Transient oscillatory processes at the balancing device operation of abrasive wheel grinder

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Abstract. In conditions of the practice development of building new production facilities following the concept of Industry 4.0 and the intensification and complication of production processes, the need for a general reduction in costs arises. One of these problems is the lengthening the life of supports of abrasive wheel grinding machines based on the balancing process automation without stopping the machines for maintenance work. This paper proposes a method for numerical calculation of the motion path of the grinding wheel spindle axis during the balancing process. Based on the proposed method, a numerical calculation was carried out and the motion path of the grinding wheel centre was built. As a result of the work, a transient balancing oscillatory process of a damping mode was built. This process does not lead to the forcing of the system in the eigenmode range, which creates prospects for the introduction of this balancing method in the grinding process and reducing production costs.

1. Automatic balancing of the grinding wheel

The modern concept of Industry 4.0 is based on digitalization and intellectualization of production processes. As a result, an increasing number of solutions from the field of information technology is used in production, such as digital twins, the industrial Internet, the Internet of things, artificial intelligence and others [1,2].

Among the various trends, we can emphasize the importance of the growth of production data amount and their available volume for subsequent analysis. This trend is rapidly developing the practice of using digital twins for analysis and information support of production processes [3], as a result, manufacturing monitoring is also being improved and developed [4, 5].

The majority of scientific and engineering investigations in the field of industrial digital twins, in addition to large sets of technical and production data, include various models of objects and processes [5–7]. The application practice of these technologies allows us to spend more efficient production management, including using already known technologies, for example, operator interfaces [8].

It is often necessary to perform different mechanical engineering products grinding in the production using grinding machines equipped with grinding wheels. Although this type of processing has used for a long time, it continues to improve and develop due to several unsolved problems [9] associated with the presence of transient oscillatory processes [1], for which the solution of the problems of ensuring their stability remains relevant [10,11].

There are few works in the scientific papers devoted to solving this problem, but taking into account related areas, we can note the existing experience of balancing vehicle wheels [12], including using
machine equipment [13], industrial balancing of grinding wheels using acoustic emission signals [14], the grinding wheel live prediction [15], the study on the surface quality after this processing [16], online balancing for reducing the unbalance [17] and others [18,19].

The publications’ analysis has confirmed that during the grinding wheel processing it becomes necessary to balance it and different balancing devices are used for this purpose, including auto-balancing ones.

Usually, to calculate the mass centre motion path of a grinding wheel during the balancing process, the real system «spindle-grinding wheel» is determined as a model with lumped parameters. This schematization way is simple and very convenient in the early stages of balancing device design. Also, certain errors introduced into the calculation by such a schematization force us to look for more accurate ways to the performance assessment of the designed balancing device.

Let’s consider the operation of an auto-balancing device with two annular corrective masses that are not kinematically interconnected and move alternately around the circumference. Such a scheme is used in some modern balancing devices, such as 3L175. Since such a device spends the balancing process under all operating modes of the machine, including grinding, it is important to consider the transient oscillatory process, since it directly affects the quality of parts processing.

The grinding wheel fixed on the spindle independent chuck, rotating with an angular velocity \( \Omega \), has its mode imbalance vector \( \vec{P}_0 \). Balancing by the device proceeds as follows. First, two adjusting wheels coaxial with the circle begin to rotate alternately, in the direction of reducing vibrations. One wheel, having an imbalance \( \vec{P}_1 \), is motionless relative to the circle at the first moment; the other wheel having an imbalance \( \vec{P}_2 \) rotates with the speed exceeding the circle rotation speed by \( \omega \), and \( \vec{P}_1 = \vec{P}_2 \). So, turning one of the corrective masses, we achieve the imbalance partial reduction, then we turn the other than the first one again and so on until the vibration drops to the lowest possible level.

