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REGULIARITIES IN THE OCCURRENCE AND PROPAGATION OF WAVE PROCESSES IN THE KINEMATIC CHAINS OF A MOBILE ROBOTIC MACHINE TOOL

BY

VASILIY STRUTYNSKIY[1], YURI BURENNIKOV[2] and LEONID KOZLOV[2,*]

[1]National Technical University of Ukraine “Igor Sykorskiy Kyiv Polytechnic Institute”
[2]Vinnytsia National Technical University

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Abstract. The paper considers regularities in the occurrence and propagation of wave processes in the kinematic chains of a mobile robotic machine tool. Mobile robotic machine tools are designed for handling dangerous objects under field conditions. Such device comprises a movable actuating member with a tool that moves in space by means of six bars of a variable length. A spatial rod structure is prone to the occurance of vibration processes, which is an undesirable phenomenon. For description of vibration processes a dynamic model in the form of separate masses with elastic-dissipative bonds has been developed. Longitudinal and transverse waves, caused by longitudinal, torsional and bending vibrations, have been described. It is pointed out that propagation of the waves in a rod system is accompanied by their breaking and interference. To reduce the intensity of wave phenomena, installation of special damping connections in the rod system is recommended as well as the use of dampers in the places, where crests of rod vibrations are observed.

Keywords: carrying system; dynamic model; elastic-dissipative bonds; interference of the waves; damping of vibrations.

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[*]Corresponding author; *e-mail*: osna2030@gmail.com

1. Introduction

Mobile robotic machine tools with parallel kinematics are designed to handle dangerous objects under field conditions. Therefore, mobile robotic machine tools have minimal mass and are made in the form of spatial rod structures, which include parallel kinematic chains. The spatial rod structure is prone to complex dynamic vibration processes of a wide frequency range. Vibration processes in mobile robotic machine tools are undesirable phenomena that significantly reduce their accuracy. Therefore, investigation of vibration processes in rod systems of mobile robotic machine tools is a relevant task.

Dynamics of multicoordinate mobile robotic machine tools with spatial rod structures has certain special features. For present, dynamic phenomena in robotic machine tools with parallel rod structures have not been adequately investigated. The reason is complexity of the processes and absence of the theoretical models of these processes. Therefore, the problem in a general form consists in determining special features of the dynamic processes occurring in mobile robotic machine tools.

The problem involves important scientific and practical tasks of creating mobile robotic machine tools intended for handling dangerous objects under field conditions.

Recent research and publications give considerable attention to investigation of the technological equipment based on spatial mechanisms (Zhang and Dai, 2015). The most common are robotic systems based on a hexapod mechanism (Ritzen *et al*., 2016). Results of studying various manipulator types are available (Jiang and Cripps, 2015). Some research works deal with mobile machine tools and robots based on the mechanisms of a parallel structure (Zhao *et al*., 2015; Hu *et al*., 2016). The research is related to the general problems of kinematics and dynamics of robots (Alghooneh *et al*., 2016; Briot and Khalil, 2015). Special features of dynamic processes occurring in machine tools with parallel kinematic structures have been investigated (Strutynskyi *et al*., 2015). It has been determined that the dynamic processes are distinguished by significant complexity. The reason is special elastic properties of the elastic rod systems of mobile robotic machine tools (Strutynskyi and Hurzhii, 2017).

All of the known information sources present results of the research on vibrational processes corresponding to polyharmonic functions. The available literature contains no results of the research on wave processes. Investigation of the regularities in the wave process onset in mobile robotic machine tools has been found to be an unsolved aspect of the general problem.

Respectively, the research aims at determining regularities in the occurrence and realization of wave vibrational processes in the rod system of a mobile robotic machine tool. The research tasks are mathematical description of the wave source and determination of the regularities of wave phenomena in a machine tool carrying system.

2. Presentation of the Main Research Material

A mobile robotic machine tool with kinematic links includes a movable actuating member 1 with tool 2 that moves in space by means of six bars of a variable length (Fig. 1).

Actuators 4 of the bars are installed on frame 5 that is hinged to bars 6 located at the mobile vehicle. System 7 is designed for moving the frame by its pivoting on hinges 8.

The mobile robotic machine provides the tool motion according to the coordinates through the change of the rod structure configuration.

