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Teoretična in eksperimentalna analiza dinamike mehanizmov Rolamite

A Theoretical and Experimental Investigation of the Dynamics of Rolamite-Type Mechanisms

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Prispevek najprej poroča o teoretični raziskavi dinamičnega modela valja in jermena pri mehanizmu tipa Rolamite (RTM). Dinamični model valja in jermena mehanizma RTM je opisan z diferencialnimi enačbami ter analiziran na dva načina, tj. v primeru, ko na sistem ne vpliva vibracijsko vzbujanje, in v primeru, ko le-to vpliva na sistem. Ugotovili smo, da je mogoče s parametri vibracijskega vzbujanja spreminjati parametre nezdrsnega območja. To nas je vodilo do predpostavke, da je v dejanskih mehanizmih tipa Rolamite mogoče vplivati na trenje med strukturnimi elementi mehanizma RTM.

Eksperimentalni del raziskave dinamike mehanizma tipa Rolamite z vibrajočimi elementi je bil izveden v treh smereh: kot analiza dinamičnih pojavov v mehanizmu valja in jermena, kot analiza postopkov obvladovanja trenja med elementi mehanizma RTM in kot analiza dinamičnih pojavov v vibramotorjih Rolamite. Povečanje amplitude električne napetosti, zmanjšanje sile obremenitve gibkega jermena, in zmanjšanje kota, pod katerim gibki jermen ovija valj, povzročijo povečanje amplitude vibracij valjev. Mehanizem RTM z vrtečima se in vibrajočima valjema ima v primerjavi z vibrajočima valjema drugačnega tipa boljšo ležajno zmogljivost in je precej bolj občutljiv. Največjo sinhronost vrtenja rotorjev zagotavlja vibramotor Rolamite, ki se sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki hkrati vrti oba rotorja. Ta vibramotor primerjamo z vibramotorjem Rolamite z dvema rotorjema in piezo električnim pretvornikom vibracij, ki vrti le enega od rotorjev.

(Ključne besede: Rolamite mehanizmi, zdrsavanje, vibracije, dinamični modeli)

This paper looks at theoretical research of the dynamic model of the "roller-band" system of the Rolamite-type mechanism (RTM). The dynamic model of the "roller-band" system of the RTM is described by differential equations and is investigated in two ways, i.e., when the system is not influenced by the excitation of vibrations, and when it is influenced by them. We have established that with the parameters of excitation of vibration it is possible to operate the parameters of a non-slipping zone. This allows us to assume that in a real Rolamite-type mechanism it is possible to operate on the friction between the structural elements of the RTM.

Experimental research on the dynamics of the Rolamite-type mechanism with vibrating elements was done in three directions: research on the dynamic processes running in the RTM roller-band system; research on the processes of friction-control between the elements of the RTM; and research on the dynamic processes running in the Rolamite vibromotors. The increase of the supply-voltage amplitude, the decrease of the force-load magnitude of the flexible band, and the decrease of the angle of the wrapping of the roller by the flexible band cause an increase in the amplitude of the roller vibrations. An RTM with rotating vibrating rollers, compared with the vibrating rollers of the other type, has a better bearing capacity and is much more sensitive. The highest synchronicity of rotation of the rotors is provided by a Rolamite vibromotor with two rotors and a piezoelectric converter of vibrations that simultaneously rotates both rotors. This is compared with the Rolamite vibromotor with two rotors and a piezoelectric converter of vibrations, which rotates only one rotor.

(Keywords: Rolamite type mechanisms, slipping, vibrations, dynamic models)

0 UVOD

Donald F. Wilkes je izumil kotalni mehanizem tipa Rolamite (RTM) v letu 1967 ([1] in [2]).

Pri mehanizmu RTM se valja brez trenja kotalita po jermenu, vendar pa avtorji [3] tudi navajajo, da valja pri določenih parametrih mehanizma zdrsneta, čeprav ne podajo nikakršne teoretične podlage za ta pojav. V viru [4] sta navedeni dve vrsti zdrsa mehanizma RTM (geometrični in vzmetni zdrs).

Raziskovalci so predlagali, da bi trenje zmanjšali z uporabo vibracij [5]. Ena izmed metod za zmanjšanje trenja temelji na rabi usmerjevalnih vibracij, ki jih dovajamo stičnim telesom.

Če se vibracijska amplituda vibrajočih elementov med stičnima telesoma povečuje, se pojavi plast vibracij (FV). Mehanizem nastanka plasti vibracij je temeljito opisan v viru [6], zato tu ne bo podrobno opisan.

Najpogosteje uporabljeni sestavni deli vibrajočih elementov mehanizma RTM so piezoelektrični pretvorniki, tj. piezo keramični elementi. Pri mnogih konstrukcijah mehanizma RTM le-ti tvorijo dele valjev, ki imajo osi pritrjene na nepremično osnovo. Okrov mehanizma RTM se giblje premočrtno pod vplivom zunanjih sil ali sil, ki jih ustvarja kinematično delovanje valjev in jermena. Pri takšnem mehanizmu valja opravljata funkcijo vibracijske podpore (VS) ali vibramotorja (VM) [7].

Ko načrtujemo precizni mehanizem RTM z nadzorovanim trenjem za določitev ali krmiljenje napetosti gibkega jermena kakor tudi natančni mehanizem vlečnega jermena z vibramotorjem Rolamite (RVM), moramo teoretično in eksperimentalno raziskati dinamične pojave, ki potekajo v mehanizmu RTM in vibramotorju.

Vibramotorje lahko v mehanizmu RTM uporabimo kot vibracijsko podporo usmerjevalnemu gibanju, mehanizmu vlečnega jermena, črevesni črpalki, mikro vpenjalni čeljusti, idr.

Namen teoretične raziskave je razkritje razmer, v katerih se valja in jermen mehanizma RTM gibljejo sinhrono in brez zdrsov, saj je poznavanje teh razmer bistvenega pomena pri oblikovanju natančnega mehanizma. Drugi namen raziskave pa je analiza mehanizma RTM, ki je izpostavljen vplivu vibracij.

Namen eksperimentalnega dela raziskave je seznanitev z dinamičnimi pojavi v mehanizmu valjev in jermena, analiza možnosti krmiljenja trenja

0 INTRODUCTION

Donald F. Wilkes invented the Rolamite-type mechanism (RTM) in 1967 ([1] and [2]).

In the RTM the rollers are rolling on the band without friction; however, the authors of [3] specify that the rollers slip for certain parameters of the mechanism, although they do not provide any theoretical substantiation for this. In [4] two variants of slipping in the Rolamite-type mechanism (geometric and springy slipping) are defined.

Researchers have suggested that the friction could be decreased using vibrations [5]. One of the methods to reduce friction is to provide directional vibrations to the contact bodies.

If the vibrational amplitude of the vibrating elements is increasing between the interacting bodies a film of vibrations (FV) occurs. The mechanism of the FV's formation is comprehensively described in [6], thus we will not describe it in detail here.

The most frequently used constituent parts of the RTM's vibrating elements are piezoelectric converters i.e., piezoceramic elements. In many RTM constructions they form part of the rollers that have their axes fixed to the immobile base. The casing of the RTM under external forces or forces arising in the kinematic roller-band pair performs a rectilinear movement. The rollers of such an RTM perform the functions of vibrosupport (VS) or vibromotors (VM) [7].

When designing precision Rolamite mechanisms with controlled friction for positioning or regulating the tension of the flexible band, as well as precision band-pulling mechanisms with a Rolamite vibromotor (RVM), the dynamic processes taking place in the RTM and RVM have to be explored both theoretically and experimentally.

Vibromotors may be used in a RTM as the vibrosupport of the directional movement, band-pulling mechanisms, peristaltic pumps, micro-manipulator grips, etc.

The purpose of theoretical research is to reveal the conditions under which the bodies of the roller-band of an RTM are moving synchronically without slipping, because their detection is the most important problem in the design of precision mechanisms. The other purpose is to investigate the RTM that is influenced by vibrations.

The purpose of experimental research is to explore the dynamic processes in the RTM roller-band system; to explore the possibilities of control-mehanizma RTM ter analiza dinamičnih pojavov v vibramotorjih.

Predmeti ekperimentiranja so izvirni mehanizmi RTM, pri katerih sta valja vzbujena z visokofrekvenčnim vibriranjem, ter vibramotor, čigar delovanje je utemeljeno na visokofrekvenčnih diagonalnih udarcih piezokeramične plošče ob vrteči se rotor.

1 DINAMIČNI MODELI VALJEV IN JERMENA PRI MEHANIZMU RTM

Dinamični modeli sklopa valjev in jermena pri mehanizmu RTM v primeru deformiranih in nedeformiranih stikov med telesi (sl. 1) so opisani z deferencialnimi enačbami in analizirani.

Slika 1 kaže, da so z_i (i = 1 do 6) pomiki elementov (v pravokotni in strižni smeri), $\vec{P}_1$ je vlečna sila, $\vec{P}_6$ je sila odpora gibanju, $\vec{P}_4$ je sila, ki pritiska maso m ob maso M (in povzroča pravokotno tlačno silo N).

Jermen opišemo z dvojicami parametrov. V sredini stičnega predela med valjem in jermenom je zgoščena masa jermena m, preostali del jermena pa je v vzdolžni smeri spremenjen v elastični element c_{12} in razsipni element H_{12}, ki sta povezana vzporedno. Masa jermena m je zbrana vzdolž površine valja. Valj je izražen kot telo z maso M.

Medsebojno delovanje jermena in valja je prikazano s prej omenjeno zgoščenje mase m ter s primerno povezavo elastičnih elementov c_{23}, c_{45} in razsipnih elementov H_{23}, H_{45}, ti omogočajo deformacije jermena, ki se dotika valja v strižni in pravokotni smeri. Zgoščena masa se deloma nanaša na maso m in deloma na maso m_3, ki je zbrana neposredno na površini valja in pridobljena s telesom, ki ima maso M.

ling the friction of the Rolamite mechanisms; and to explore the dynamic processes in Rolamite vibromotors.

The objects of the experiments are the original RTM, where the rollers are induced by high-frequency vibration, and the RVM, where the functioning is based on high-frequency diagonal hits by the piezoceramic plate onto the rotor that is rotating.

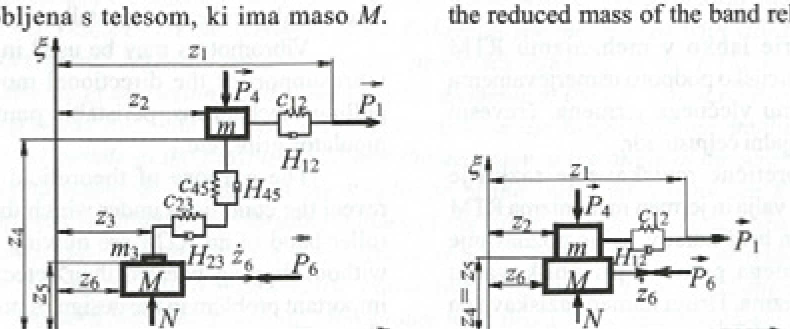
1 DYNAMIC MODELS OF THE ROLLER-BAND SYSTEM OF AN RTM

Dynamic models of the roller-band system of an RTM in the cases of deformed and non-deformed contact between the bodies (Fig. 1) were described by differential equations and investigated.

