ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
Vol. 11, Issue 4, April 2024

Amplitude-Frequency Characteristics of the Damper Rod of Observational Design Parameters

Farkhod Matqurbonovich Matmurodov, Ruziev Shokhrukh Shoim ugli

PhD, Professor of the Department of General Technical Disciplines, Chirchik High Tank Command Engineering School, Chirchik Uzbekistan;
Senior Lecturer of the Department of General Technical Disciplines, Chirchik High Tank Command Engineering School, Chirchik, Uzbekistan

ABSTRACT: The article analytically describes the amplitude-frequency characteristics of the action of machine mechanisms, taking into account kinematic characteristics. For the first time, the amplitude-frequency characteristics of the drive shaft quenching process were analytically determined, taking into account the load-design parameters. As special cases, the amplitude-frequency characteristics of the driveshaft are studied, taking into account the design parameters during the damping process and without damping

KEYWORDS: amplitude-frequency response; vibration; damping; drive and driveshaft; kinematic parameter; load-constructive parameter, disturbing effect.

I. INTRODUCTION

There are hygienic, technical and diagnostic vibration standards. *Hygienic* regulation consists of *limiting* the parameters of vibration affecting a person. During *technical standardization*, design vibration parameters are established when designing equipment, and the influence of vibration on the technological process is taken into account. During *diagnostic standardization*, zones of the technical condition of equipment are established: A - when the equipment is accepted for operation, B - during operation without a time limit, C - during operation until the next preventive measures. Zone D corresponds to the emergency condition of the equipment [1].

For inter-industry equipment, diagnostic standards are established by international or national standards, and in some cases – by industry standards.

Vibration, according to the way it affects a person, is divided into general and local. General vibration is transmitted to the body of a person sitting or standing on a vibrating surface of a machine, foundation, or structure through the supporting parts of his body.

According to the source of occurrence, general vibration is divided into three categories [2-3]:

1) transport vibration affecting operators of mobile machines and vehicles when they move on roads; at timber industry enterprises, such vibration is created, for example, by timber trucks, boats, and in-plant transport;

2) transport and technological vibration affecting machine operators with limited movement only on specially prepared surfaces of production premises and industrial sites; it is created by cranes, feller skidders, forklifts, and floor-mounted production vehicles;

3) technological vibration affecting operators of stationary machines or transmitted to workplaces that do not have sources of vibration; occurs during the operation of woodworking and paper-making equipment, electrical machines, pumps and fans.

Local vibration is transmitted to a person through the hands. It occurs when using hand-held machines: on handles, levers and other controls of machines and units; when a worker's hands come into contact with vibrating equipment. Vibration affecting a person through his body parts in a wide range of frequencies excites vibrations of his body and internal organs. In this case, resonant oscillations are possible. The frequencies of natural vibrations of internal organs range from 3 to 6 Hz, the frequencies of natural vibrations of the shoulder girdle, hips, head in the "standing" position are 4…6 Hz, the head relative to the shoulders in the "sitting" position is 25…30 Hz. External vibrations with a frequency of less than 0.7 Hz form a pitching motion that disrupts the normal activity of the vestibular apparatus in a person.

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
Vol. 11, Issue 4, April 2024

Low-frequency vibrations and infrasonic waves (with a frequency of less than 16 Hz) depress the nervous system and cause a feeling of anxiety. Oscillations with a frequency of 3...7 Hz are especially dangerous, causing resonance of internal organs. Long-term exposure to general and local vibration in a wide range of frequencies leads to vibration disease.

From the above opinions it is clear that vibrations above the limit greatly affect human health (operator and passenger of mobile and technological machines) and the reliability of ground equipment. Until now, the amplitude-frequency characteristics of the drive shaft damping process have been mainly described mathematically only taking into account kinematic parameters. In this work, the analytical determination of the amplitude-frequency characteristics of the damping process of the drive shaft takes into account the load (force, moment) and design (geometric dimensions of the mechanisms) parameters. And so an analytical solution is considered one of the ways to solve this vibration problem.