Dynamics of multicoordinate robotic machine tools with parallel rod structures has special features connected with the occurrence of wave phenomena. These phenomena emerge in the process of a machine tool operation as well as during investigation of its dynamic characteristics. For present, wave phenomena in robotic machine tools with rod structures have not been adequately investigated.

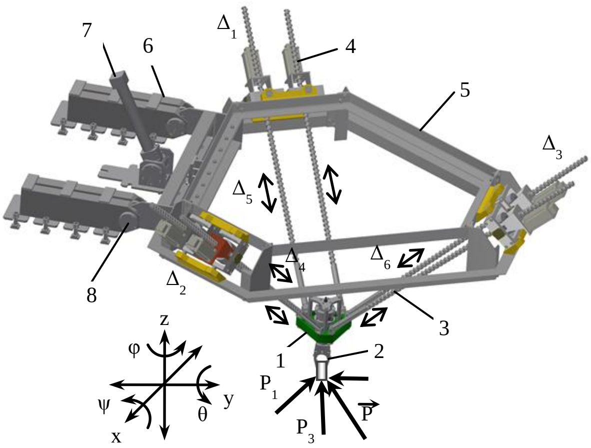

Fig. 1 – Rod system of the mobile robotic machine drive having six bars of a variable length.

The reason is complexity of the processes and absence of the models of the processes. A new theoretical model of the machine tool dynamic rod system is proposed, which makes it possible to take into account regularities in the emergence and propagation of wave processes in kinematic chains and in the machine tool carrying rod system. The wave model of the machine tool dynamic system is presented in the form of a set of individual masses with elastic-dissipative bonds. The number of masses, their values and locations as well as the values of elastic and dissipative coefficients of the bonds between individual masses are described in terms of fuzzy sets. The wave model of the machine tool enables numerical and qualitative description of the wave processes occurring in the machine tool dynamic system. The main source of dynamic wave processes in machine tools are cutting forces. Therefore, the tool is assumed to be the source of waves *D* (Fig. 2).

For studying wave processes, the wave source is assumed to have a shock nature and a certain direction. Therefore, the source intensity (disturbance) is described by a spatial pulse (shock) process. For mathematical description of such process delta function of a certain direction is used. Accordingly, the dynamic disturbance force is described by the vector

$$\vec{D} = \vec{e}_D \cdot \delta(t),$$

where $\vec{e}_D$ is a unit vector that describes direction of the dynamic disturbance action.

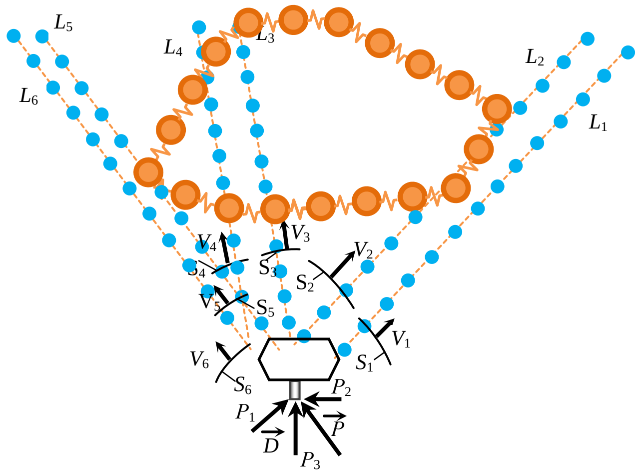

Fig. 2 – Representation of the machine tool dynamic system in the form of a fuzzy set of masses, linked by elastic-dissipative bonds.

This description determines non-stationary dynamic disturbance that corresponds to the shock load acting on the tool.

Disturbances that emerge at the cutting part of the tool spread gradually to the kinematic chains, capturing portions of the bar and of the machine tool carrying system.

In each bar ($L_1 – L_6$) disturbances will spread with the speeds $V_1 – V_6$. Taking into account that the bar length is much larger than its cross-sectional size, it could be assumed that a flat wave propagates in each bar. Dynamic processes occurring in the bars have the form of elastic waves that propagate in the elastic media of the bar.

