -** is the distance to the center of mass of the unbalanced element, **l** is the length of the unbalance shaft, The oscillations of the pendulum occur according to the following law:

$$\psi = \psi_a \sin(\omega t - \theta_1) \quad (24)$$

θ_1 −phase shift angle of the pendulum oscillations relative to the phase of the unbalance rotation; ψ_a − the angular amplitude of the pendulum oscillations, which, under weak elastic and dissipative constraints imposed on the pendulum, can be represented by the relationship

$$\psi_a = \frac{m_0 r}{m_1 a + m_0 l} \quad (25)$$

$\psi_a < 0,1$ and therefore our proposal about the smallness of the angle ψ is justified.

Using dependencies (24) and (25), we rewrite equality (23) in the following formи

$$\rho = \frac{(m_0 r)^2 \omega^2}{m_1 a + m_0 l}(\cos^2(\omega t - \theta_1) \quad (26)$$

Or it can be written in the following form

$$\rho = \rho_{cp} + \rho_{cp}\cos 2(\omega t - \theta_1) \quad (27)$$

Where is the average value of centrifugal force

$$\rho_{cp} = \frac{(m_0 r)^2 \omega^2}{2(m_1 a + m_0 l)} = \frac{1}{2}\psi_a\, m_0 r \omega^2 \quad (28)$$

This force will cause a shift in the average position of the working element by an amount

$$x_{cp} = \frac{x_{cp}}{c_x} = \frac{(m_0 r)^2 \omega^2}{2(m_1 a + m_0 l)c_x} = \frac{\psi_a m_0 r \omega^2}{2c_x} \quad (29)$$

As shown by the second term on the right side of equality (27), the working element, under the action of the centrifugal force of the pendulum, will perform additional oscillations with double the frequency. The amplitude of these oscillations, under weak elastic and dissipative constraints imposed on the working element, can be determined by the expression

$$x_{2a} = \frac{\rho_{cp}}{(m_2 + m_1 + m_0)\omega^2} = \frac{(m_0 r)^2}{2(m_1 a + m_0 l)(m_2 + m_1 + m_0)} \quad (30)$$

Under the same conditions, the amplitude of the fundamental tone of the oscillations, as follows from the differential equation

$$(m_2 + m_1 + m_0)\ddot{x} + b_x \dot{x} + c_x x = m_0 r \omega^2 \cos\omega t; \quad (31)$$

Discussing (30) of the differential equation we find

$$x_a = \frac{m_0 r}{(m_2 + m_1 + m_0)} \quad (32)$$

Comparing these two equalities, we find $x_{2a} = \frac{1}{2}\psi_a x_a$ (33)

DISCUSSION

The amplitude of the second harmonic is usually small compared to the amplitude of the fundamental oscillation of the working element. However, the shift in the mean position of the working element, determined by relation (29), can be quite significant with a low stiffness coefficient cx*cx*. The use of a power of 140 J/s helps optimize the performance of milling machines. This, in turn, helps to stabilize the production process and achieve higher quality results.

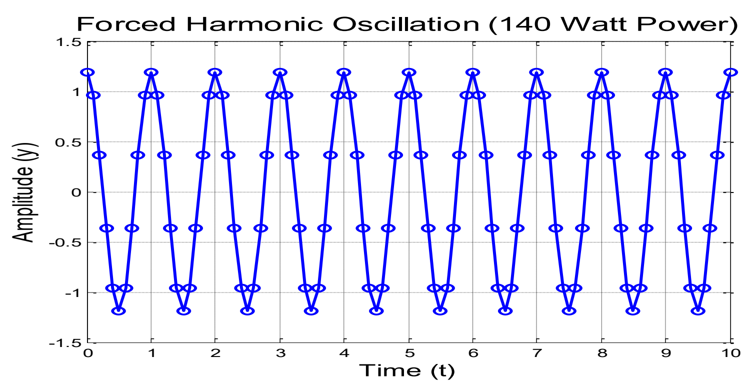

FIGURE 2. Forced oscillations with constant power, where energy (power) is used to calculate the amplitude of milling machines equipped with vibration exciters

The use of vibration exciters significantly facilitates the operation of milling machines, improving safety and working conditions for operators. This minimizes delays and obstacles, creating a comfortable environment for employees. The use of vibration exciters allows for the automation of milling machines, increasing production speed and precision. This helps ensure competitiveness and enables differentiation based on product quality. Its graphical function, dependent on time, is shown in Figure 2.

Let's analyze the obtained results. Numerical values of certain data for an accurate representation of the graphical relationship are provided in Table 1.

TABLE 1. Numerical values of some data for the vibration drive of milling machines.

№	Designations	Parameter name	Numerical values or their limits
1	β	Relative attenuation of the vibration drive	$0.1 \div 0.2$
2	γ	Vibration drive vicinity	$1 \div 3$
3	ξ	Dimensionless quantity	1.68
4	ψ	Vibratory drive imbalance angle	$35^0 \pm 5^0$
5	τ	Harmonic of the vibration drive of milling machines	48 Hz
6	ω	Angular velocity of rotation of the unbalance	4π rad/s
7	t	Time (for calculation)	120sec
8	m_1	The mass of the pendulum having a vibration drive	0.2 kg
9	m_0	Vibration drive imbalance mass	1.2 kg
10	$\mathcal{H}$	Dimensionless quantity	4.1
11	J	Moment of inertia of the unbalance relative to the axis of rotation	$0.002\text{kg} \cdot \text{m}^2$
12	r	Eccentricity of the imbalance mass relative to the axis of rotation	1.4m
13	μ	Power balance of milling machines using vibration exciters	140 J/sec
14	c	Spring stiffness coefficient 3	200N/m
15	m_2	The mass of the working element having a vibration drive	0.8 kg
16	α^2	A parameter dependent on the moment of inertia, the mass of the pendulums and the mass of the unbalance of the vibration drive of milling machines	1410
17	b	Damper resistance coefficient	$120\ \text{N} \cdot \text{s/m}$
18	M	Constant moment on the unbalance shaft	40Nmm

CONCLUSION

Dynamic calculation of the oscillation frequency of milling machine vibration exciters enables the design of such machines with increased reliability and operational dependability. The conducted analysis of the energy balance of a milling machine incorporating a vibration-exciting device, which supplies 140 J/s to the system, demonstrated the effectiveness of this approach for stabilizing the dynamic characteristics of the equipment. The additional energy input allows for the compensation of internal losses occurring during the machining process and contributes to improved cutting stability. A reduction in the negative impact of vibrations is observed, which, in turn, positively affects surface quality and machine reliability. Thus, the use of controlled vibration excitation proves to be a well-founded technical solution that enhances the operational efficiency of the equipment under real-world conditions.

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