II.RESEARCH METHODS

Theory of stability of steady motions of mechanical systems; Routh's method for differential equations of mechanical motion of machine mechanisms, the laws of theoretical mechanics with the subsequent identification of periodic motion, which automatically collects stable solutions and allows you to find all the necessary characteristics.

III. RESULTS AND DISCUSSION

An increase in machine productivity leads to an increase in dynamic loads, an increase in dynamic errors in the laws of motion of working bodies, and thus necessitates a comprehensive dynamic analysis at the design stage [4]. The causes of dynamic loads may be uneven rotation of the drive motor, non-synchronization and compliance of the transmission, variable workloads, and dynamic effects of the control system.

The drive with cardan transmission is asynchronous. The transmission causes kinematic disturbance in the machine unit [5]. In a system with a cardan transmission, parametric and forced oscillations occur. The problem of controlling vibrations in oscillatory mechanical systems of a rotational type can be solved by introducing additional connections into the original oscillatory system.

Additional connections can be implemented using various methods. One of the possible solutions is a method of dynamic damping of rotational vibrations, based on the introduction of additional second-order connections interacting with the field of inertial forces.

Amplitude-frequency characteristics of the system taking into account kinematic parameters [6]

$$A(\omega_s) = [\omega_{12}^2 + k_1\omega_d^2 - k_2\omega^2]/[\omega_{12}^2 + k_1\omega_d^2 - \omega^2] \quad (1)$$

Amplitude-frequency characteristics of the absorber taking into account kinematic parameters (Fig.1)

$$A(\omega_d) = [\omega_{12}^2 + k_1\omega_d^2 - k_2\nu^2\omega^2]/[\omega_{12}^2 + k_1\omega_d^2 - \nu^2\omega^2] \quad (2)$$

$\omega_{\text{В}} = \nu\omega_d$ stimulating influence.

The harmonic order of the extinguisher $\nu = \omega_{\text{В}}/\omega_d$.

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
Vol. 11, Issue 4, April 2024