Waves propagating in separate bars have flat fronts $S_1 – S_6$. Union of the flat fronts forms a general front of the wave, which is an analog of the wave surface or a spatial front of the wave. The bars are connected to the machine-tool platform at separate points of the platform. Therefore, dynamic disturbance, acting on the tool, causes different disturbances at each of the bars (Fig. 3).

When dynamic disturbance acts on the platform, the total wave that propagates from the tool to the platform breaks into six waves emerging in the lower hinges of the bars. Breaking of the waves is characterized by Botha change of the wave intensity and phase. Dynamic disturbances in the hinges of the bars spread along the wave paths $T_1 – T_6$ that include the bars, the hinges and the machine tool drives.

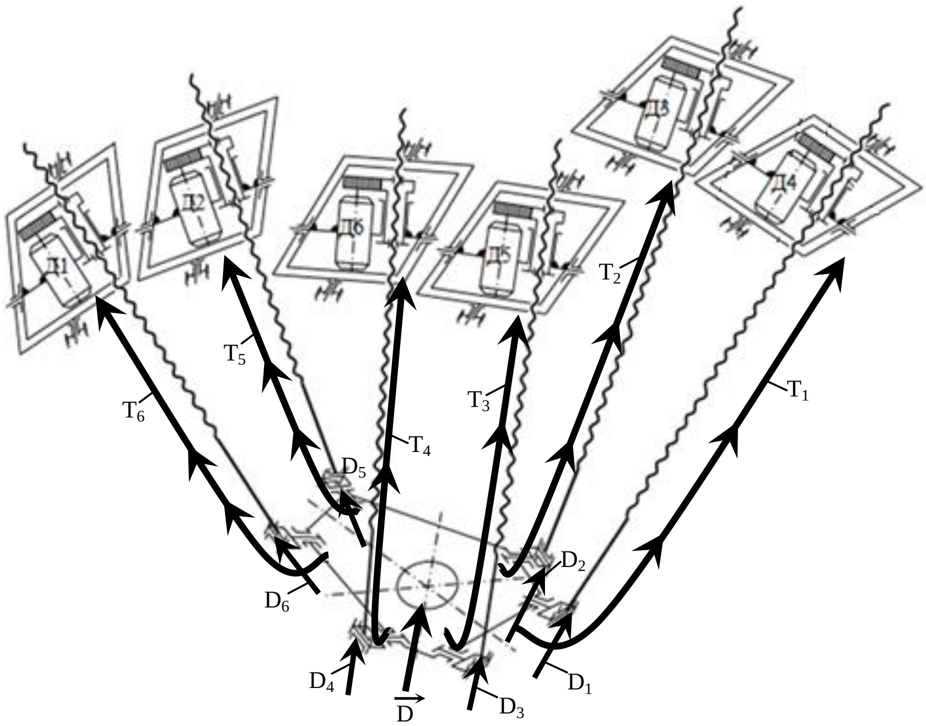

Fig. 3 – The scheme of breaking the vibrational wave disturbance, acting on the tool of the machine with parallel kinematics, into separate dynamic disturbances in separate bars.

Both longitudinal and transverse waves propagate in the bars and carrying system of the machine. A longitudinal wave is characterized by vibration x_K of the bar cross-section y_K, z_K in the direction of the wave (Fig. 4).

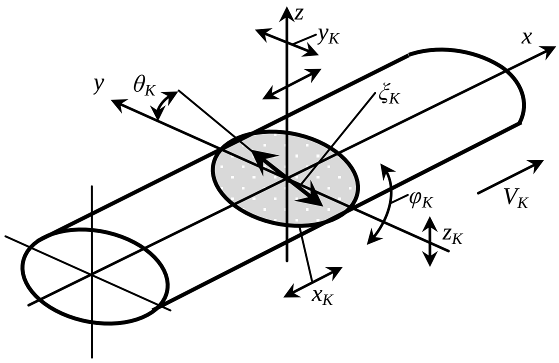

Fig. 4 – Vibrations of the bar cross-section, which correspond to longitudinal x_K, transverse y_K, z_K, ξ_K and torsional φ_K waves, propagating with speed V.

A transverse wave correspond to the flexural vibrations of the bar. Transverse waves occur when the cross-sections move in different directions.