2. Mathematical model of grinding wheel balancing

2.1. Thesaurus

To build a mathematical model of balancing, we introduce a number of designations and summarize them into table 1.

| Designation | Name                                                                 | Dimensionality |
|-------------|----------------------------------------------------------------------|----------------|
| \( m \)    | the grinding wheel weight with independent chuck and corrective masses | kg             |
| \( \vec{j}_m \) | mass moment of inertia                                                | kg m^2         |
| \( m_i \)  | belt-drive weight                                                     | kg             |
| \( m_0 \)  | spindle weight distributed along the length                           | kg / m         |
| \( \vec{E} \) | first-kind elasticity modulus (Young's modulus) of the spindle material | N / m^2       |
| \( J \)    | moment of inertia of the spindle cross-section                        | m^4            |
| \( \vec{P}(t) \) | the total load caused by the imbalance of the grinding wheel and the rotating imbalances of the corrective masses during the balancing process | N             |
| \( \omega \) | rotation speed exceeding of the second abrasive wheel                 | s^-1           |
| \( \Omega \) | grinding wheel rotation speed                                          | s^-1           |
| \( \vec{P}_0 \) | eigenmode imbalance of the grinding wheel                             | N              |
| \( \vec{P}_1 = \vec{P}_2 \) | Imbalances of corrective wheel coaxial with a circle                    | N              |
2.2. Mathematical model development

The design model of the grinding wheel spindle with the installed balancing device is shown in figure 1.

![Design model of the grinding machine spindle](image)

Figure 1. Design model of the grinding machine spindle.

The equation of beam deflection has the form as follows:

$$EJ\frac{\partial^4 x}{\partial z^4} + m_0 \frac{\partial^2 x}{\partial t^2} = P(z,t)$$

(1)

There is a forcing load on the right-hand side of equation (1). This load should be taken in a series expansion of natural vibration modes. To do this, we represent it as a product of two quantities, one depends only on the coordinate $z$, and the other only on the time $t$.

The time-dependent factor is defined as:

$$P(t) = 2P_2 \cos\left(\alpha + \frac{\omega t}{2}\right),$$

where $\alpha$ is the angle between the vector $\vec{P}_0$ and the sum of the vectors $\vec{P}_1$ and $\vec{P}_2$ at the initial moment.

It is necessary to pay attention to the fact that the force $P$ at each moment is applied at one specific point. Mathematically, such force application can be described using the Dirac delta-function (figure 2).

![The Dirac delta-function](image)

Figure 2. The Dirac delta-function.

The delta-function allows us to describe a load concentrated or applied at one point. A feature of using the delta-function in our case is the ability to concentrate the application of force at one specific point. There will be no force at all other points in this situation, at any given moment. Using the delta-function, the right-hand side of equation (1) can be represented in the following form:

$$P(z,t) = P(t) \cdot P(z - l_3)$$

(2)

Since the natural modes of vibration of the spindle axis cannot be precisely determined, we set them in a form corresponding to the design scheme (figure 1):

$$X_i = f(K_i, z),$$

(3)

where $K_i$ is the eigenvalue of the $i$-th mode of vibration.

Since the solution $x(z, t)$ takes in a series expansion in terms of the eigenmodes:

$$x(z, t) = \sum_{i=1}^{\infty} X_i(z) \cdot T_i(t),$$

(4)
lets us take in a series expansion the disturbing load into its eigenmodes. Since we are looking for a solution in the form of a series of harmonic components, the delta-function should be also taken in a Fourier series.

In general, the function will be taken into a Fourier series by the formula:

\[ f(z) = \frac{a_0}{2} + \sum_{k=1}^{\infty} a_k \cos \frac{k\pi z}{l} + \sum_{k=1}^{\infty} b_k \sin \frac{k\pi z}{l}, \]

where

\[ a_0 = \frac{1}{l} \int_0^l f(z)dz, \quad a_k = \frac{1}{l} \int_0^l f(z) \cos \frac{k\pi z}{l}dz, \quad b_k = \frac{1}{l} \int_0^l f(z) \sin \frac{k\pi z}{l}dz. \]

Let us take the delta function in a series in its eigenmodes, using only sinuses, since it can quite be considered odd:

\[ \delta(z - l_3) = \sum_{i=1}^{\infty} b_i \sin \frac{i\pi z}{2l_3}, \]

where \( b_i \) is coefficient of the \( i \)-th harmonic component of the decomposition.