It is shown in Fig. 1 that z_i (i – 1 to 6) are the displacements of the elements (in the normal and tangential directions), $\vec{P}_1$ is the pulling force, $\vec{P}_6$ is the force of the resistance to movement, $\vec{P}_4$ is the force that is pressing the mass m to mass M (causing the normal pressing force N).

The band is represented by coupled parameters. In the center of the contact zone between the roller and the band is concentrated the band's reduced mass m, the remaining piece of the band in a longitudinal direction is changed to the elastic element c_{12} and the dissipative element H_{12}, connected in parallel. The mass of the band m is concentrated along the roller's surface. The roller is expressed as a body with the mass M.

The interaction of a band with a roller is shown by means of the above-mentioned reduced mass m, and connecting in appropriate way the elastic c_{23}, c_{45} and dissipative H_{23}, H_{45} elements, which allow deformations of the band contacting the roller in the tangential and normal directions. The part of the reduced mass of the band relates to the mass m



Sl. 1. *Dinamična modela valja in jermena pri mehanizmu RTM v primeru stika med telesoma: a – deformirani stik, b – nedeformirani stik*

Fig. 1. *Dynamic models of the roller-band system of the RTM in the case of a contact between bodies: a – deformed, b – non-deformed*

Deformacija stika med jermenom in obema valjema je mogoča zaradi elastičnih in razsipnih elementov c_{23}, c_{45}, H_{23}, H_{45} ter tudi zaradi mase m_3.

Analiziran je primer neprekinjenega gibanja mas m in M v pravokotni smeri.

Poenostavljeni dinamični model natančnega mehanizma valja in jermena, kakršnega poznamo pri sistemu RTM, je opisan z diferencialnimi enačbami in analiziran.

V skladu s sliko 1a imajo diferencialne enačbe gibanja naslednje izraze:

and the part to mass m_3, concentrated directly to the surface of a roller adopted by the way of a body with mass of M. The deformed contact between a band on both rollers is allowed by the elastic and dissipative elements c_{23}, c_{45}, H_{23}, H_{45} and also by mass m_3.

An example of the continuous movement of the masses m and M in the normal direction is analyzed.

The simplified dynamic model of a precision roller-band mechanism such as the RTM roller-band system was described by differential equations and investigated.

The differential equations of movement, in accordance with Fig.1 a, have these expressions:

$$\begin{gathered} H_{12}(\dot z_1 - \dot z_2) + c_{12}(z_1 - z_2) = P_1 \\ m\ddot z_2 - H_{12}(\dot z_1 - \dot z_2) - c_{12}(z_1 - z_2) + H_{23}(\dot z_2 - \dot z_3) + c_{23}(z_2 - z_3) = 0 \\ m_3\ddot z_3 - H_{23}(\dot z_2 - \dot z_3) - c_{23}(z_2 - z_3) + \left[H_{45}(\dot z_4 - \dot z_5) + c_{45}(z_4 - z_5)\right]\cdot f_0 sign(\dot z_3 - \dot z_6) + f(\dot z_3 - \dot z_6) = 0 \\ M\ddot z_6 - \left[H_{45}(\dot z_4 - \dot z_5) + c_{45}(z_4 - z_5)\right] f_0 \cdot sign(\dot z_3 - \dot z_6) - f(\dot z_3 - \dot z_6) = -P_6 \\ m\ddot z_4 + H_{45}(\dot z_4 - \dot z_5) + c_{45}(z_4 - z_5) = -P_4 \end{gathered} \quad (1).$$

V skladu s sliko 1b imajo diferencialne enačbe gibanja naslednje izraze:

The differential equations of movement, in accordance with Fig.1 b, have the following expressions:

$$\begin{gathered} H_{12}(\dot z_1 - \dot z_2) + c_{12}(z_1 - z_2) = P_1 \\ m\ddot z_2 - H_{12}(\dot z_1 - \dot z_2) - c_{12}(z_1 - z_2) + Nf_0 sign(\dot z_2 - \dot z_6) + f(\dot z_2 - \dot z_6) = 0 \\ M\ddot z_6 - Nf_0 sign(\dot z_2 - \dot z_6) - f(\dot z_2 - \dot z_6) = -P_6 \\ m\ddot z_2 + P_4 = N \end{gathered} \quad (2),$$

H_{12}, H_{23}, H_{45} - koeficienti viskoznega dušenja;
c_{12}, c_{23}, c_{45} - koeficienti togosti;
f_0 - koeficient suhega trenja;
f - koeficient mokrega trenja.

H_{12}, H_{23}, H_{45} – viscous damping coefficients;
c_{12}, c_{23}, c_{45} – coefficients of stiffness;
f_0 – coefficient of dry friction;
f – coefficient of wet friction.

Včasih je koeficient mokrega trenja f sorazmeren vrednosti pravokotne reakcije, tj.:

Sometimes the coefficient of wet friction, f, is proportional to the value of the normal reaction, i.e.:

$$f = f_1 N_{45} \quad (3),$$

f_1 - koeficient mokrega trenja.

f_1 – coefficient of wet friction.

$$N_{45} = H_{45}(\dot z_4 - \dot z_5) + c_{45}(z_4 - z_5) \quad (4).$$

V primeru zunanjega vzbujanja:

For external excitation:

$$\begin{gathered} P_1 = A_1 + B_1\dot z_1 + D_1\sin(\omega t + \alpha_1) \\ P_4 = A_4 + B_4\dot z_4 + D_4\sin(\omega t + \alpha_4) \\ P_6 = A_6 + B_6\dot z_6 + D_6\sin(\omega t + \alpha_6) \\ z_5 = D_5\sin(\omega t + \alpha_5) \\ \cdot = \frac{d}{dt} \end{gathered} \quad (5).$$

A_1, A_4, A_6 komponente nespremenljivih sil;
B_1, B_4, B_6 stalnice, ki določajo linearno razmerje med silami in ustreznimi hitrostmi;
D_1, D_4, D_6 amplitude komponent harmoničnih sil;
ω kotna frekvenca;
$\alpha_1, \alpha_4, \alpha_6, \alpha_5$ faze.

A_1, A_4, A_6 constant force components;
B_1, B_4, B_6 constants defining the linear relationship between the forces and the appropriate velocities;
D_1, D_4, D_6 amplitudes of the harmonic force components;
ω angular frequency;
$\alpha_1, \alpha_4, \alpha_6, \alpha_5$ phases.

Da bi ocenili delovanje sistema, moramo upoštevati osnovne značilke, na primer delo ali moč, gonilne sile, silo koristnega upora itn.

To estimate the work of the system's performance we consider basic characteristics, for example, the work or power, the moving forces, the force of useful resistance, etc.

Glede na sliko 1 je delo gonilnih sil opisano kot:

According to Fig.1, the work of moving forces is described as:

$$A_m = \int_0^{H_z} P_1 dz_1 = \int_0^T P_1 \dot{z}_1 dt \qquad (6).$$

T - čas integracije;
H_z - značilna razdalja opravljena v času T.

T - time of integration;
H_z - appropriate distance covered during the time T.

Dejansko delo je podano z naslednjim izrazom:

The useful work is given by the expression:

$$A_u = \int_0^{H_z} P_6 dz_6 = \int_0^T P_6 \dot{z}_6 dt \qquad (7).$$

Izkoristek je podan z naslednjim izrazom:

The efficiency is given by the expression:

$$\eta = \frac{A_u}{A_m} = \frac{\int_0^T P_6 \dot{z}_6 dt}{\int_0^T P_1 \dot{z}_1 dt} \qquad (8).$$

Nepravilnost hitrosti gibanja je podana z naslednjim izrazom:

The irregularity of the motion velocity is given by the expression

$$\vartheta_{\dot{z}} = \frac{\dot{z}_{max} - \dot{z}_{min}}{\bar{\dot{z}}} \qquad (9),$$

$$\bar{\dot{z}} = \frac{\dot{z}_{max} + \dot{z}_{min}}{2} \qquad (10).$$

Z vnosom nove spremenljivke dobimo naslednje:

By entering a new variable we obtain:

$$x_i = \frac{z_i}{l} \quad (i = 1,...,6)\,;\; p = \sqrt{\frac{c_{12}}{m}}\,;\; \tau = pt\,;\; ' = \frac{d}{d\tau}\,;\; v = \frac{\omega}{p}\,;\; 2h_{rs} = \frac{H_{rs}}{pm} \quad (rs = 12, 23, 45)\,;\; N' = \frac{N}{p^2 ml}\,;\; \mu = \frac{M}{m}\,;\; \mu_3 = \frac{m_3}{m}\,;$$
$$F_j = \frac{P_j}{p^2 ml} = \frac{P_j}{c_{12} l} \quad (j = 1,4,6)\,;\; a_j = \frac{A_j}{c_{12} l}\,;\; b_j = \frac{B_j}{pm}\,;\; d_j = \frac{D_j}{c_{12} l}\,; d_5 = \frac{D_5}{l}\,; \delta_{23} = \frac{c_{23}}{c_{12}}\,;\; \delta_{45} = \frac{c_{45}}{c_{12}} \qquad (11),$$

l - dolžina jermena;
p in τ sta novi spremenljivki.

l - band length;
p and τ are new variables.

Če upoštevamo novo spremenljivko (11), diferencialne enačbe gibanja (1) spremenimo takole:

Considering a new variable (11), the differential equations of movement (1) are converted to the type:

$$\begin{gathered}
2h_{12}(x_1' - x_2') + (x_1 - x_2) = F_1, \\
x_2'' - 2h_{12}(x_1' - x_2') - (x_1 - x_2) + 2h_{23}(x_2' - x_3') + \delta_{23}(x_2 - x_3) = 0 \\
\mu_3 x_3'' - 2h_{23}(x_2' - x_3') - \delta_{23}(x_2 - x_3) + \left[2h_{45}(x_4' - x_5') + \delta_{45}(x_4 - x_5)\right] \cdot \left[f_0 sign(x_3' - x_6')\right] + f(x_3' - x_6') = 0 \\
\mu x_6'' - \left[2h_{45}(x_4' - x_5') + \delta_{45}(x_4 - x_5)\right] f_0 \cdot sign(x_3' - x_6') - f(x_3' - x_6') = -F_6 \\
x_4'' + 2h_{45}(x_4' - x_5') + \delta_{45}(x_4 - x_5) = -F_4
\end{gathered} \qquad (12),$$

h_{12}, h_{23}, h_{45} so koeficienti dušenja

h_{12}, h_{23}, h_{45} – coefficients of damping.

Če upoštevamo novo spremenljivko (11), diferencialne enačbe gibanja (2) spremenimo takole:

Considering the new variable (11), the differential equations of movement (2) are converted to the type:

$$\begin{gathered}
2h_{12}(x_1' - x_2') + (x_1 - x_2) = F_1 \\
x_2'' - 2h_{12}(x_1' - x_2') - (x_1 - x_2) + N' f_0 sign(x_2' - x_6') + f(x_2' - x_6') = 0 \\
\mu x_6'' - N' f_0 sign(x_2' - x_6') - f(x_2' - x_6') = -F_6 \\
x_5'' + F_4 = N'
\end{gathered} \qquad (13).$$

Nato enačbo (3) spremenimo:

Furthermore, Equation (3) is converted to the type:

$$f = f_1\left[2h_{45}(x_4' - x_5') + \delta_{45}(x_4 - x_5)\right] \qquad (14).$$

Zunanje vzbujanje (5) v neparametričnem izrazu vodi v naslednje:

The external excitation (5) in the non-parametric expression leads to:

$$\begin{aligned} F_1 &= a_1 + b_1 x_1' + d_1 \sin(\nu\tau + \alpha_1) \\ F_4 &= a_4 + b_4 x_4' + d_4 \sin(\nu\tau + \alpha_4) \\ F_6 &= a_6 + b_6 x_6' + d_6 \sin(\nu\tau + \alpha_6) \\ x_5 &= d_5 \sin(\nu\tau + \alpha_5) \end{aligned} \quad (15),$$

kjer je $\nu = \omega / p$.