Vibrations of the cross-sections y_K, z_K in the directions of the coordinate axes x, y are characterized by the cross-section displacement in the arbitrary direction ξ_K. Taking into account connection between the polar and Cartesian rectangular coordinate systems, we obtain relationships between the displacements:

$$y_K = \xi_K \cos\theta, \; z_K = \xi_K \sin\theta.$$

Accordingly, $\xi_K = \sqrt{y_K^2 + z_K^2}$, $\theta_K = \operatorname{arctg}\dfrac{z_K}{y_K}$.

As for the bar, which is an elastic rod, pivoting displacement of its cross-section φ_K occurs, which corresponds to a torsional wave. Waves of this type correspond to torsional vibrations of the bar as a system with distributed parameters.

The wave propagation velocity is equal to the distance covered by a wave surface point in a unit time. Propagation velocity is different for different types of the waves (longitudinal, transverse, torsional). Therefore, waves of different types propagate in the bar at different velocities. The velocity of a longitudinal wave propagation in an elastic rod is defined as:

$$V_K = \sqrt{\frac{E}{\rho}},$$

where E – elasticity modulus of the rod material; ρ – density of the material.

For a steel rod, the velocity of a longitudinal wave propagation is of the order $V \cong 5000$ m/s. The time of the longitudinal wave propagation in a 2 meter long bar is estimated as $\Delta t_x \cong 0.0004$ c = 0.4 ms.

The velocity of a torsional wave propagation in an elastic rod is given by

$$V_K = \sqrt{\frac{G}{\rho}},$$

where G – shearing elasticity modulus.

Propagation of torsional waves in a steel rod is approximately $V \cong 3000$ m/s. The estimated time of a torsional wave passing through a bar of 2 m length is

$$\Delta t_\varphi \cong 7.0 \text{ ms}.$$

The velocity of a torsional wave propagation in a bar in the form of an elastic rod depends on the shape and size of the cross-section and the bar support conditions. For a bar with hinges at the ends, propagation velocity of the wave of transverse vibrations is determined by the formula:

$$V = \sqrt{\frac{EI_\tau}{\rho F}} \cdot \frac{i\pi^2}{L},$$

where I_τ – inertia moment of the rod bending in the plane that corresponds to the direction ξ_K; F – cross-section area of the bar; L – the bar length; i – the vibration shape number

In the first approximation, we assume that the bar is a round solid rod of a diameter d. In this case:

$$I_\tau = \frac{\pi d^4}{64},\ F = \frac{\pi d^2}{4}.$$

Accordingly, the transverse wave propagation velocity will be given by

$$V = \sqrt{\frac{E}{\rho}} \frac{\pi^2 id}{L\sqrt{8}}.$$

For a typical bar (d=25 mm, L=2 m) and the fifth shape of vibrations the velocity is of the order $V \cong 1250$ m/s.

Respectively, the estimated time of a transverse wave passing through a bar of 2 m length is

$$\Delta_{t\xi} \cong 17 \text{ ms}.$$

The length of sinusoidal wave λ is the smallest distance between the bar cross-sections, vibrations of which have the phase difference of 2π:

$$\lambda = \frac{2\pi}{K} = VT = \frac{V}{\nu}, \quad (1)$$

where $T = \frac{2\pi}{\omega}$ – the wave period; ω – circular velocity; v – cyclic velocity of the wave; $K = \frac{2\pi}{\lambda}$ – wave number. The wave number indicates the number of waves in 2π interval.

The law of vibration of an arbitrary bar cross-section at distance x of the wave source is described by the dependence

$$\xi = A\cos[\omega(t-\tau)], \tag{2}$$

where ξ – the bar cross-section deviation from the nominal position; A – the amplitude of cross-section deviation; τ – time required for disturbance from cross-section x=0 to reach cross-section x, namely, $\tau = x/V$. Combining (1) and (2), we obtain:

$$\xi = A\cos(\omega t - \frac{x\omega}{V}) = A\cos(\omega t - Kx)\,. \tag{3}$$

Dependence (3) describes a flat travelling wave in the bar. The phase of dependence (3) is the function of two variables t and x.

