АНАЛИЗ ЭФФЕКТИВНОСТИ РАБОТЫ КЛАПАННЫХ УЗЛОВ ДИФФЕРЕНЦИАЛЬНОГО РАСТВОРОНАСОСА

Методика расчета эффективности работы клапанов основана на анализе конструкции клапанных узлов и определении времени их открытия и закрытия. Угол срабатывания рассмотрен как универсальный показатель, позволяющий сравнивать между собой насосы разных конструкций с различными технологическими характеристиками и позволяющий усовершенствовать их конструкцию, добиваясь уменьшения этого показателя.

Ключевые слова: дифференциальный растворонасос, угол срабатывания клапанов, клапанный узел, оптимизация конструкции клапанов.

Pavelieva Anna, PhD, Associate Professor, Department of Foreign Philology and Translation, Poltava National Technical Yuri Kondratyuk University, Ukraine, e-mail: kunsite.zi@gmail.com, ORCID: http://orcid.org/0000-0002-2306-1928

Vasyliev Ievgen, PhD, Associate Professor, Department of Construction Machinery and Equipment, Poltava National Technical Yuri Kondratyuk University, Ukraine, e-mail: vas.eugene@gmail.com, ORCID: http://orcid.org/0000-0001-5133-3989

Popov Stanislav, PhD, Associate Professor, Department of Manufacturing Engineering, Poltava National Technical Yuri Kondratyuk University, Ukraine, e-mail: psv@pntu.edu.ua, ORCID: http://orcid.org/0000-0003-2381-152X

Vasyliev Anatoly, PhD, Associate Professor, Department of Manufacturing Engineering, Poltava National Technical Yuri Kondratyuk University, Ukraine, e-mail: vas.anatoly@gmail.com, ORCID: http://orcid.org/0000-0002-1767-8569

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Hnitko S., Shpylka A., Shpylka N., Kravchenko S.

MATHEMATICAL MODELING OF VIBRATIONAL SYSTEMS FOR TRANSVERSE GRINDING BY WHEEL PERIPHERY

Розглянуто модель вібраційної системи шліфування, яка дозволяє простежити основні закономірності руху, одержати аналітичні залежності руху шліфувальних кругів, перехід системи через проміжні резонанси, шляхи перетворення й розсіювання енергії в системі. Дана модель створює передумови для підвищення ефективності процесу шліфувальної обробки.

Ключові слова: *вібраційна система шліфування, закономірності руху, шліфувальний круг.*

1. Introduction

When cutting of metals in the system machine – tool – device – part vibrations occur. They lead to a decrease in the quality of the machined surfaces. In order to apply effective methods of quenching these vibrations, it is necessary to develop rational methods for determining their amplitude-frequency characteristics.

One of the reasons for occurrence of mechanical vibrations during grinding operations is radial runout on the grinding wheel. Radial runout occurs when the grinding wheel is first mounted on the machine in the absence of straightening and balancing. Also, the radial runout increases after a certain time of the grinding operation due to the appearance of a ripple on the working surface of the wheel [1–4].

The statement of problems of the dynamics of vibrational systems can be very diverse. In the most general formulation, these problems are reduced to investigating the motion of the vibrating machine itself and the drive system – the vibrating machine.

The relevance of the research is creation of a mathematical model that will allow to investigate the motion of the grinding system and obtain dependencies for determining the main parameters of the system.

2. The object of research and its technological audit

The object of research is the vibrational grinding system.

The vibrational grinding system is a stochastic system, so it is very difficult to determine its state at any given time. One of the problem areas of the research object is the determination of the way in which the regularities of mutual influence of the main parameters of the system's motion will be taken into account.

The question of taking into account the energy dissipation in elastic elements in the study of the vibrations of an elastic system is quite complex, since internal friction depends on a number of factors whose influence is quite complex and practically not subject to direct account. Among the many hypotheses describing dissipative forces, the Kelvin-Voigt hypothesis has recently become most widespread.

With comparative simplicity, this hypothesis with sufficient accuracy characterizes the dissipative forces arising during the deformation of elastic bonds and ensures sufficient convergence with the experimental data [5]. Therefore, in this study let's use this hypothesis to describe the forces that arise in the elastic bonds of the vibrational system.