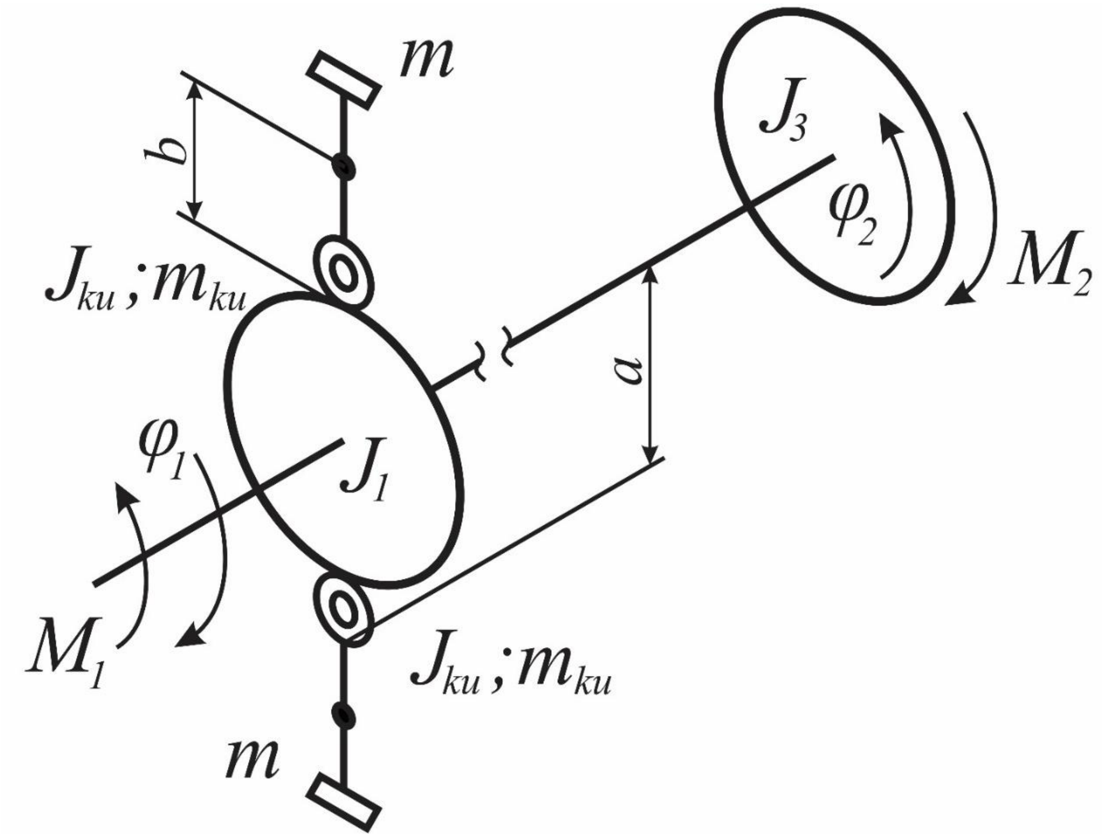

Fig.1

To study the influence of the design parameters of the damper on its dynamic properties, we write the expressions in expanded form

$$A(\omega_d) = \begin{Bmatrix} C_{20+nm_d abi^2\omega_d^2 cos\alpha_d -} \\ -m_{ku}[(J_d + m_d b^2)(i+1) + m_d b^2(i-1) - m_d abcos\alpha_d]v^2\omega_d^2 \end{Bmatrix} /$$

$$/\begin{Bmatrix} C_{20+nm_d abi^2\omega_d^2 cos\alpha_d -} \\ -m_{ku}[(J_d + m_d b^2)(i+1)^2 + m_d(b^2(i-1) - 2ab(i-1)cos\alpha_d)]v^2\omega_d^2 + J_3 v^2\omega_d^2 \end{Bmatrix} \quad (3)$$

$$v_d = \sqrt{\frac{nm_d abi^2 cos\alpha_d}{m_{ku}[(J_d+m_d b^2)(i+1)+m_d b^2(i-1)-m_d abcos\alpha_d]}}, \quad (4)$$

$$\alpha_d = \frac{nJ_2 i(i+1)}{J_3+nm_{2ku}a^2+nJ_2(i+1)},$$

$$-\omega_3 J_3 A_1 + \left(-\frac{\omega_3^2 J_3}{k_{21}} + 1\right) M_1 - \mathrm{M}_3 = 0,$$

$$\omega_3 J_3 A_1 = \left(-\frac{\omega_3^2 J_3}{k_{21}} + 1\right) M_1 - \mathrm{M}_3.$$

.

Amplitude-frequency characteristics of the cardan shaft taking into account design parameters without damping

$$A_1 = \left[\left(1 - \frac{\omega_3^2 J_3}{k_{21}}\right) M_1 - \mathrm{M}_3\right] / \omega_3 J_3, \quad (5)$$

Let's use this equation and determine the amplitude-frequency characteristics of the cardan shaft, taking into account design parameters with damping

$$A_{1d} = \left[\left(1 - \frac{(v_{2ku}/r)^2 J_{2ku}}{k_{2ku}}\right) M_1 - M_2^d\right] / (v_{2ku}/r) J_{2ku}, \quad (6)$$

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
Vol. 11, Issue 4, April 2024

$$k_{2ku} = \frac{m_2[(J_d+m_d b^2)(i+1)+m_d b^2(i-1)-m_d ab cos\alpha_d]}{n\{(J_d+m_d b^2)(i+1)^2+m_d[a^2+b^2(i-1)^2-2ab(i-1)cos\alpha_d]\}+J_3}. \quad (7)$$

Amplitude-frequency characteristics of the damper for rod taking into account design parameters

$$A_2 = A_{1d} + \frac{1}{k_{2ku}} M_1, \quad (8)$$

here A_{1d} , A_2– system oscillation amplitudes; ω_1, ω_d – stimulating and damper frequency; M_1 – reactive torque amplitude; $M_2^d = -\omega_d J_2 A_2$ – amplitude of dynamic damping acting on a mass with a reduced moment of inertia J_2 ; $\omega_1=\pi n/30$ – angular frequency of shaft vibration; n – shaft vibration frequency per minute; ν– order of the harmonic of the disturbing influence (the number of harmonic periods that fits in one revolution of the shaft).