Let us consider a separate cross-section of the kinematic chain rod and carrying system of the machine tool where wave processes occur. This cross-section (x=const) is characterized by the presence of a harmonic vibration process described by the dependence:

$$f(t) = A\sin(\omega t + \psi_0)\,,$$

where A – the amplitude of wave process that describes vibrations of a separate cross-section; ψ_0 – the initial phase, determined for this cross-section, $\psi_0 = \frac{\pi}{2} - Kx_0$.

In the cross-sections of the rods complex various-scale wave processes occur. For a simplest case, the sum of wave processes is determined using the superposition principle.

Let us consider displacement of a single rod cross-section that is at a distance x_0 from the wave disturbance source. The displacements are described by the dependences:

$$y = A_y \cos\left(\omega t_x - \frac{x_0\omega}{V_1}\right), z = A_z \cos\left(\omega t_x - \frac{x_0\omega}{V_2} + \psi_{z0}\right). \tag{4}$$

We choose a time reference point t from the condition:

$$\omega t = \omega\left(t_x - \frac{x_0}{V_1}\right),\ t = t_x - \frac{x_0}{V_1}\,.$$

Accordingly, formulas (4) will take the form of

$$y = A_y \cos(\omega t_x)\,,\ z = A_z \cos(\omega t_x + \psi_0)\,, \tag{5}$$

where $\psi_0 = \psi_{zo} + \frac{x_0\omega}{V_1} - \frac{x_0\omega}{V_2}$.

From formula (5) it follows that in the presence of wave phenomena of this type cross-section of the rod will move along an elliptical trajectory. From the first equation (5) we determine that

$$\cos\omega t = \frac{y}{A_y}. \qquad \text{Accordingly,} \qquad \sin\omega t = \pm\sqrt{1-\left(\frac{y}{A_y}\right)^2}.$$

We can present the cosines of the sum of two angles in the second equation (5) in the form of a combination of trigonometric functions:

$$\cos(\omega t + \psi_0) = \cos(\omega t)\cdot\cos(\varphi_0) - \sin(\omega t)\sin(\varphi_0) = \frac{z}{A_z}.$$

Respectively, the equations of the resultant displacements of the rod cross-section will be given by

$$\frac{y}{A_y}\cos(\varphi_0) \mp \sqrt{1-\left(\frac{y}{A_y}\right)^2}\cdot\sin(\varphi_0) = \frac{z}{A_z}$$

These equations could be presented in the transformed form as

$$\left(\frac{y}{A_y}\right)^2 + \left(\frac{z}{A_z}\right)^2 - \frac{2yz}{A_yA_z}\cos(\varphi_0) = \sin^2\varphi_0$$
.

This equation corresponds to the ellipse, the axes of which are turned by an angle ψ_0 relative to the coordinate axes *y* and *z* .

For the case, when phase angle $\varphi_0=\pi m$ (m=0, ±1, ±2, …), the ellipse degenerates into a straight line:

$$z = \pm\frac{A_z}{A_y}y. \qquad (6)$$

The resultant vibrations of the cross-section are harmonic vibrations with the amplitude $\sqrt{A_y^2 + A_z^2}$.

The plus sign in formula (6) corresponds to the even values of *m*, *i.e*. to the addition of co-phased vibrations, and the minus sign – to the odd values of *m*, *i.e.* addition of the vibrations that occur in the counter-phase.

At the intersections of the rods, forming the machine tool dynamic system, amplification of the wave phenomena is observed, which is caused by interference of the waves. Interference involves waves of different nature – longitudinal, transverse and torsional.

When interference of two waves occur, the resultant vibration is a geometric sum of the vibrations of both waves in the corresponding sections of the rod. Under interference conditions, the superposition principle is, as a rule, accurate, and is broken only in separate cases, when the vibration amplitude is significant (nonlinear processes).

Let us consider two waves of the same frequency, which are observed in one section of the rod located at different distances x_1 and x_2 from the disturbance source. This is the case when wave paths have similar characteristics, but different lengths (Fig. 5).