Izkoristek je podan z naslednjim izrazom:

here $\nu = \omega / p$.

The efficiency is given by the expression:

$$\eta = \frac{A_u}{A_m} = \frac{\int_0^T F_6 x_6' d\tau}{\int_0^T F_1 x_1' d\tau} \quad (16).$$

Nepravilnost hitrosti gibanja je podana takole:

The irregularity of the motion's velocity is given by the expression:

$$\delta x_s' = \frac{x_{s\max}' - x_{s\min}'}{\overline{x_s'}} \quad (s = 1, 2, 6) \quad (17),$$

$$\overline{x_s'} = \frac{x_{s\max}' + x_{s\min}'}{2} \quad (18).$$

2 ANALIZA DINAMIČNIH POJAVOV V DELOVANJU SKLOPA VALJEV IN JERMENA PRI MEHANIZMU RTM, KADAR NANJE NE VPLIVA VIBRACIJSKO VZBUJANJE

Namen te raziskave je definiranje dela, ki ga opravi sklop valjev in jermena pri mehanizmu RTM v odsotnosti vibracijskega vzbujanja.

Raziskava je opravljena na nezdrsnem območju (NZ) upoštevajoč parametre mehanizma. Nezdrsnost je okoliščina, v kateri se telesa raziskovanega mehanizma pomikajo skupaj brez zdrsovanja, tj. $x'_1 = x'_2 = x'_6$ (sl. 2 do 7):

2 THE ANALYSIS OF DYNAMIC PROCESSES OCCURRING IN THE "ROLLER-BAND" SYSTEM OF THE RTM, NOT INFLUENCED BY THE EXCITATION OF VIBRATIONS

The purpose of the research is to define the work of the "roller-band" system of the RTM in the absence of vibrating excitation.

It is presented in the research of a non-slipping zone of bodies (NZ), depending on the parameters of the system. The non-slipping zone is understood as the condition when the bodies of the investigated system are moving together without slipping, i.e., $x'_1 = x'_2 = x'_6$ (Figs. 2 to 7):

$$d_1 = d_4 = d_5 = d_6 = 0 \quad (19),$$

kakor se to dogaja v obravnavanem primeru. Raziskava analizira dinamične pojave, ki potekajo v mehanizmu valjev in jermena v sistemu natančnega valja, tj. v jermenskem mehanizmu z nedeformirajočim stikom med telesi.

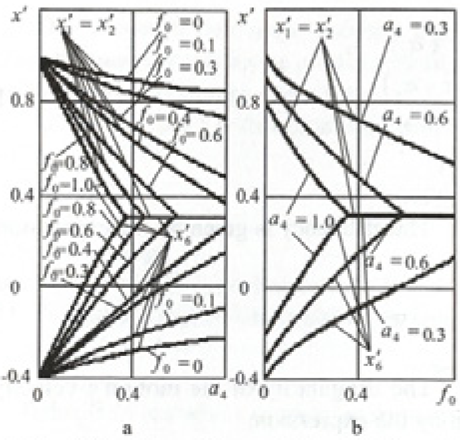
Značaj dinamičnih pojavov v mehanizmu valjev in jermena z deformirajočim stikom med telesi v enakomernem gibanju je analogen pojavom, ki so v primeru nedeformirajočega stika. Iz krivulj diagrama zato lahko določimo razmerja med enakomernim gibanjem in njegovimi parametri. Pod vsakim diagramom so v oklepaju prikazani parametri sistema z deformirajočim stikom med telesi.

Glede na sliko 2 pri definirani vrednosti a_4 hitrosti x'_1, x'_2 in x'_6 proučevanih teles postanejo istovetne, kar pomeni, da telesa ne zdrsujejo v tolikšni meri, da bi se znatno povečal koeficient suhega trenja

as in the considered case. It is presented as research of the dynamic processes occurring in the "roller-band" system of a precision roller, i.e., band mechanisms with a non-deforming contact between the bodies.

The character of the dynamic processes occurring in the "roller-band" system with a deforming contact between the bodies in steady motion is analogous to the processes occurring in this system with a non-deforming contact. Furthermore, from the graphical curves we can determine the relations of steady motion with its parameters. In brackets, under the figures, the parameters of the system with a deforming contact between the bodies are indicated.

According to Fig. 2, at a defined value a_4, the velocities x'_1, x'_2 and x'_6 of the investigated bodies become identical, i.e., the bodies do not slip between themselves at a sufficient increase of the coefficient of



Sl. 2. *Grafa kažeta odvisnost hitrosti teles sistema pri enakomernem gibanju od parametrov gibanja*

Fig. 2. *Graphs showing the dependence of the velocity of bodies of a system in steady motion on the parameters of motion*

$a_1 = 0,5;\ a_6 = -0,2;\ b_1 = b_6 = -0,5;\ 2h_{12} = 0,2;$
$\mu = 1,0;\ f_1 = 0,1\ (2h_{23} = 2h_{45} = 2,0;\ \delta_{23} = \delta_{45} = 10;$
$\mu_3 = 0,01;\ b_4 = -0,5)$

f_0 (sl. 2 a). Spoznali bomo, da večja ko je vrednost a_4 pri f_0 = konst., hitreje nastane nezdrsno območje med telesi (sl. 2 b). To pokaže tudi slika 3.

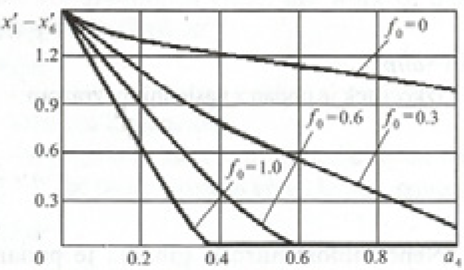
Tako je, pri definirani vrednosti f_0, zvečanje vrednosti a_4 pomembno le, dokler se približuje vrednosti NZ. Ko pa to vrednost doseže, izgubi pomen, saj se telesa zdaj začnejo gibati z enakimi hitrostmi (sl. 2 in 3).

S povečanjem a_4, pri koeficientu suhega trenja f_0 = konst., NZ postane definirana vrednost a_4 ne glede na koeficient viskoznega trenja f_1 (sl. 4 a). Pri dani vrednosti a_4 povečana vrednost f_1 povzroči le rahlo zmanjšanje hitrosti x'_1 (krivulje odvisnosti v diagramu postanejo bolj izbočene) (sl. 5 b); vendar pa NZ, pri dani vrednosti f_0, postane definirana vrednost a_4, hitrosti x'_1, x'_2, x'_6 pa se izenačijo (sl. 4 a in b).

Pri definirani vrednosti a_4 se hitrosti x'_1, x'_2, x'_6 obravnavanih teles izenačijo, hkrati pa se zmanjša vrednost a_1, kar pomeni, da manjša ko je vrednost a_1, večje je nezdrsno gibanje med telesi (sl. 5 a). Pri a_1 = konst. se zdrsovanje teles zmanjša ob povečanju a_4 (sl. 5 b). Krivulje odvisnosti na sliki 5 se ostro prelomijo ob vstopu v NZ.

Če je a_6 = konst. in se vrednost a_4 zveča, se poveča hitrost nastanka pojava NZ. Če je a_4 = konst., se navor nezdrsnosti med telesi pojavi ob povečanju a_6 do svoje določene vrednosti (sl. 6 a in b).

V času povečevanja a_6 se povečajo tudi hitrosti $x'_1 = x'_2 = x'_6$ (Sl. 6 a); kljub temu je pojav NZ



Sl. 3. *Graf kaže odvisnost hitrosti zdrsovanja med telesi sistema pri enakomernem gibanju od parametrov gibanja*

Fig. 3. The *graph showing the dependence of the velocity of slipping between bodies of a system in steady motion on the parameters of motion*

$a_1 = 0,5;\ a_6 = -0,2;\ b_1 = b_6 = -0,5;\ 2h_{12} = 0,2;$
$\mu = 1,0;\ f_1 = 0,1\ (2h_{23} = 2h_{45} = 2,0;\ \delta_{23} = \delta_{45} = 10;$
$\mu_3 = 0,01;\ b_4 = -0,5)$

dry friction, f_0 (Fig. 2 a). It will be clear that the greater the value a_4 at f_0 = const, the faster it makes a non-slipping zone of the bodies (Fig. 2 b). Fig. 3 also shows this.

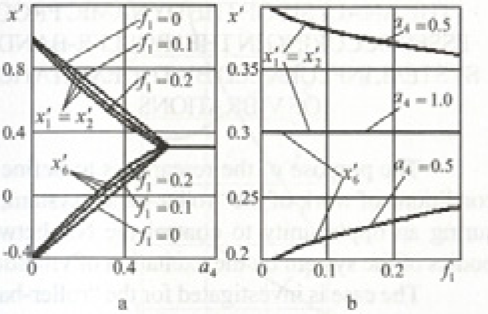
Furthermore, at a defined value, f_0, the increasing value of a_4 is meaningful only up to NZ. After reaching this value it loses sense, as the bodies start to move with an identical velocity (Fig. 2 and 3).

With increasing a_4 at the coefficient of dry friction f_0 = const, NZ becomes a definite value a_4, irrespective of the coefficient of viscous friction, f_1 (Fig. 4 a). The increasing f_1 resulted only in a minor decrease of the velocities x'_1 at a given data a_4 (the dependence curves in the figure have a more convex character) (Fig. 5 b); however, NZ at a given data f_0 becomes a defined value a_4, and the velocities x'_1, x'_2, x'_6 become equal (Fig. 4 a and b).

At a defined value a_4 the velocities x'_1, x'_2, x'_6 of the investigated bodies level off, with a decreasing of value a_1, i.e., the smaller is a_1, the faster the non-slipping movement between the bodies comes (Fig. 5 a). At a_1 = const the slipping of the bodies decreases with an increase of a_4 (Fig. 5 b). The curves of dependences in Fig. 5 break sharply when going into NZ.

If a_6 = *const* and value a_4 increases, NZ becomes faster. If a_4 = const, the moment of non-slipping between the bodies comes with an increase of a_6 up to its definite value (Fig. 6 a and b).