Analysis of the results shows that the greatest influence on the dynamic properties of the damper is exerted by the ratio of design parameters a and b. Changing their ratio $k_d = a / b$ within $3{,}75 \geq k_d \geq 0{,}94$ allows us to provide a dynamic damping mode in the range of changes ω_2 from 80 sec $^{-1}$ до 500 сек $^{-1}$ and a self-tuning mode in invariant systems at 1, 2, 3 harmonics of the carrier circular frequency.

Shaft torque without damping

$$M_1 = \frac{\omega_1^2 J_1}{-\frac{\omega_1^2 J_1}{k_1}+1}, \quad (9)$$

Shaft torque with damping

$$M_{1d} = \frac{-\omega_{2ku}^2 J_2+\omega_1^2 J_1}{-\frac{\omega_1^2 J_1}{k_1}+1} = \frac{-(v_{2ku}/r)^2 J_2+\omega_1^2 J_1}{-\frac{\omega_1^2 J_1}{k_1}+1}, \quad (10)$$

A point located at a distance R from the axis passes the path $\Delta s = R\Delta\phi$.

The linear speed of the point is equal to $\upsilon = \lim \frac{\Delta s}{\Delta t} = \lim \frac{R\Delta\phi}{\Delta t} = R\frac{d\phi}{dt} = R\omega$,

for rod $v_{2ku} = r\omega_{2ku}$ $\omega_{2ku} = v_{2ku}/r_{2ku}$

here v_{2ku}, r_{2ku} - linear speed and curac radius.

$$2k_{21} = \frac{b_{21}}{nJ_3(i+1)+nm_3a^2+J_3}$$

$$\omega_3 = \sqrt{\frac{c_{12}}{nJ_3(i+1)+nm_3a^2+J_3}},\ v_{2ku}/r_{2ku} = \sqrt{\frac{c_{12}}{nJ_3(i+1)+nm_3a^2+J_3}} \quad (11)$$

Using the above equation, we describe

$$v_{2ku} = r_{2ku}\sqrt{\frac{c_{d2}}{nJ_{2ku}(i+1)+nm_{2ku}a^2+J_d}} \quad (12)$$

IV.CONCLUSION

The amplitude-frequency characteristics of the operation of machine mechanisms are analytically described, taking into account kinematic characteristics. For the first time, the amplitude-frequency characteristics of the vibration damping process of the drive shaft were analytically determined, and the load-design parameters were taken into account. Taking into account the load and design parameters when damping vibration will help more accurately carry out the design and technological design work of new mobile and technological machines being developed.

REFERENCES

1.Kutsubina, N.V., Sannikov, A.A. Theory of vibration protection and acoustic dynamics of machines: textbook. –Ekaterinburg: Ural State Forestry University, 2014. - 167 p.

2.GOST 12.1.012-90. Vibration safety. General safety requirements. –M.: Standards Publishing House, 1990. – 23 p.

3. Sannikov A.A. and others. Vibration and noise of technological machines and equipment in the forestry industry: monograph. –Ekaterinburg: Ural State Forestry University, 2006. – 484 p.

4. Kolovsky M.Z. Machine dynamics. M.: Mechanical Engineering, Leningrad Branch, 1989. 263 p.

5.Grudinin V.G. Study of the influence of additional connections in oscillatory mechanical systems of rotational type // Bulletin of Irkutsk State Technical University. 2011. No. 2. pp. 34–40.

6.Grudinin V.G. Dynamic analysis of angular vibration damper. Bulletin of Irkutsk State Technical University No. 4 (63) 2012, pp. 34-37.