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3. The aim and objectives of research

The aim of research is obtaining a mathematical model of the vibrational grinding system, which allows tracing the basic laws of motion, obtaining analytical dependencies of the motion of grinding wheels, transition of the system through intermediate resonances, ways of transforming and dissipating energy in the system.

To achieve this aim it is necessary:

1. To obtain analytical dependencies of the grinding system motion with longitudinal feed.

2. Using the Nielsen equations, to obtain the dependence of the vibrational system motion without taking into account the roughness of the grinding wheel profile.

4. Research of existing solutions of the problem

The three-phase asynchronous motor is used as the driving motor of the grinding machine. Since any real energy source has a limited power, it becomes necessary to take into account the processes of interaction with the vibrational machine. The importance and complexity of this problem have been determined by a large number of studies devoted to it [6–12].

The authors of [13] point out that the dynamic processes in the mechanical part of the vibrational system and the dynamic phenomena in the electric motor are in direct connection.

However, in studies [14] it is shown that, provided that the electromechanical time constant is more than three times the value of the electromagnetic time constant, the influence of electromagnetic processes on the dynamics of the mechanical system can be ignored. Thus, it becomes possible to use in the calculation not a dynamic but a static characteristic of the engine.

Research [15] show that when using induction systems of asynchronous electric motors, the electromagnetic time constant is negligibly small compared with the electromechanical time constant.

The author of [16] suggests an approach based on the use of static characteristics of energy sources in performing dynamic calculations.

The authors of [17] point out that the motion law of the rotor of a drive motor looks like this:

$$\varphi=\omega\cdot t+d(t), \tag{1}$$

where $d(t)$ – a periodic function of time.

Thus, periodic vibrations are superimposed on uniform rotation of the rotor with angular velocity ω, but the magnitude of these vibrations is small, which is confirmed on the basis of experimental studies [18].

5. Methods of research

In the course of the work, analytical methods of research are used, based on the basic principles of the theory of mechanical vibrations and theoretical developments of scientists in this field.

In the present work, when compiling the differential equations of motion of the vibration grinding system, the question of taking into account the unevenness of the rotation of the drive shafts of machine tools is not specially raised, but the chosen design scheme should allow for such accounting.

6. Research results

Let's consider a vibrational system with transverse grinding. Then the grinding wheel will rotate in planes perpendicular to the direction of the workpiece movement. Let's consider the motion of the vibrational system with a workpiece rigidly fixed on the machine, so that the vibratory system can be considered a suited model. The calculation scheme of the vibrational system is adopted in the form of a system with four degrees of freedom (Fig. 1).

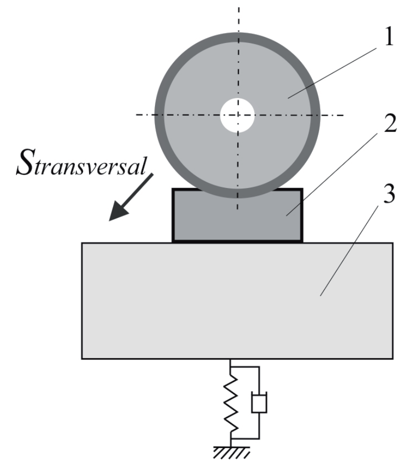

Fig. 1. Scheme of the vibrational system with transverse grinding: 1 – grinding (diamond) wheel; 2 – sample; 3 – clamping device

The generalized coordinates are: vertical displacement z, horizontal displacements x and y and angle of rotation φ of the grinding wheel. The question of the connection between the main masses moving in a real vibrational system is quite complex, and it is practically impossible to determine all the factors that describe the interaction. Therefore, in order to determine the main regularities of the motion of the vibrational system, design schemes are adapted. They are based on assumptions common to most applied problems of the dynamics of vibrational systems.

In this case, the main masses of the moving are represented by absolutely rigid ones. The masses of the elastic bonds, because of their relative smallness, are not taken into account in the calculations.