During the increase of a_6 the velocities of bodies $x'_1 = x'_2 = x'_6$ are increased (Fig. 6 a); however,



Sl. 4. *Grafa kažeta odvisnost hitrosti teles sistema pri enakomernem gibanju od parametrov gibanja*

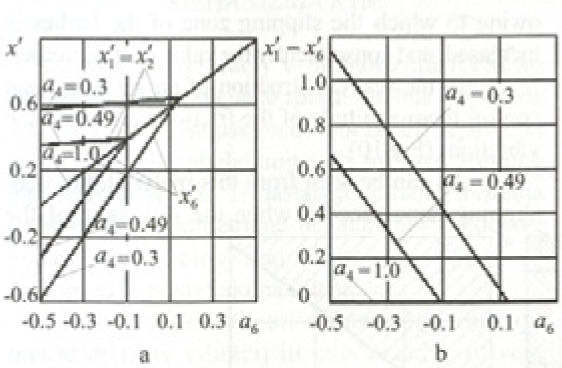
Fig. 4. *Graphs of the dependence of the velocity of the bodies of a system in steady motion on the parameters of motion*

$a_1 = 0{,}5;\ a_6 = -0{,}2;\ b_1 = b_6 = -0{,}5;\ 2h_{12} = 0{,}2;\ \mu = 1{,}0;\ f_0 = 0{,}6\ (2h_{23} = 2h_{45} = 2{,}0;\ \delta_{23} = \delta_{45} = 10;\ \mu_3 = 0{,}01;\ b_4 = -0{,}5)$

mogoč, in sicer pri zmanjšani vrednosti a_4 (Sl. 7 a). Če je a_4 = konst., NZ nastane ob povečanju f_0, v nasprotnem primeru pa, ko je f_0 = konst., NZ nastane ob povečanju a_4 (Sl. 7 b).

Glede na sliko 7 povečanje a_4 in a_6 povzroči nezdrsno območje med telesi. Pri definirani vrednosti a_6 naraščanje vrednosti a_4 po vstopu teles v območje NZ nima več pomena, ker se telesa zdaj gibljejo z enakimi hitrostmi. Na spodnjih slikah je območje NZ šrafirano.

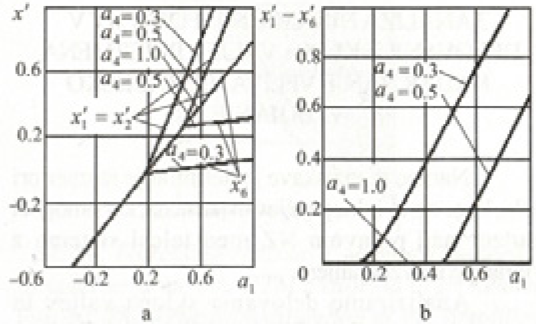
Poudariti je treba, da je $f_0\ a_4$ = konst.



Sl. 6. *Grafa kažeta odvisnost hitrosti teles sistema pri enakomernem gibanju od parametrov gibanja*

Fig. 6. *Graphs of the dependence of the velocity of the bodies of a system in steady motion on the parameters of motion*

$b_1 = b_6 = -0{,}5;\ 2h_{12} = 0{,}2;\ f_0 = 0{,}6;\ f_1 = 0{,}1\ (2h_{23} = 2h_{45} = 2{,}0;\ \delta_{23} = \delta_{45} = 10;\ \mu_3 = 0{,}01;\ b_4 = -0{,}5;\ a_1 = 0{,}5)$



Sl. 5. *Grafa kažeta odvisnost hitrosti teles sistema pri enakomernem gibanju od parametrov gibanja*

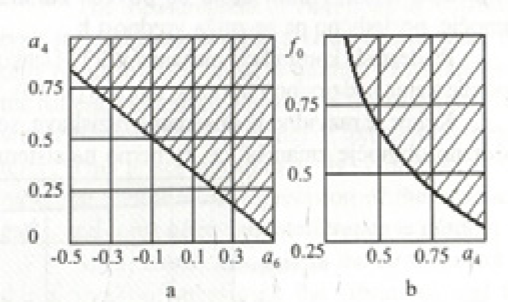
Fig. 5. *Graphs of the dependence of the velocity of the bodies of a system in steady motion on the parameters of motion*

$b_1 = b_6 = -0{,}5;\ 2h_{12} = 0{,}2;\ f_0 = 0{,}6;\ f_1 = 0{,}1;\ a_6 = -0{,}2\ (2h_{23} = 2h_{45} = 2{,}0;\ \delta_{23} = \delta_{45} = 10;\ \mu_3 = 0{,}01;\ b_4 = -0{,}5)$

the coming of NZ is possible, and at a smaller value of a_4 (Fig. 7 a). At $a_4 = const$ NZ comes with increasing f_0 and, in contrast, at $f_0 = const$ with increasing a_4, we achieve NZ (Fig. 7 b).

According to Fig. 7, the increasing a_4 and a_6 resulted in a non-slipping between the bodies. At a defined value a_6 the increasing value of a_4 after the arrival of the bodies in NZ has no sense, because they are moving with identical velocities. NZ is hatched.

It is necessary to point out that $f_0\ a_4$ = const.



Sl. 7. *Področja obstoja menjavajočih se enakomernih režimov gibanja teles, odvisnih od parametrov gibanja*

Fig. 7. *The areas of existence of varying steady regimes of motion of a system of bodies depending on its parameters*

$a_1 = 0{,}5;\ b_1 = b_6 = -0{,}5;\ f_1 = 0{,}1;\ 2h_{12} = 0{,}2\ (2h_{23} = 2h_{45} = 2{,}0;\ \delta_{23} = \delta_{45} = 10;\ \mu_3 = 0{,}01;\ b_4 = -0{,}5)$: a- $f_0 = 0{,}6$; b- $a_6 = -0{,}2$

3 ANALIZA DINAMIČNIH POJAVOV V DELOVANJU SKLOPA VALJEV IN JERMENA, KADAR NANJE VPLIVA VIBRACIJSKO VZBUJANJE

Namen te raziskave je definiranje razmer pri delu, ki ga opravi sklop valjev in jermena, kar omogoči nadzor nad pojavom NZ med telesi sistema z vibracijskim vzbujanjem.

Analiziramo delovanje sklopa valjev in jermena pri mehanizmu RTM, na katerega vplivamo z vibriranjem, tj. z ustvarjanjem naslednjega pogoja: $d_1, d_4, d_5, d_6 \neq 0$.

Da bi zagotovili pojav NZ, moramo povečati f_0 (pri tem je v = konst.), kar omogoči zmanjšanje vrednosti a_4, povečanje vrednosti v (pri čemer je f_0 = konst.) pa povzroči neznatno zmanjšanje vrednosti a_4 (sl. 8 a).

Če na sistem ne vplivamo z vibracijami, kar pomeni, da je $d_4 = 0$, NZ, in je f_0 = konst., ima a_4 manjši pomen kakor sicer (sl. 8 a in b).

Slednje lahko razložimo z zmanjšanjem sile trenja v primeru, ko vibracije delujejo na sistem in nastane območje NZ, ki ima povečane vrednosti a_4. Večja ko je amplituda vzbujajočih vibracij d_4, večja mora biti vrednost a_4, kar je potrebno za zagotovitev območja NZ (sl. 8 b).

Vrednost povprečne hitrosti $\overline{x_1'}$ se zveča, medtem ko se vrednost $\overline{x_6'}$ zmanjša s povečanjem amplitude vibracijskega vzbujanja d_5; koeficient izkoristka η se tako strmo zmanjša (sl. 9).

Dogajanje lahko razložimo z zmanjšanjem sile trenja med telesi, zaradi česar se poveča zdrsno območje, posledično pa se zniža vrednost h .

Pri dejanski konstrukciji mehanizma RTM lahko moč sile trenja nadziramo z uporabo vibracij (sl. 10).

Kakor je razvidno iz dosedanje raziskave, se nezdrsno območje zmanjša, če začnemo na sistem

3 THE ANALYSIS OF THE DYNAMIC PROCESSES OCCURING IN THE "ROLLER-BAND" SYSTEM, INFLUENCED BY THE EXCITATION OF VIBRATIONS

The purpose of the research is to define the conditions of work of the "roller-band" system, ensuring an opportunity to control the NZ between bodies of the system by the excitation of vibrations.

The case is investigated for the "roller-band" system of RTM when it is influenced by vibrations, i.e., $d_1, d_4, d_5, d_6 \neq 0$.

To ensure NZ, it needs to increase f_0 (having v=const), which ensures the decrease of a_4 and the increase of v (having f_0=const) ensures the non-considerable decrease of a_4 (Fig. 8 a).

If the system is not influenced by vibrations, i.e., d_4=0, NZ, when the meanings of f_0 = const, it will appear to have fewer meanings of a_4 (Fig. 8 a and b).

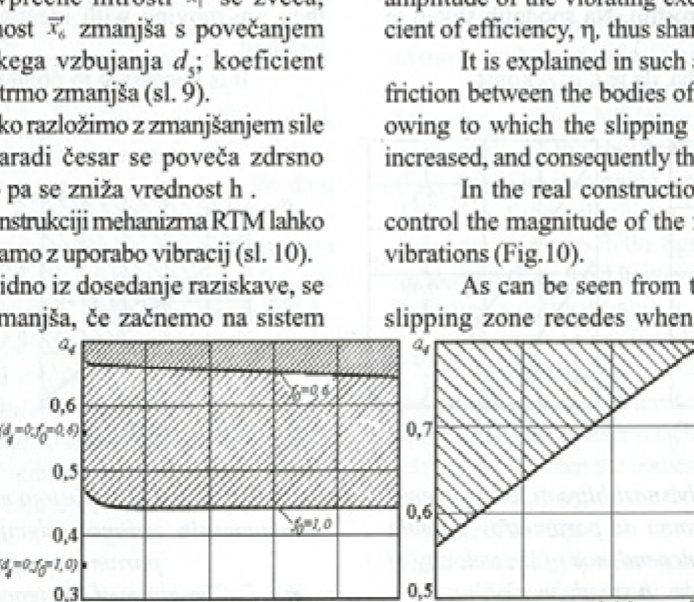
This can be explained by the friction force decreases in the case of vibrations influencing the system and NZ is reached, having greater meanings of a_4. The greater the amplitude of the exciting vibrations d_4, the greater must be the meaning of a_4; which is necessary to ensure NZ (Fig. 8 b).

The value of the average velocity $\overline{x_1'}$ increases, and $\overline{x_6'}$ decreases with the increase of the amplitude of the vibrating excitations d_5; the coefficient of efficiency, η, thus sharply decreases (Fig. 9).

It is explained in such a way that the force of friction between the bodies of the system decreases, owing to which the slipping zone of the bodies is increased, and consequently the value of η decreases.

In the real construction of an RTM we can control the magnitude of the friction force with the vibrations (Fig.10).