In section Π interference of two waves is observed. These waves are described by sinusoidal processes y_1 and y_2 of the following form:

$$y_1 = A_1 \sin(\omega t - K_x x_1) = A_1 \sin\varphi_1,\ y_2 = A_2 \sin(\omega t - K_x x_2) = A_2 \sin\varphi_2 .$$

The resultant total wave is represented in the form of

$$y = y_1 + y_2 = A_1 \sin\varphi_1 + A_2 \sin\varphi_2 = (A_1 + A_2 \cos\psi)\sin\varphi_1 - A_2 \sin\psi \cos\varphi_1,$$

where phase difference of the two vibrations is

$$\psi = \varphi_1 - \varphi_2 = K_x (x_2 - x_1) = \frac{2\pi\Delta_x}{\Delta_x},$$

where $\Delta_x = x_2 - x_1$ – difference between the distances of two waves from the disturbance source.

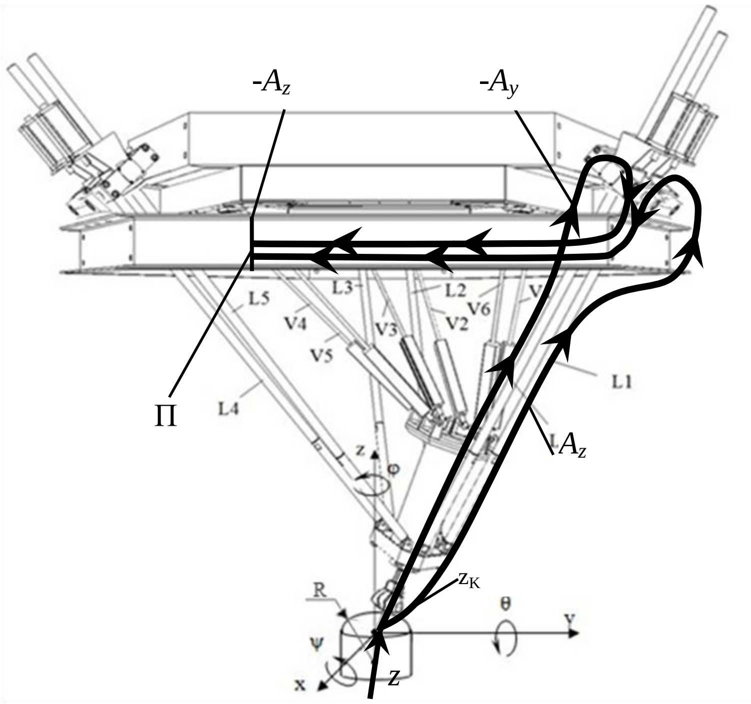

Fig. 5 – The scheme of wave paths of different lengths, which cause interference in the rod sections.

Let us introduce new parameters A, θ according to the depenences:

$$A_1 + A_2\cos\psi = A\cos\theta, \qquad A_2\sin\psi = A\sin\theta.$$

Then the squared amplitude of the total vibration will be given by

$$A^2 = A_1^2 + A_2^2 + 2A_1A_2\cos\psi.$$

Parameters θ and ψ are related as

$$tg\,\theta = \frac{A_2\sin\psi}{A_1 + A_2\cos\psi}.$$

Accordingly, the total wave process will be described by the following equation:

$$y(x_1,t) = A\cos\theta\sin\varphi_1 - A\sin\theta\cos\varphi_1 = A\sin(\varphi_1-\theta) = A\sin(\omega t - k_x x_1 - \theta).$$

Let us consider two separate cases. The first case corresponds to the maximum squared amplitude. For this case

$$\psi = \frac{2\pi\Delta_x}{\lambda_x} = 0, \pm 2\pi, \pm 4\pi, ..., \pm N_x 2\pi.$$

Respectively, $\Delta_x = 0, \pm\lambda_x, \pm 2\lambda_x, ..., \pm N_x\lambda_x$, where N_x – integer positive or negative number. Maximum value of the squared modulus of the amplitude will be given by:

$$(A^2)_{\max} = (A_1 + A_2)^2.$$

In the other case we have the minimum value of the squared amplitude:

$$(A^2)_{\min} = (A_1 - A_2)^2.$$

The values of the phase difference and the waves travel difference will be given by:

$$\psi = \frac{2\pi\Delta_x}{\lambda_x} = 0, \pm\pi, \pm 3\pi, ..., \pm(2N_x+1)\pi,$$

$$\Delta_x = 0, \pm\lambda_x/2, \pm 2\lambda_x/2, ..., \pm(N_x+1)\lambda_x/2.$$

If vibration amplitudes are equal, A_1=A_2, the total vibration amplitude will be:

$$A^2 = 2A_1^2(1+\cos\theta) = 4A_1^2\cos^2\theta/2 = 4A_1^2\cos^2\left(\frac{\pi\Delta_x}{\lambda_x}\right).$$

Maximum value of the squared amplitude $(A^2)_{\max} = 4A_1^2$ and minimum value – $(A^2)_{\min} = 0$.