To construct a mathematical model of the vibrational system, let's use Nielsen equations. In this case, they will have the following form:

$$\frac{\partial\dot{T}}{\partial\dot{x}}-2\frac{\partial T}{\partial x}=Q_x;\quad \frac{\partial\dot{T}}{\partial\dot{y}}-2\frac{\partial T}{\partial y}=Q_y;$$

$$\frac{\partial\dot{T}}{\partial\dot{z}}-2\frac{\partial T}{\partial z}=Q_z;\quad \frac{\partial\dot{T}}{\partial\dot{\varphi}}-2\frac{\partial T}{\partial\varphi}=Q_\varphi, \tag{2}$$

where T – the kinetic energy of the system; $\dot{T}$ – the total time derivative of the kinetic energy; Q_x, Q_y, Q_z, Q_φ – generalized forces corresponding to generalized coordinates.

Let's find the kinetic energy of the system as the sum of the kinetic energies of the feed mechanism in translational motion:

$$T_k = \frac{mv_0^2}{2} = \frac{m\left(\dot{x}^2 + \dot{y}^2 + \dot{z}^2\right)}{2};$$

and rotating parts that are in a complex (flat) motion:

$$T_D = \frac{m_u v_c^2}{2} + \frac{I_{cx}\dot{\varphi}^2}{2},$$

where v_c – the velocity of the point C (the center of mass of the unbalanced parts); I_{cx} – the moment of inertia of the unbalanced parts with respect to the axis passing through the point C.

The velocity of point C consists of the portable and relative velocities:

$$\vec{v}_c = \vec{v}_e + \vec{v}_r; \quad \vec{v}_e = \vec{i}\dot{x} + \vec{j}\dot{y} + \vec{k}\dot{z};$$

$$\vec{v}_r = \vec{j}\, v_r \cos\varphi - \vec{k}\, v_r \sin\varphi; \quad v_r = \dot{\varphi}e.$$

In this case:

$$v_c^2 = \dot{x}^2 + \left(\dot{y} + \dot{\varphi}\, e\cos\varphi\right)^2 + \left(\dot{z} - \dot{\varphi}\, e\sin\varphi\right)^2.$$

Hence the total kinetic energy of the system will be:

$$T = \frac{m + 2m_u}{2}\left(\dot{x}^2 + \dot{y}^2 + \dot{z}^2\right) +$$
$$+ m_u e\dot{\varphi}\left(\dot{y}\cos\varphi - \dot{z}\sin\varphi\right) + \frac{I_{rx}\cdot\dot{\varphi}^2}{2}, \quad (3)$$

where $I_{rx} = I_{cx} + m_u e^2$.

Let's define the total derivative of the kinetic energy with respect to time:

$$\dot{T} = \left(m + 2m_u\right)\cdot\left(\ddot{x}\cdot\dot{x} + \ddot{y}\cdot\dot{y} + \ddot{z}\cdot\dot{z}\right) +$$
$$+ m_u e\ddot{\varphi}\left(\dot{y}\cos\varphi - \dot{z}\sin\varphi\right) + m_u e\dot{\varphi}\left(\ddot{y}\cos\varphi - \ddot{z}\sin\varphi\right) -$$
$$- m_u e\dot{\varphi}^2\left(\dot{z}\cos\varphi + \dot{y}\sin\varphi\right) + I_{rx}\cdot\dot{\varphi}\cdot\ddot{\varphi}.$$

Let's write the partial derivatives that appear in the left-hand side of the Nielsen equations:

$$\frac{\partial\dot{T}}{\partial\dot{x}} = \left(m + 2m_u\right)\ddot{x};$$

$$\frac{\partial\dot{T}}{\partial\dot{y}} = \left(m + 2m_u\right)\ddot{y} + m_u e\ddot{\varphi}\cos\varphi - m_u e\dot{\varphi}^2\sin\varphi;$$

$$\frac{\partial\dot{T}}{\partial\dot{z}} = \left(m + 2m_u\right)\ddot{z} - m_u e\ddot{\varphi}\sin\varphi - m_u e\dot{\varphi}^2\cos\varphi;$$

$$\frac{\partial\dot{T}}{\partial\dot{\varphi}} = I_{ox}\ddot{\varphi} + m_u e\left[\left(\ddot{y}\cos\varphi - \ddot{z}\sin\varphi\right) - 2\dot{\varphi}\left(\dot{y}\sin\varphi + \dot{z}\cos\varphi\right)\right];$$

$$\frac{\partial T}{\partial x} = \frac{\partial T}{\partial y} = \frac{\partial T}{\partial z} = 0;$$

$$\frac{\partial T}{\partial\varphi} = -m_u e\dot{\varphi}(\dot{y}\sin\varphi + \dot{z}\cos\varphi).$$