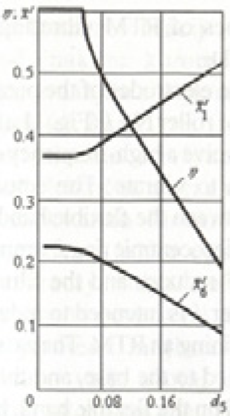
As can be seen from this material, the non-slipping zone recedes when the influence of the



Sl. 8. *Polja gibanja teles v različnih razmerah, odvisnih od parametrov vibracijskega vzbujanja*

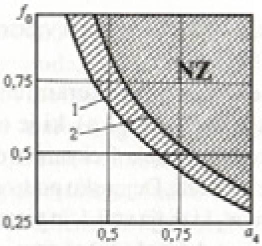
Fig. 8. *Fields of movements of bodies in different types of conditions, depending on the parameters of the exciting vibrations*

$a_1 = 0,5;\ a_6 = -0,2;\ b_1 = b_6 = -0,5;\ f_1 = 0,1;\ x_1' = x_2' = x_6' = 0,3;$ a) $d_4 = 0,1;$ b) $f_0 = 0,6;\ \nu/2\pi = 0,5.$



Sl. 9. *Prikaz odvisnosti povprečnih hitrosti gibanja teles in izkoristka od parametrov vibracijskega vzbujanja, ko je $f_0 = 1,0$*

Fig. 9. *The diagram of the dependence of the average velocities of the movement of the bodies and the efficiency on the parameters of the vibrating excitation at $f_0 = 1.0$*



Sl. 10. *Teoretična raziskava poenostavljenega dinamičnega modela RTM z valjema in jermenom*

Fig. 10. *Theoretical investigation of the simplified dynamic model of RTM "roller-band" system*

$a_1 = 0,5;\ a_6 = -0,2;\ b_1 = b_6 = -0,5;\ f_1 = 0,1;$
$x_1' = x_2' = x_6' = 0,3;\ 2h_{12} = 0,2;\ 2h_{23} = 2h_{45} = 2,0;$
$\delta_{23} = \delta_{45} = 10;\ \mu = 1,0,\ \mu_3 = 0,01;$

Sistem: 1 – vibriranje ni aktivirano ($d_4 = 0$); 2 – vibriranje je aktivirano ($d_4 = 0,1;\ \nu/2\pi = 0,1$), območje NZ je šrafirano

The system: 1 - vibrations do not influence ($d_4 = 0$), 2 - vibrations influence ($d_4 = 0.1; (\nu/2\pi)$ 0.1), NZ is hatched

vplivati z vibracijami. To pomeni, da je v dejanskem sistemu mogoče nadzirati zdrsovanje med elementi mehanizma RTM, tj. nadzirati navor odpornosti za gibanje. To je pomembno, kadar oblikujemo konstrukcije RTM, saj lahko tako zagotovimo boljšo kakovost in natančnost ter ustvarimo bolj natančne dinamične značilke.

system by vibrations starts. This means that in a real system it can be possible to control the slipping-by between the elements of the RTM, i.e., control of the moment of resistance to motion. This is important when designing the constructions of the RTM, which ensures higher quality and precision and more precise dynamic characteristics.

4 ANALIZA DINAMIČNIH POJAVOV MEHANIZMA RTM

Meritve vibracij v mehanizmu RTM so težavna naloga, toda ne le zaradi majhne amplitude vibracij v valjih, ampak predvsem zaradi dejstva, da poleg vibracij zabeležimo tudi njihove številne medsebojne vplive. Ti nastanejo zaradi vibriranja podstavka in geometrične nenatančnosti izdelave posameznih delov mehanizma; ti vplivi se najpogosteje pojavljajo naključno.

Veliko pozornosti namenjamo razvoju metod meritev vibracij in izločevanju vplivov. Metode stične raziskave so omejene, ker vplivi pogosto popačijo rezultate beleženja in tako znatno vplivajo na izid poizkusa. Precej obetavne pa so nestične optične metode meritev, ki v polni meri izpolnjujejo zahteve poizkusov, tj. potrebe po nestičnih, zelo natančnih meritvah in veliki ločljivosti. Te metode so bile uporabljene v analizi značilk vibriranja v valjih mehanizma RTM.

4 RESEARCH OF THE DYNAMIC PROCESSES RUNNING IN THE RTM ROLLER-BAND SYSTEM

The measurement of vibrations in Rolamite-type mechanisms is quite a complicated task, not only because of the small amplitude of vibrations in the rollers, but mainly because of the fact that along with vibrations there is plenty of interference registered. This arises because of the vibration of the base, the geometrical imprecision of the produced parts, and most often the interference is random.

Much attention is paid to the development of the methods of measuring the vibrations and the separation of interference. The methods of contact research are limited, because the interference quite often distorts the results of the registration and significantly influences the results of the experiment. Quite promising are the non-contact optical methods of measurement, which fully meet the demands of the experiment, i.e. non-contact, high-precision measurements, high dimensional resolution. These methods were used in the research of the characteristics of vibration in the RTM rollers.

Raziskana sta bila dva tipa vibrirajočih valjev mehanizma RTM (sl. 11).

Ko elektrode piezokeramičnega obroča pritrjenega valja št. *1* (Sl. 11 a), ki je ovit z gibkim jermenom, sprejmejo visokofrekvenčni električni signal, valj začne vibrirati. Dejansko področje stika med gibkim jermenom, ki ovija valj *1*, in piezokeramičnim obročem *2*, ki obdaja elastični torni obroč *3*, se zmanjša in pojavi se plast vibracij. Valj *1* je namenjen zmanjšanju trenja v napravah, ki vključujejo mehanizem RTM. Os valja *1* je čvrsto pritrjena na osnovo, tako da se valja mehanizma RTM ne kotalita po gibkem jermenu, ampak drsita.

Eksperimentalni valj št. *2* (sl. 11 b) je namenjen zmanjšanju trenja v napravah, ki vključujejo mehanizem RTM. Toda valja takšnega mehanizma se kotalita po gibkem jermenu.

Valj (sl. 11 b) je sestavljen iz osi *1*, ki je čvrsto pritrjena na osnovo, ter iz piezokeramičnega obročastega elementa, ki je montiran na os. Na notranjo stran piezokeramičnega obroča *2* je pritrjen elastični torni obroč *3*, ki je v stiku z osjo *1*; z zunanje strani pa je obroč *2* prekrit s togim obročastim pokrovom, ki je čvrsto pritrjen na obroč in je v stiku z ovijajočim gibkim jermenom. Vibracije valja med montiranim obročnim piezokeramičnim elementom, elastičnim tornim obročem *3* in pritrjeno togo osjo *1* ustvarjajo tlačeno plinasto plast (PGF).

Raziskali smo vibracije valjev v prečni smeri ter v primerih obremenitve in neobremenitve (sl. 12).

V primeru obremenitve je valj ovit z gibkim jermenom pod kotom od $\pi/4$ do $3/2\ \pi$. En konec jermena je pritrjen na togo osnovo, utež pa je pritrjena na drugi konec jermena. Kakor je očitno iz izvedenega preizkusa, se amplituda in

Two types of RTM vibrating rollers were explored (Fig. 11).

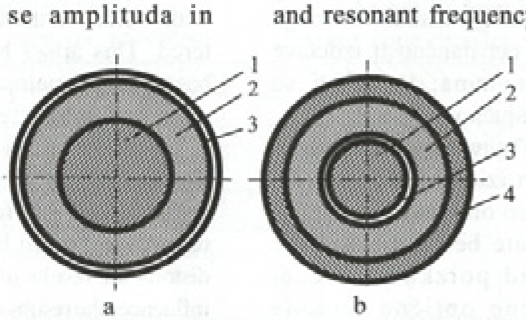
When the electrodes of the piezoceramic ring of the fixed RTM roller No *1* (Fig. 11 a) wrapped by a flexible band receive a high-frequency electric signal, the roller starts to vibrate. The actual area of the contact zone between the flexible band wrapping the roller *1* and the piezoceramic ring *2* wrapping the elastic frictional ring *3* reduces and the film of vibrations occurs. The roller *1* is intended to reduce the friction in devices containing an RTM. The axis of the roller *1* is fixedly attached to the base, and the rollers of the RTM do not roll on the flexible band, but slip.

The experimental roller No *2* (Fig. 11 b) is intended to reduce the friction in devices containing an RTM. But the rollers of such an RTM roll on the flexible band.

The roller (Fig. 11 b) consists of an axis *1*, fixedly attached to the base, and of an assembled piezoceramic ring element that is put on it. It is made of a piezoceramic ring *2*, in which the internal side of an elastic frictional ring *3* is fixed; this contacts axis *1*, and on the outside ring *2* is wrapped by a rigid ring-like cover *4*, attached to it fixedly, which in turn contacts with the wrapping flexible band. A vibrations-pressed gas film (VPGF) is produced by vibrations of the roller between the covering assembled ring-like piezoceramic element elastic frictional ring *3* and the fixed rigid axis *1*.

The vibrations of the rollers in a radial direction in the loaded and unloaded functioning modes were investigated (Fig 12).

In the loaded mode, the roller is wrapped by the flexible band at angles from $\pi/4$ to $3/2\ \pi$. One end of the band is fixed to the rigid basis, and a weight is fixed on the other end of the band. As is obvious from the data of the performed experiment, the amplitude and resonant frequency of the roller vibrations,



Sl. 11. *Vibrirajoča valja mehanizma RTM: a – pritrjeni valj (1); b – vrteči se valj (2); 1 – os, ki je trdno pritrjena na osnovo; 2 – piezokeramični obroč; 3 – elastični torni obroč; 4 – tesno prilegajoči se obročasti pokrov, ki ga ovija gibki jermen*

Fig. 11. *RTM vibrating rollers: a – fixed (1); b – rotating (2); 1 – axis, fixedly attached to the base; 2 – piezoceramic ring; 3 – elastic frictional ring; 4 – tight ring-like cover, wrapped by a flexible band*

resonančna frekvenca vibracij valja ovitega z jermenom pod enakima kotoma in z različno obremenitvijo, zmanjšata s povečanjem ravni bremena jermena (sl. 12 a).

Ko analiziramo odvisnost amplitude vibracij valja od električne moči z nespremenljivo resonančno frekvenco in različno stopnjo obremenitve jermena, lahko ugotovimo, da se amplituda vibracij valja zmanjša, če se obremenitev jermena poveča (sl. 12 b). Kadar je obremenitev jermena majhna, je odvisnost amplitude vibracij valja skoraj linearna. Če je dovedena napetost povečana, se poveča tudi amplituda vibracij.

Iz grafov (sl. 12 c in d), ki opisujeta odvisnost sprememb amplitude in frekvence vibracij valja od kota, pod katerim jermen ovija valj pri različnih ravneh obremenitve, lahko povzamemo, da se amplituda in frekvenca zmanjšata, če se kot ovijajočega jermena poveča.

Krivulje eksperimentalne raziskave valja št. *2* so prikazane na sliki 13.

Značilnosti frekvenčne amplitude vibracij valja št. *2*, ki se sestoji iz piezokeramičnega obroča piezokeramičnega elementa *2* z dejujočo (notranjo)

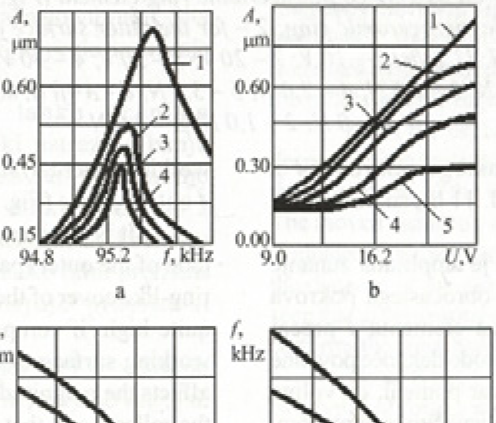
wrapped by a band at equal angles and with loads of different magnitude, diminish when increasing the level of load of the band (Fig.12 a).

When analyzing the dependence of the amplitude of the roller vibrations on the magnitude of the power-supply voltage, with a constant resonance frequency and a different level of load on the band, one can draw a conclusion, that if the load of the band is increased, the amplitude of the roller vibrations diminishes (Fig. 12 b). When the load on the band is small, the dependence of the amplitude of the roller vibrations is almost linear. If the power-supply voltage is increased, the amplitude of the roller vibrations increases as well.

From the graphs provided (Fig. 12 c and d), describing the dependence of the changes of amplitude and frequency of the roller vibrations on the angle of the roller wrapping band, with a different level of load applied, one may draw a conclusion that, if the wrapping angle of the band increases, the amplitude and frequency become reduced.