Let us consider interference of two waves in the rod section Π, which is at the same distance from the disturbance source, though waves in this section propagate in opposite directions (Fig. 6). This process is characteristic of annular wave paths.

In the machine tool rod system there are closed annular wave paths. The annular path includes two symmetrical parts of the length *x*. The waves propagate symmetrically from the disturbance source along approximately identical wave paths. In the rod with section Π standing waves are observed.

Analytical description of the two waves in cross-section Π has the following form:

$$\begin{Bmatrix} \xi_1 = A\cos(\omega t - Kx) \\ \xi_2 = A\cos(\omega t + Kx) \end{Bmatrix}. \tag{7}$$

We will use the following known trigonometric dependence of the cosine sum of two angles α and β:

$$\cos\alpha + \cos\beta = 2\cos\left(\frac{\alpha+\beta}{2}\right)\cos\left(\frac{\alpha-\beta}{2}\right). \tag{8}$$

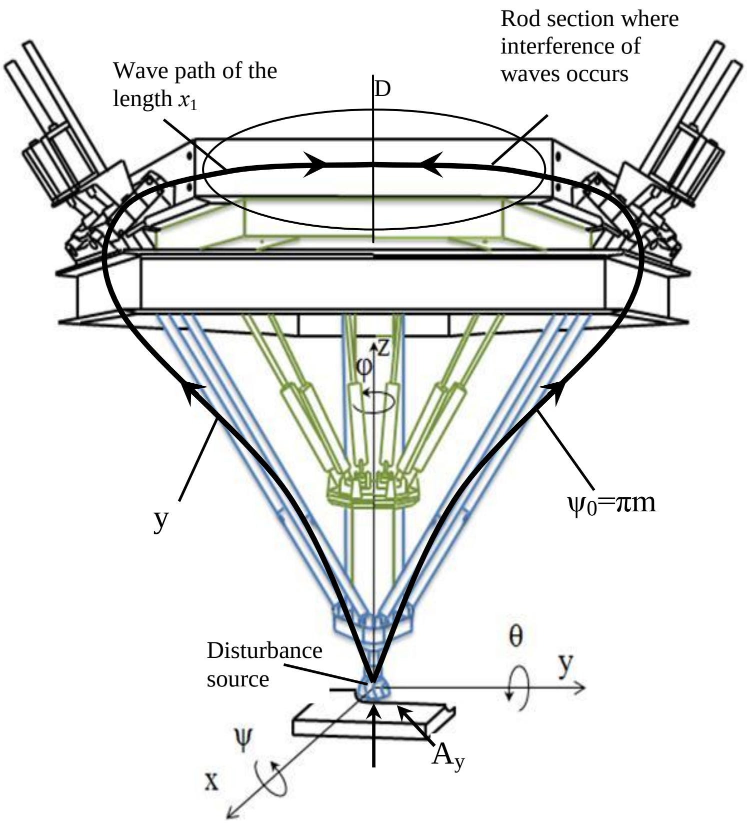

Fig. 6 – Propagation of the waves in two symmetrical dendritic wave paths with the formation of standing waves in the rod of the machine tool carrying system.

After adding Eq. (7) and performing transformation according to the cosine sum formula (8), we obtain:

$$\xi = \xi_1 + \xi_2 = 2A\cos\left(\frac{\omega t - Kx + \omega t + Kx}{2}\right)\cos\left(\frac{\omega t - Kx - \omega t - Kx}{2}\right).$$

Taking into account parity of the cosine function $\cos(-\alpha) = \cos\alpha$, we will obtain $\xi = 2A\cos(\omega t)\cos(Kx)$.