Multiply the left and right sides of this equality by \( \sin \frac{i\pi z}{2l_3} \) and integrate along the length:

\[ \int_0^{l_3} \sin \frac{i\pi z}{2l_3} \cdot \delta(x - \pi iz) dz = \sum_{i=1}^{\infty} b_i \cdot \int_0^{l_3} \sin \frac{i\pi z}{2l_3} \cdot \sin \frac{i\pi z}{2l_3} dz. \]

Due to the orthogonality of the functions \( \sin \frac{i\pi z}{2l_3} \), we get:

\[ \sin \frac{i\pi z}{2l_3} = b_l \int_0^{l_3} \frac{1 - \cos \frac{i\pi z}{2l_3}}{2} dz, \quad \int_0^{l_3} \sin \frac{i\pi z}{2l_3} \cdot \delta(z - \pi it) dz = \sum_{i=1}^{\infty} b_i \int_0^{l_3} \sin \frac{i\pi z}{2l_3} \cdot \sin \frac{i\pi z}{2l_3} dz, \]

where \( \sin \frac{i\pi z}{2l_3} = \frac{i}{2} \cdot l_3 \) and, as a result, decomposition coefficient will be: \( b_i = \frac{2}{l_3} \sin \frac{i\pi z}{2l_3} \).

Since the natural modes of vibration of the beam (spindle) \( x_i \) are chosen arbitrarily, they should not satisfy the deflection equation (1). Therefore, to find the time function \( T_i \) included in expression (4), one should use any method of computational mechanics, for example, the Galerkin method:

\[ \int_0^{l_3} \left[ EJx_i^{IV} + m_0x_i + P(t) b_i x_i \right] dx = 0, \]

whence \( T_i \) is found up to arbitrary constants of integration determined from the initial conditions:

\[ t = 0 \text{ and } z = l_3 \begin{cases} x(l_3, 0) = x_0, \\ \dot{x}(l_3, 0) = \Omega x_0. \end{cases} \]

where \( t = 0 \) – is the initial moment,

\( x_0 \) – is the initial axis spindle in point \( l_3 \).

To use these nonzero initial conditions, it is necessary to take \( x_0 \) in a series in terms of natural vibration modes:

\[ x_0 = \sum_{i=1}^{\infty} a_i x_i. \]

To find the series coefficients \( a_i \), you need to multiply the left and right sides of expression (10) by \( x_i \) and integrate over the length of the beam. Since \( t = \frac{\Psi}{\Omega + \omega} \), where \( \Psi \) is the spindle rotation angle, taking into account the initial conditions, we obtain the expression for \( x \):

\[ x(z, \Psi) = \sum_{i=1}^{\infty} x_i(z) \cdot T_i(\Psi), \]

which allows you to build the motion path of the spindle axis during balancing depending on the angle of rotation (figure 3).
2.3. The numerical experiment
Using the described methodology, a numerical calculation of the trajectory of the grinding wheel spindle axis during the balancing process was carried out. The calculation was carried out 3L175 machine with integrated balancing device and 600 mm installed grinding wheel. The motion path of the grinding wheel center, built from the calculated data, is shown in figure 3.

![Figure 3. The motion path of the grinding wheel center.](image)

The calculation proved that the transient process during balancing is attenuated, that it does not lead to the energization of the system in the natural frequency range, and it opens up possibilities for a wide range of automatic balancing of grinding wheels during processing without removing the wheel from the machine.

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