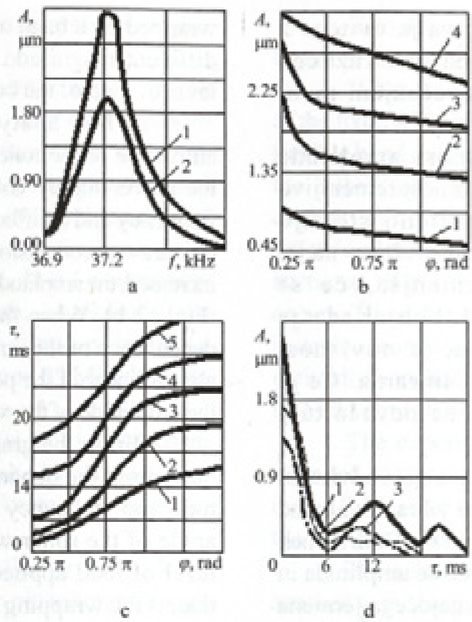
The curves of the experimental research of the RTM experimental roller No *2* are presented in Fig. 13.

The frequency amplitude vibration characteristics of the roller No *2*, consisting of the piezoceramic ring of the piezoceramic ring element *2*



Sl. 12. *Krivulje eksperimentalne raziskave vibrirajočega valja 1, ovitega s tesno prilegajočim se (obremenjenim) jermenom*

Fig. 12. *The curves of an experimental research of an RTM vibrating roller 1, wrapped by a tight (loaded) band: a - $A=f(f)$, ko/if $U = 40$ V; $\varphi=3/2\pi$; P je/is: 1 – 0 N; 2 – 0,5 N; 3 – 1,0 N; 4 – 2,0 N; b - $A=f(U)$, ko/if $\varphi=\pi$; P je/is: 1 – 0,5 N; 2 – 1,0 N; 3 – 1,5 N; 4 – 2,0 N; 5 – 2,5 N; c - $A=f(\varphi)$, ko/if $U = 40$ V; P je/is: 1 – 1,0 N; 2 – 2,0 N; d – $f=f(\varphi)$, ko/if $U = 40$ V; P je/is: 1 – 1,0 N; 2 – 2,0 N*



Sl. 13. *Krivulje eksperimentalne raziskave vibrirajočega valja 2, ovitega s tesno prilegajočim se (obremenjenim) jermenom: a - A = f(f) – velja za montirani piezokeramični obročasti element (sl. 2 b), ko je U = 60 V; 1 –velja za delovno površino piezokeramičnega obroča; 2 – velja za zunanjo površino togega obročastega pokrova 4*

Fig. 13. *The curves of an experimental research of RTM vibrating roller No 2, wrapped by a tight (loaded) band: a - A=f(f) for an assembled piezoceramic ring element (Fig. 2 b), when U = 60 V; 1 – for the working surface of the piezoceramic ring; 2 – for the outer surface of a rigid ring-like cover 4; b - A=f(φ), ko/if P = 1,0 N; U je/is: 1 – 10 V; 2 – 20 V; 3 – 30 V; 4 – 40 V; c - τ=f(φ), ko/if U = 30 V; P je/is: 1 – 0,5 N; 2 – 1,0 N; 3 – 1,5 N; 4 – 2,0 N; 5 – 3,0 N; d - A=f(τ), ko/if U = 40 V; φ=π; P je/is: 1 – 0 N; 2 – 1,0 N; 3 – 2,0 N*

površino in togega obročastega pokrova *4* z delujočo zunanjo površino (sl. 11 b), so podane na sliki 13 a.

Iz krivulj je očitno, da je amplituda zunanje (škodljive) frekvence togega obročastega pokrova piezokeramičnega obročastega elementa *4* precej velika, če jo primerjamo z amplitudo delujoče površine piezokeramičnega obroča *2*, kar pomeni, da vpliva na moč trenja med valjem *2* in ovijajočim se jermenom.

Slika 13 b kaže krivulje, ki opisujejo odvisnost amplitude montiranih piezokeramičnih obročastih elementov od kota, pod katerim gibki jermen ovija valj *2*, z različno električno močjo. Iz krivulj lahko razberemo, da se s spremembo kota ovijajočega se jermena od $\pi/4$ do $\pi/2$ amplituda frekvence znatno zmanjša. Če kot ovijajočega se jermena še nadalje povečamo do 3/2 π, se amplituda frekvence, v skladu z zakonom sorazmerja, nekoliko zmanjša. Ko povečamo dovedeno električno napetost, se amplituda frekvence zveča v primeru vseh kotov, pod

working (internal) surface and a rigid ring-like cover *4* outer surface (Fig. 11 b) are given in Fig. 13 a.

It is obvious from the curves that the amplitude of the outer (parasitical) frequency of the rigid ring-like cover of the piezoceramic ring element *4* is quite high, if compared with the amplitude of the working surface of the piezoceramic ring *2*, i.e., it affects the magnitude of the friction power between the roller *2* and that of the wrapping band.

Fig. 13 b displays curves describing the dependence of the amplitude of the assembled piezoceramic ring element on the angle of wrapping of the roller *2* by the flexible band with a different magnitude of power-supply voltage. One can see from the curves that as the wrapping angle changes from $\pi/4$ to $\pi/2$, the amplitude of the frequency significantly reduces. When continuing to increase the wrapping angle up to 3/2 π , the amplitude of the frequency reduces a little according to a linear law. When the power-supply voltage is increased, the amplitude of the frequency grows in all ranges of the angle of the katerimi jermen lahko ovija valj. Ko povečamo kot obremenjenega gibkega jermena, ki ovija valj *2*, se podaljša čas nastajanja VPGF.

Do podobnega sklepa pridemo tudi v primeru, ko povečamo obremenitev jermena in ko kot, pod katerim jermen ovija valj, ostane nespremenjen (sl. 13 c). Ko elektrodam piezokeramičnega obroča *2* ustavimo dovod energije, je trajanje dušenja VPGF odvisno od obremenitve jermena, ki ovija valj (sl. 13 d). Bolj ko je jermen obremenjen, hitreje je udušen VPGF.

band wrapping the roller. When increasing the angle of the loaded flexible band wrapping the roller *2*, the duration of the VPGF formation increases.

A similar conclusion can be drawn when the load of the band is increased and when the wrapping angle stays the same (Fig. 13 c). When the power supply, which is connected to the electrodes of the piezoceramic ring *2*, is switched off the duration of the damping of the vibrations-pressed gas film depends on the magnitude of the load of the band wrapping the roller (Fig. 13 d). The more the band is loaded, the more quickly the VPGF dampens.

5 ANALIZA POJAVOV NADZORA TRENJA MED ELEMENTI MEHANIZMA RTM

Vibrirajoče elemente mehanizma RTM smo raziskali s stojalom za preizkuse, ki je bil oblikovan posebej za te raziskave. Njegova funkcija temelji na nadzoru koeficienta trenja med elementi mehanizma RTM (sl. 14).

Sila trenja F_{tr} vpliva na deformacije vzmeti *2*, ko sila G_1 vpliva na valja *3*. Ogrodje *1* se giblje, to gibanje pa zaznata zaznavalo *8* in zapisovalna naprava *10*. Ko so visokofrekvenčni električni signali poslani v elektrode valjev *3*, ti začnejo vibrirati. Sila trenja se zmanjša za vrednost ΔF_{tr} in pride do drsenja med valjema in jermenom (sl. 14).

Na oscilogramu *1* lahko ocenimo zmanjšanje sile trenja ΔF_{tr}, ki ustreza gibanju ogrodja raziskovanega mehanizma RTM. Razdaljo premika *s* in čas gibanja *t* lahko ocenimo z oscilogramom *2*, ki ustreza razdalji, na kateri se gibljeta valja. Očitno je, da raziskovalni preizkus (sl. 14, oscilogram *1*) potrjuje pravilnost teoretičnega dela raziskave.

Analizirani so trije tipi valjev (sl. 15), od katerih dva tipa vključujeta vibrirajoče in pritrjene valje (sl. 15 a in b), tretji tip pa označuje vibrajoče in vrteče se valje (sl. 15 c).

Preizkuse smo izvedli z zveznim ter diskretnim napajanjem valjev z visokofrekvenčno napetostjo.

Preizkuse z zveznim signalom smo izvedli zato, da bi ugotovili odvisnosti sile trenja od amplitude vibracij, od dobe dovajanja električnih sunkov valju, od kota, pod katerim jermen ovija valj, od vlečne sile valja in od natezne sile jermena.

Preizkuse z diskretnim krmilnim signalom pa smo izvedli zato, da bi ugotovili odvisnosti

5 RESEARCH OF THE PROCESSES OF FRICTION CONTROLLING BETWEEN THE ELEMENTS OF THE RTM

The RTM with vibrating elements was investigated with an experiment stand [8], especially designed for these investigations. Its function is based on controlling the coefficient of friction between the elements of the RTM (Fig. 14).

The friction force F_{tr} affects the deformations of the springs *2* when the force G_1 influences the rollers *3*. The frame *1* moves, and this movement is registered by sensor *8* and the registration device *10*. When high-frequency electrical signals are sent to the electrodes of rollers *3*, they start to vibrate. The frictional force decreases by the magnitude ΔF_{tr}, and slipping between the rollers and the band appears (Fig.14).

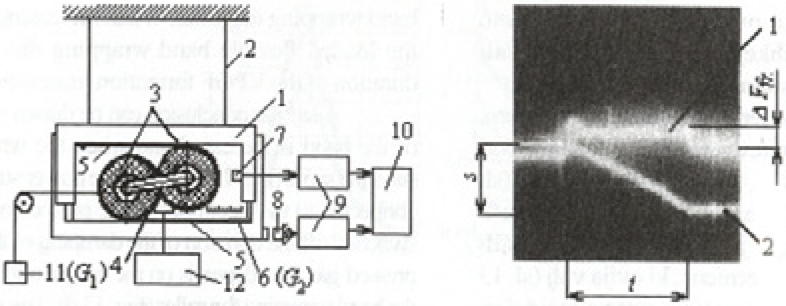
From the oscillogram *1* we can evaluate the decrease of the frictional force, ΔF_{tr}, which conforms to the movement of the frame of the investigated RTM. The moved distance *s* and the movement time *t* can be evaluated from the oscillogram *2*, which conforms to the distance of the movement of the rollers. It can be seen that experimental research (Fig. 14, oscillogram *1*) confirms the correctness of the theoretical investigation.

Three types of rollers are explored (Fig. 15): two types of rollers are vibrating and fixed (Fig. 15 a and b), the rollers of the third type are vibrating and rotating (Fig. 15 c).

The experiments were performed by feeding the rollers with a high-frequency voltage in continuous and start-stop modes.

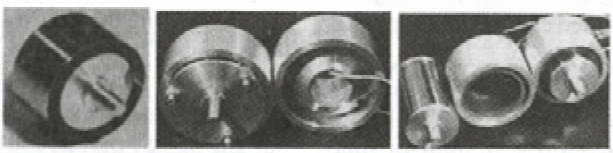
Experiments with the continuous signal mode were performed in order to find the frictional force dependencies on the amplitude of vibrations, on the period of feeding the electric impulses to the roller, on the wrapping angle of the roller by the band, on the pulling force of the roller, and on the force of stretching the band.