Let's define generalized forces corresponding to generalized coordinates of the system:

$$Q_x = T_n - P^x; \quad Q_y = -P_{fr}^y; \quad Q_z = -cz - \alpha\dot{z};$$

$$Q_\varphi = -m_u g e\sin\varphi + M_E - M_R, \quad (4)$$

where C – stiffness coefficient characterizing the elastic forces arising in the system; α – coefficient characterizing the scattering of energy.

Let's define the projection of the cutting force on the Z, X and Y axes as follows [15]:

$$P_{x(z)} = \frac{N_w 1000\eta_1}{V_w};$$

$$P_{y(x)} = \frac{\left(0,1...0,2\right)N_w 1000\eta_1}{V_w}, \quad (5)$$

where N_w – the power required to rotate the wheel, kW; η_1 – drive efficiency of the diamond wheel; V_w – linear velocity of the wheel rotation:

$$V_w = \dot{\varphi}r,$$

where r – the radius of the grinding wheel.

Let's substitute the results obtained in the Nielsen equation and obtain the result:

$$\left(m + 2m_u\right)\ddot{x} = T_n - P_x; \quad (6)$$

$$\left(m + 2m_u\right)\ddot{y} + m_u e\ddot{\varphi}\cos\varphi - m_u e\dot{\varphi}_2\sin\varphi = -P_y; \quad (7)$$

$$\left(m + m_u\right)\ddot{z} - m_u e\ddot{\varphi}\sin\varphi - m_u e\dot{\varphi}_2\cos\varphi = -cz - \alpha\dot{z}; \quad (8)$$

$$I_{rx}\ddot{\varphi} + m_u e\left(\ddot{y}\cos\varphi - \ddot{z}\sin\varphi\right) = -m_u g e\sin\varphi + M_E - M_R. \quad (9)$$

The mathematical model in this case will look like this:

$$\left(m + 2m_u\right)\ddot{x} - T_n + P_x = 0; \quad (10)$$

$$\left(m + 2m_u\right)\ddot{y} + P_y = -m_u e\frac{d^2}{dt^2}\sin\varphi; \quad (11)$$

$$\left(m + 2m_u\right)\ddot{z} + cz + \alpha\dot{z} = -m_u e\frac{d^2}{dt^2}\cos\varphi; \quad (12)$$

$$I_{rx}\ddot{\varphi} + m_u e\left(\ddot{y}\cos\varphi - \ddot{z}\sin\varphi\right) + m_u g e\sin\varphi = M_E - M_R, \quad (13)$$

where M_E – the engine torque applied to the machine shaft; m – the mass of the moving parts of the machine; m_u – the mass of unbalanced parts; M_R – the moment of the resistance forces on the machine shaft; I_{rx} – the moment of inertia of the rotating parts, relative to the axis of the machine shaft.

From the experience of solving simpler problems, let's conclude that it is impossible to express the solution of the reduced systems of differential nonlinear equations either in elementary functions or in quadratures, therefore it is expedient to use numerical methods to analyze the compiled mathematical models.

7. SWOT analysis of research results

Strengths. Among the strengths of this research, it should be noted that with the help of the obtained model of the vibrational grinding system, it is possible to analytically investigate the interdependence of the main grinding parameters to determine their effective relationship based on the analysis of the obtained dependences.

Weaknesses. The mathematical model of the vibrational system is developed without taking into account the uneven profile of the grinding wheel, leads to a certain disagreement between the adopted model and the real system.