In accordance with the wave number definition $K = \frac{2\pi}{\lambda}$, the standing wave equation will be obtained in the form of

$$\xi = 2A\cos\left(\frac{2\pi}{\lambda}x\right)\cos(\omega t). \tag{9}$$

The standing wave amplitude is found from formula (9):

$$A_s = 2A\cos\left(\frac{2\pi}{\lambda}x\right).$$

At the points of the carrying system, where the coordinates satisfy the condition

$$\frac{2\pi}{\lambda}x = \pm n\pi \quad (n=1, 2, 3,\ldots);\ \cos\left(\frac{2\pi}{\lambda}x\right) = 1,$$

the total amplitude is equal to the maximum value $A_s = 2A$. These points are the standing wave crests. Coordinates of the crests are determined by the following dependence:

$$x_n = \pm n\frac{\lambda}{2}. \tag{10}$$

At the points, the coordinates of which satisfy the condition

$$\frac{2\pi}{\lambda}x = \pm(2n+1)\frac{\pi}{2}\ (n = 0,1,2,3,\ldots);\ \cos\left(\frac{2\pi}{\lambda}x\right) = 0$$

and the total amplitude is zero, the standing wave nodes are observed. Coordinates of the nodes are found by the formula:

$$x_n = \pm(2n+1)\frac{\lambda}{4}.$$

No vibrations are observed in the rod sections, located in the nodes, and they are fixed.

The coordinates of the crests, determined by formula (10), serve as a basis for development of recommendations on the reduction of wave phenomena in the rod system of a mobile robotic machine tool. It is recommended to install inertial vibration dampers in the places where crests are observed. One of the options of the efficient inertial damper is a device, having a massive ball installed in a spherical body filled with a viscous liquid.

To reduce total intensity of the wave phenomena, it is recommended to use damping connections in the rod system. They are implemented in the form of a package of metal or elastomeric plates and serve to reflect the waves and reduce their total intensity level.

3. Conclusions

1. In the rod systems of mobile robotic machine tools with parallel kinematics wave processes occur. Their source is a cutting process and a rational mathematical model of the disturbance is delta function acting in a certain direction.

2. To describe the wave process, it is recommended to use a dynamic model in the form of separate masses with elastic-dissipative bonds, located along the axes of the rods of kinematic chains and the machine tool carrying system.

3. In the rod system of a mobile robotic machine tool, longitudinal and transverse waves emerge. They are caused by longitudinal, torsional and bending vibrations of the rods. Average time, required for the waves of different types to pass through the bar of the machine tool kinematic chain, is 0.4; 7; 17 ms.

4. Propagation of waves in a mobile robotic machine tool is accompanied by breaking and interference of the waves. Besides, in the closed wave passages of the carrying system there are stable waves with nodes and crests, where minimum and maximum vibration amplitudes are observed.

5. To reduce intensity of the wave phenomena, installation of special damping connections in the rod system is recommended as well as the use of vibration dampers, which should be installed in the places where the crests of the rod vibrations are observed.

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REGULARITĂȚI ÎN APARIȚIA ȘI PROPAGAREA PROCESELOR DE UNDĂ ÎN LANȚURILE CINEMATICE ALE UNEI MAȘINI ROBOTICE MOBILE DE PRELUCRARE

(Rezumat)

Lucrarea analizează regularitățile apariției și propagării proceselor de undă în lanțurile cinematice ale unei mașină-unealtă mobile robotizate. Aceasta este destinată manipulării obiectelor periculoase în condiții grele de teren. Un astfel de dispozitiv cuprinde un element de acționare mobil, dotat cu o unealtă, care se deplasează în spațiu cu ajutorul a șase bare de lungime variabilă. O structură spațială a tijei este predispusă la apariția proceselor nedorite de vibrație. Pentru cazul considerat, în vederea descrierii proceselor de vibrație a fost dezvoltat un model dinamic cu mase separate între care există legături elastic disipative. Au fost analizate undele longitudinale și transversale cauzate de vibrațiile longitudinale, de torsiune și de încovoiere. Se subliniază faptul că propagarea undelor într-un sistem de tije este însoțită de fenomene de interferența și de distrugerea tijelor. Pentru a reduce intensitatea fenomenelor cauzate de vibrații, se recomandă instalarea unor conexiuni speciale de amortizare în sistemul de tije, precum și folosirea amortizoarelor în locurile în care se observă maxime de vibrații ale tijelor.