The start-stop experiments regulating the signal mode were carried out in order to find the dependencies of



Sl. 14. *Eksperimentalno ogrodje za raziskovanje mehanizma RTM z vibrirajočima valjema (1 – ogrodje; 2 – ploska vzmet; 3 – vibrirajoča valja; 4 – plošča; 5 – gibki jermen; 6 – utež (sila G_2); 7,8 – elementa za zaznavanje gibanja; 9 – ojačevalnik signala; 10 – zapisovalna enota; 11 – utež (sila G_1); 12 – visokofrekvenčni tok in značilni oscilogram gibanja ogrodja ter vibrirajočih valjev preučevanega mehanizma RTM*

Fig. 14. *The experimental stand for investigating RTM with vibrating rollers (1 – frame; 2 – flat spring; 3 – vibrating roller; 4 – plate; 5 – flexible band; 6 – weight (force G_2); 7,8 – motion-sensing element; 9 – signal amplifier; 10 – registering unit; 11 – weight (force G_1); 12 – high-frequency current and the characteristic oscillogram of movements of the frame and the vibrating rollers of the investigated RTM*



a b c

Sl. 15. *Vibrirajoči valji RTM: a - nepremični (1), glej sliko 11 a; b - nepremični (1) z valovnim vodilom; c - rotacijski (2), glej sliko 11 b*

Fig. 15. *RTM vibrating rollers: a – immovable (1), see Fig. 11 a; b – immovable (1) with a waveguide; c – rotary (2), see Fig. 11 b*

dolžine premika *s* (razdalje, za katero se valj premakne v enem obdobju krmilnega sunka) od dolžine sunka t_{imp}, - $s = f(t_{imp})$, od natezne sile jermena G_1, - $s = f(G_1)$, od vlečne sile valja G_2, - $s = f(G_2)$, in od kota, pod katerim jermen ovija valja $\alpha - s = f(\alpha)$.

Iz grafa (sl. 16 a, krivulja 1) je razvidno, da je ob daljšem sunku t_{imp} daljša tudi razdalja *s*, ki jo naredita valja v enem obdobju krmilnega sunka.

Sila trenja F_{tr} in čas *t* sta potrebna, da valja naredita razdaljo *s*, ko se vlečna sila jermena G_2 = konst. zmanjšuje, če na valja vpliva večja amplituda dovedene napetosti in ju vleče večja sila G_1 (sl. 16 b).

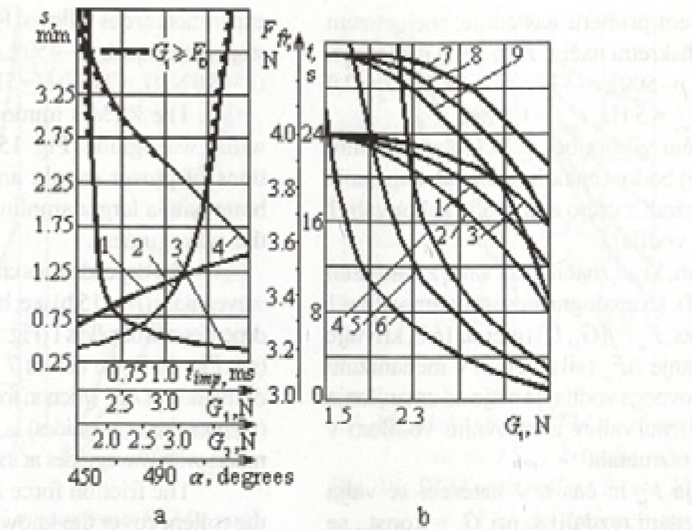
Del preizkusa smo izvedli tako, da smo vibrajočim valjem energijo dovajali zvezno pri α = 530°, G_2 = 2,0 N = konst.. Razdalja *s*, na kateri se premakneta valja v eni dobi krmilnega sunka, se poveča, kadar je povečana vlečna sila valja G_1. To je razvidno iz grafa (sl. 16 a, krivulja *2*). Če je $G_1 > F_0$ in je F_0 zanemarljiva vrednost sile lepenja, se v primeru, ko je ta vrednost presežena, se valja začneta gibati

the roller skip length *s* (the distance covered by the roller in one period of regulating impulse) on impulse duration t_{imp}, $-s = f(t_{imp})$, on the rollers' stretching force G_1, $-s = f(G_1)$, on the band's pulling force G_2, $-s = f(G_2)$, on the angle of the band wrapping the rollers $\alpha - s = f(\alpha)$.

As is clear from the graph (Fig. 16 a, curve 1), the longer is the duration of the impulse t_{imp}, the longer is the distance, *s*, that is covered by the rollers in one period of the regulating impulse.

The friction force F_{tr} and time, *t*, is needed for the rollers to cover the distance *s*, when the pulling force G_2 = *const* of the band is decreasing if the rollers are influenced by a larger amplitude of feeding voltage and they are pulled by a greater force G_1 (Fig. 16 b).

An experiment was made so that the vibrating rollers were fed by a continuous power-supply mode, when α = 530°, G_2 = 2.0 N = *const*. The distance *s* covered by the rollers in one period of regulating impulse increases, when increasing the force of the roller pulling G_1. This is clear from the graph (Fig. 16 a, curve *2*). If $G_1 > F_0$, where F_0 is the marginal value of the still friction force, so that when it is exceeded the rollers start moving



Sl. 16. Krivulje odvisnosti: *a – pomiki valja 1 v obdobju krmilnega sunka: 1 - $s = f(t_{imp})$; 2 - $s = f(G_1)$; 3 - $s = f(G_2)$; 4 - $s=f(\alpha)$; b - odvisnosti $F_{tr} = f(G_1)$ - 1 do 3; $t = f(G_1)$ - 4 do 6 (za mehanizem RTM z valji 1); $F_{tr} = f(G_1)$ - 7 do 9 (RTM z nepremičnimi valji z valovnim vodilom). Napajanje napetosti U je naslednje: 3, 4 in 7 - 10 V; 1, 5 in 8 - 20 V; 2, 6 in 9 - 30 V.*

Fig. 16. *Curves of dependencies: a – displacements of RTM roller 1 per one period of regulating impulse: 1 - $s=f(t_{imp})$; 2 - $s=f(G_1)$; 3 - $s=f(G_2)$; 4 - $s=f(\alpha)$; b – dependencies $F_{tr}=f(G_1)$ – 1 to 3; $t=f(G_1)$ – 4 to 6 (for Rolamite mechanisms with the rollers 1); $F_{tr}=f(G_1)$ – 7 to 9 (RTM with immovable rollers with waveguides). Feeding voltage U equals: 3, 4 and 7 – 10 V; 1, 5 and 8 – 20 V; 2, 6 and 9 – 30 V.*

brez stimulacije visokofrekvenčnih električnih signalov. Območje, v katerem je $G_1 \geq F_0$, je dodatno označeno s pikčasto črto (sl. 16 a). Dane razmere pri preizkusu so naslednje: mehanizem RTM deluje na diskretni način; $G_2 = 2{,}0$ N = konst.; $\alpha = 530°$; $f_r = 88{,}5$ kHz; $U = 51$ V; $f_{imp.sek} = 5$ Hz; $t_{imp} = 1{,}5$ ms; e - $G_1 = 3{,}0$ N; f - $G_1 = 3{,}5$ N; g - $G_1 = 3{,}6$ N; h - $G_1 = 3{,}7$ N.

Razdalja s, na kateri se premakneta valja v enem obdobju krmilnega sunka, se poveča, kadar se zmanjša natezna sila jermena (vlečna sila valja G_1 = konst.). To je razvidno iz grafa (sl. 16 a, krivulja 3).

Kadar je sila G_2 (nategnjeni jermen) zelo majhna, se valja začneta premikati brez stimulacije z visokofrekvenčnimi električnimi signali. P Razmere pri preizkusu so v tem primeru naslednje: mehanizem RTM deluje na diskretni način; $\alpha = 530°$; $f_r = 88{,}5$ kHz; $U = 51$ V; $G_1 = 3{,}0$ N = konst.; $f_{imp.sek} = 5$ Hz; $t_{imp} = 1{,}5$ ms; G_2: j - 4,0 N; k - 2,0 N; l - 1,7 N; m - 1,5 N.

Razdalja s, na kateri se premakneta valja v enem obdobju krmilnega sunka, se zmanjša, kadar ta poveča kot, pod katerim jermen ovije valja. To je razvidno iz grafikona (sl. 16 a, krivulja 4).

Ko je kot zavoja jermena zelo oster, $G_1 \geq F_0$, se valja začneta premikati brez stimulacije z visokofrekvenčnimi električnimi signali. Razmere pri

without being influenced by the high-frequency electrical signals. The zone in which $G_1 \geq F_0$ is additionally marked by a dotted line (Fig. 16 a). The conditions of the experiment are given as follows: RTM operating mode is start-stop; $G_2 = 2.0$ N = *const*; $\alpha = 530°$; $f_r = 88.5$ kHz; $U = 51$ V; $f_{imp.sek} = 5$ Hz; $t_{imp} = 1.5$ ms; $e – G_1 = 3.0$ N; $f – G_1 = 3.5$ N; $g – G_1 = 3.6$ N; $h – G_1 = 3.7$ N.

The distance s covered by the rollers in one period of the regulated impulse increases when this is decreasing the stretching force of band (the pulling force of roller G_1 = *const*). This is clear from the graph (Fig. 16 a, curve 3).

When the force G_2 (stretching the band) is very low, the rollers start moving without being influenced by the high-frequency electrical signals. The conditions of the experiment are as follows: RTM operating mode is start-stop; $\alpha = 530°$; $f_r = 88.5$ kHz; $U = 51$ V; $G_1 = 3.0$ N = *const*; $f_{imp.sek} = 5$ Hz; $t_{imp} = 1.5$ ms; G_2: $j – 4.0$ N; $k – 2.0$ N; $l – 1.7$ N; $m – 1.5$ N.

The distance s covered by the rollers in one period of the regulated impulse is decreasing when this is increasing the angle of the rollers' wrapping by the band. This is clear from the graphs (Fig. 16 a, curve 4).

When the wrapping angle is very low, $G_1 \geq F_0$, the rollers start moving without being influenced by the high-frequency electrical signals. The conditions of the preizkusu so v tem primeru naslednje: mehanizem RTM deluje na diskretni način; za α velja naslednje: n - 450°; o - 480°; p - 500°; r - 530°; G_1 = 3,0 N; G_2 = 2,0 N; U = 51 V; $f_{imp.sek}$ = 5 Hz; t_{imp} = 1,5 ms.

Nepremični vibrirajoči valji (1) z valovnim vodilom (sl. 15 b) bodo v enakih razmerah napajanja in vzbujanja, vibrirali z večjo amplitudo kakor valji 1 (brez valovnega vodila).

Odvisnosti, ki so značilne za valje z valovnim vodilom (sl. 15 b), so analogne odvisnostim valjev 1 (sl. 15 a), na primer, $F_{tr} = f(G_1, U)$ (glej sl. 16 b, krivulje 7 do 9). Zmanjšanje ΔF_{tr} (sila trenja) v mehanizmu valjev 1 (brez valovnega vodila) je nižje od zmanjšanja te sile v mehanizmu valjev z valovnim vodilom v enakih delovnih razmerah.