Opportunities. Additional opportunities to achieve the aim of research are reducing the volume of experimental studies to verify the analytical dependencies obtained for grinding operations. Reducing the volume of experimental research will result in saving resources and time.

Threats. Threats in the implementation of the completed developments are related to the following factors.

The first of these – grinding operation is a stochastic process, therefore it is practically impossible to take into account all the factors affecting the vibrational system.

The second factor – in the course of experimental studies it is difficult to register a tangible effect from changing the parameters of the vibrational system, so the introduction of this development will require additional costs for the experimental verification of the obtained results analytically.

Thus, SWOT analysis of the results of development allows to identify the main directions for the successful achievement of the aim. Among them:

– application of the obtained dependencies can simplify the conduct of experimental studies;
– analytical modeling of grinding processing allows to expand possibilities of increasing the process efficiency.

8. Conclusions

1. Analytical dependences of the grinding system motion with longitudinal feed in the form of differential equations that take into account the dimensions of the «drive – machine – grinding tool – workpiece» system are obtained and determine their interdependence. This allows to conduct theoretical studies of processes to establish appropriate grinding modes.

2. Using Nielsen algorithm, the dependencies of the motion of the vibrational system without taking into account the roughness of the profile of the grinding wheel are obtained, which allow to predict the position of the grinding tool during processing at any time. This makes it possible to study in detail the conditions of the vibrational grinding system under various conditions and to develop measures to improve the efficiency of the grinding process by selecting the correct grinding conditions (feed rate, grinding wheel rotation velocity, grinding depth, etc.).

References

1. Reznikov, A. N. Abrazivnaia i almaznaia obrabotka materialov [Text]: Handbook / ed. by A. N. Reznikov. – Moscow: Mashinostroenie, 1977. – 391 p.
2. Baikalov, A. K. Vvedenie v teoriiu shlifovaniia materialov [Text] / A. K. Baikalov. – Kyiv: Naukova dumka, 1978. – 207 p.
3. Burdun, G. D. Metody i sredstva kontrolia kachestva almaznogo instrumenta [Text] / G. D. Burdun, V. F. Surogin, V. G. Darevskii. – Moscow: Mashinostroenie, 1979. – 119 p.
4. Veits, V. L. Vynuzhdennye kolebaniia v metallorezhushchih stankah [Text] / V. L. Veits, V. K. Donshanskii, V. I. Chiriaev. – Leningrad: Mashgiz, 1959. – 287 p.
5. Gusev, B. V. Udarno-vibratsionnaia tehnologiia uplotneniia betonnyh smesei [Text] / B. V. Gusev, A. D. Deminov et al. – Moscow: Stroiizdat, 1982. – 152 p.
6. Srinivasan, K. Grinding Chatter and Vibrations [Text] / K. Srinivasan // Handbook of Modern Grinding Technology. – 1986. – P. 119–169. doi:10.1007/978-1-4613-1965-8_6
7. Hashimoto, F. Growing Mechanism of Chatter Vibrations in Grinding Processes and Chatter Stabilization Index of Grinding Wheel [Text] / F. Hashimoto, A. Kanai, M. Miyashita, K. Okamura // CIRP Annals. – 1984. – Vol. 33, No. 1. – P. 259–263. doi:10.1016/s0007-8506(07)61421-8
8. Han, Q. K. Nonlinearity Analysis and Wavelet Package Transform of Measured Chatter Vibrations in Grinding Process [Text] / Q. K. Han, B. C. Wen // Key Engineering Materials. – 2008. – Vol. 359-360. – P. 494–498. doi:10.4028/www.scientific.net/kem.359-360.494
9. Altintas, Y. Manufacturing Automation: Metal Cutting Mechanics, Machine Tool Vibrations, and CNC Design [Text] / Y. Altintas. – Cambridge: Cambridge University Press, 2012. – 336 p. doi:10.1017/cbo9780511843723
10. An experimental study on chatter vibrations in grinding operations [Text] // Wear. – 1958. – Vol. 2, No. 2. – P. 161. doi:10.1016/0043-1648(58)90496-4
11. Dudás, I. Grinding wheel profiling devices [Text] / I. Dudás // The Theory and Practice of Worm Gear Drives. – 2004. – P. 182–199. doi:10.1016/b978-190399661-4/50008-3
12. Ohmori, H. Highly Efficient Grinding of Ceramic Parts by Electrolytic In-Process Dressing (ELID) Grinding [Text] / H. Ohmori, I. Takahashi, B. P. Bandyopadhyay // Materials and Manufacturing Processes. – 1996. – Vol. 11, No. 1. – P. 31–44. doi:10.1080/10426919608947459
13. Veits, V. L. Nekotorye voprosy dinamiki mashin s elektroprivodom [Text] / V. L. Veits, V. L. Dobroslavskii // Trudy semira TMM. – 1962. – Vol. XXIII, No. 91. – P. 54–66.
14. Kazovskii, E. Ya. Perehodnye protsessy v elektricheskih mashinah peremennogo toka [Text] / E. Ya. Kazovskii. – Moscow; Leningrad: AN SSSR, 1962. – 624 p.
15. Poturaev, V. N. Dinamika vertikal'nih vibrokonveierov s uchetom vliianiem massy transportiruemogo materiala i svoistv istochnika energii [Text] / V. N. Poturaev, A. G. Chervonenko // Mehanika mashin. – 1971. – Vol. 29-30. – P. 25–36.
16. Kononenko, V. O. Kolebatel'nye sistemy s ogranichennym vozbuzhdeniem [Text] / V. O. Kononenko. – Moscow: Nauka, 1964. – 256 p.
17. Panovko, Ya. G. Ustoichivost' i kolebaniia uprugih sistem [Text] / Ya. G. Panovko, I. I. Gubanova. – Moscow: Nauka, 1979. – 384 p.
18. Karpov, N. A. Ustoichivost' form statsionarnyh dvizhenii vibriruiushchih organov putevyh i stroitel'nyh mashin [Text] / N. A. Karpov. – Moscow: Transport, 1970. – 168 p.
19. Mazur, N. P. Osnovy teorii rezaniia materialov [Text]: Hanbook / N. P. Mazur, Yu. N. Vnukov, A. I. Grabchenko, V. L. Dobroskok, V. A. Zaloga, Yu. K. Novoselov, F. Ya. Yakubov; ed. by N. P. Mazur, A. I. Grabchenko. – Ed. 2. – Kharkiv: NTU «KhPI», 2013. – 534 p.

МАТЕМАТИЧЕСКОЕ МОДЕЛИРОВАНИЕ ВИБРАЦИОННЫХ СИСТЕМ ДЛЯ ПОПЕРЕЧНОГО ШЛИФОВАНИЯ ПЕРИФЕРИЕЙ КРУГА

Рассмотрена модель вибрационной системы шлифования, которая позволяет проследить основные закономерности движения, получить аналитические зависимости движения шлифовальных кругов, переход системы через промежуточные резонансы, пути преобразования и рассеивания энергии в системе. Данная модель создает предпосылки для повышения эффективности процесса шлифовальной обработки.

Ключевые слова: вибрационная система шлифования, закономерности движения, шлифовальный круг.

Hnitko Serhii, *PhD, Associate Professor, Department of Engineering Technology, Poltava National Technical Yuriy Kondratyuk University, Ukraine, e-mail: 06smg2016@gmail.com, ORCID: http://orcid.org/0000-0003-2732-661X*

Shpylka Andrii, *Senior Lecturer, Department of Engineering Technology, Poltava National Technical Yuriy Kondratyuk University, Ukraine, e-mail: anshpilka@gmail.com, ORCID: https://orcid.org/0000-0002-6282-1336*

Shpilka Nikolay, *PhD, Associate Professor, Department of Construction Machine and Equipment, Poltava National Technical Yuriy Kondratyuk University, Ukraine, e-mail: nikolajspilka@gmail.com, ORCID: https://orcid.org/0000-0002-5016-018X*

Kravchenko Serhii, *PhD, Associate Professor, Department of Engineering Technology, Poltava National Technical Yuriy Kondratyuk University, Ukraine, e-mail: 050ser09@i.ua, ORCID: http://orcid.org/0000-0003-1453-4444*