Sila trenja F_{tr} in čas t, v katerem se valja premakneta po znani razdalji s, pri G_1 = konst., se zmanjšata, če valjema povečamo amplitudo napetosti in zmanjšamo natezno silo G_2. To je razvidno iz grafov (sl. 17 a in b). Preizkus je bil izveden ob zveznem napajanju vibrirajočih valjev pri α = 530°.

Sila trenja F_{tr} in čas t, v katerem se valja premakneta po razdalji s, pri $G_1 = G_2$ = konst., se povečujeta, ko se povečuje kot jermenovega zavoja α (sl. 17 c). Ta del preizkusa je bil izveden ob zveznem napajanju vibrirajočih valjev 1.

Raba vrtečih se valjev 2 ima pri mehanizmu RTM poseben pomen (glej sl. 15 c).

Iz rezultatov eksperimentalne raziskave, prikazanih na sliki 18, je razvidno, da se sila trenja F_{tr} zmanjša, če: a) valjem dovajamo napetost z večjo amplitudo U, b) valja izpostavimo večji vlečni sili G_1 (sl. 18 a), c) zmanjšamo natezno silo jermena G_2 (sl. 18 b), d) zmanjšamo kot α, pod katerim jermen ovija valja (sl. 18 c).

Kakor je razvidno iz navedenih rezultatov preizkusov, ima mehanizem RTM z vrtečima in vibrirajočima valjema 2, v primerjavi z drugimi tipi, boljšo ležajno zmogljivost in je bolj občutljiv. Primerno ga je uporabljati v sklopu izjemno občutljivih sistemov. Od vibracij vsiljena plinasta plast se pri valjih 2 pojavi zelo hitro, ker je uporabljena celotna delovna površina piezokeramičnega obroča, kar pa, zaradi konstrukcije samega mehanizma RTM, ni mogoče v primeru mehanizma z valjema drugačnega tipa.

Še več, v mehanizmu RTM z vibrirajočima nepremičnima valjema (sl. 15 a in b), ki ju ovija gibki jermen, je ovira nastanku plasti vibracij med vibrirajočim valjem in jermenom v tem, da je gibki jermen relativno tanek in se hitro deformira.

experiment are as follows: RTM operating mode is start-stop; α are equal: $n - 450°$; $o - 480°$; $p - 500°$; $r - 530°$; G_1 = 3.0 N; G_2 = 2.0 N; U = 51 V; $f_{imp.sek}$ = 5 Hz; t_{imp} = 1.5 ms.

The RTM's immovable (1) vibrating rollers with a waveguide (Fig. 15 b), given the same conditions of power supply and of impact on them, vibrate with a larger amplitude than rollers 1 (without the waveguides).

The dependencies characteristic of the rollers with waveguides (Fig. 15 b) are, by their type, in analogy to the dependencies of rollers 1 (Fig. 15 a), for example, $F_{tr} = f(G_1, U)$ (see Fig. 16 b, the curves 7 to 9). The magnitude of the decreasing of ΔF_{tr} (friction force) in the RTM with rollers 1 (without the waveguides) is lower than in the RTM with rollers with waveguides at the same working conditions.

The friction force F_{tr} and the time t, in which the rollers cover the known distance s, whereas G_1 = *const*, decreases, if the rollers are provided with increased amplitude voltage and lowered band-stretching force G_2. This is evident from the graphs (Fig. 17 a and b). The experiment is carried out in continuous power-supply mode of the vibrating rollers, α = 530°.

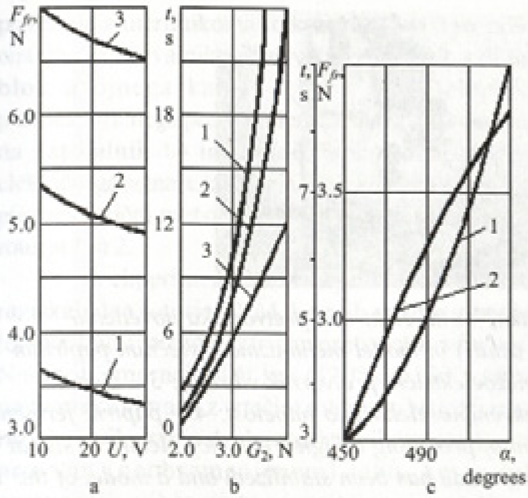
The frictional force F_{tr} and the time t, in which the rollers cover the distance s, when $G_1 = G_2$ = *const*, is increasing when the angle of the band wrapping the rollers α is increasing (Fig. 17 c). The experiment is carried out in the continuous power-supply mode of the vibrating rollers 1.

The application of the rotating rollers 2 in RTM is of a special interest (see Fig. 15 c).

From the results of the experimental research provided in Fig. 18 it is clear that the friction force F_{tr} is decreased if: a) the rollers are supplied with a higher amplitude power-supply voltage U, b) the rollers are influenced by a larger pulling force G_1 (Fig. 18 a), c) the stretching force G_2 of the band decreased (Fig. 18 b), d) the decreased angle α of the band wrapping the rollers (Fig. 18 c).

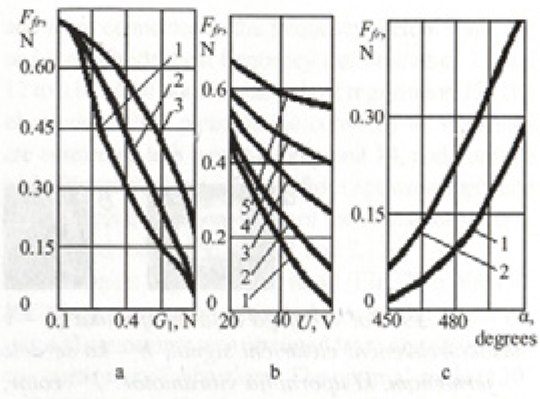
As one can see from the results of the experiments, the RTM with the rotating vibrating rollers 2, compared with the other, has a better bearing capacity and is much more sensitive. It is advisable to use them in exceptionally sensitive systems. The vibrations-pressed gas film appears in the rollers 2 very quickly, because all the working surface of the piezoceramic ring is used, which is impossible in the RTM with rollers of another type, because of the design of the RTM itself.

Moreover, in the RTM with vibrating immovable rollers (Fig. 15 a and b), wrapped by a flexible band, the obstacle of the formation of FV between the vibrating roller and the band is that the flexible band is comparably thin and quickly deforms.



Sl. 17. *Krivulje odvisnosti mehanizma RTM z valji*

Fig. 17. *Dependency curves of RTM with rollers 1: a – $F_{tr} = f(U)$, ko/if $G_1 = 3{,}5$ N = const; G_2 je/is: 1 – 2,0 N; 2 – 3,0 N; 3 – 4,0 N; b – $t = f(G_2)$, ko/if $G_1 = 3{,}5$ N = const; U je/is: 1 – 10V; 2 – 20 V; 3 – 30 V; c – 1 – $F_{tr} = f(\alpha)$; 2 – $t = f(\alpha)$, ko/if U = 40V; $G_1 = G_2 = 2{,}0$ N*



Sl. 18. *Mehanizem RTM z valjema 2 (glej sl. 11 b in sl. 15 c) krivulje odvisnosti*

Fig. 18. *RTM with rollers 2 (see Fig. 11 b and Fig. 15 c) dependency curves a – $F_{tr} = f(G_1)$, ko/if $\alpha = 500°$; $G_2 = 5{,}0$ N; U je/is: 1 – 50 V; 2 – 35 V; 3 – 20 V; b – $F_{tr} = f(U)$, ko/if $G_1 = 0{,}5$ N; $\alpha = 500°$; G_2 je/is: 1 – 4,0 N; 2 – 5,0 N; 3 – 7,0 N; 4 – 8,0 N; 5 – 9,0 N; c – $F_{tr} = f(\alpha)$; $G_1 = 0{,}5$ N; $G_2 = 5{,}0$ N; U je/is: 1 – 50 V; 2 – 30 V*

Krivulje iz teoretičnega dela raziskave smo porabili za oblikovanje vibramotorjev Rolamite.

6 ANALIZA DINAMIČNIH PROCESOV V VIBRAMOTORJIH ROLAMITE

Prvi vibramotorji so izkoriščali diagonalno udarjanje piezokeramičnega pretvornika vibracij (piezokeramični elementi) ob gibljiv element, tj. ploščo ali valj [7].

Ko na elektrode pretvornika dovedemo visoko frekvenčni električni signal, se končni del piezoelektričnega pretvornika, ki ima netogi stik z rotorjem, prične eliptično gibati in obračati rotor (sl. 19).

Določili smo najboljši kot piezokeramičnega pretvornika vibracij 2, ki udarjajo ob rotor 1, pri katerem je zagotovljeno največje število vrtljajev rotorja 1. Pri raziskovani konstrukciji vibramotorja je najboljša vrednost kota α 125°.

Ob konstruiranju vrtilnega vibramotorja Rolamite ter mehanizmov vlečnega jermena je bilo treba ugotoviti, kakšen vpliv ima na sinhronost vrtenja valjev uporaba načela Rolamite v mehanizmih vlečnega jermena.

Model mehanizma prenosa gibanja je bil oblikovan kot vrtilni vibramotor Rolamite (sl. 20 a), na podlagi tega pa je bilo oblikovano eksperimentalno stojalo (sl. 20 b).

The curves from the theoretical research were used when designing the Rolamite vibromotors.

6 RESEARCH OF THE DYNAMIC PROCESSES RUNNING IN THE ROLAMITE VIBROMOTORS

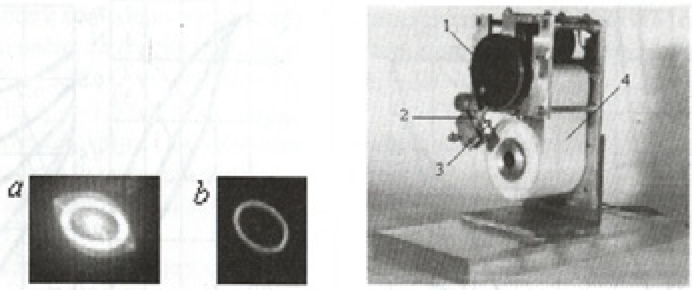
The first vibromotors used diagonal hits by the piezoelectric converter of the vibrations (piezoceramic elements) onto a movable element, i.e., a plate or a roller [7].

After providing a high-frequency electrical signal on the electrodes of the converter, the end of the piezoelectric converter, elastically connected to the rotor, starts making elliptical movements and turns the rotor (Fig. 19).

The optimal angle of the piezoelectric converter of the vibrations 2 hitting the rotor 1, at which the maximum rotations of rotor 1 are ensured, was found. The optimal value of the angle α in the explored design of the vibromotor is 125°.

When designing the rotary RVM and the mechanisms of band pulling, it was necessary to find what impact the applying of the Rolamite principle in the mechanisms of band pulling has on the synchronicity of the rotation of the rollers.

The model of a mechanism of movement transmission was designed as a rotary Rolamite vibromotor (Fig. 20 a), and based on that an experimental stand was designed (Fig. 20 b).



Sl. 19. *Pot vrhnjega dela menjalnika (a – v trenutku, ko na elektrode pretvornika dovedemo visokofrekvenčni električni signal; b – ko se delovanje ustali) in model mehanizma z vlečnim papirnim jermenom, ki uporablja vibramotor: 1 – rotor; 2 – piezoelektrični pretvornik vibracij; 3 – žica, ki piezoelektričnemu pretvorniku vibracij dovaja visokofrekvenčno električno napetost; 4 – papirni jermen*

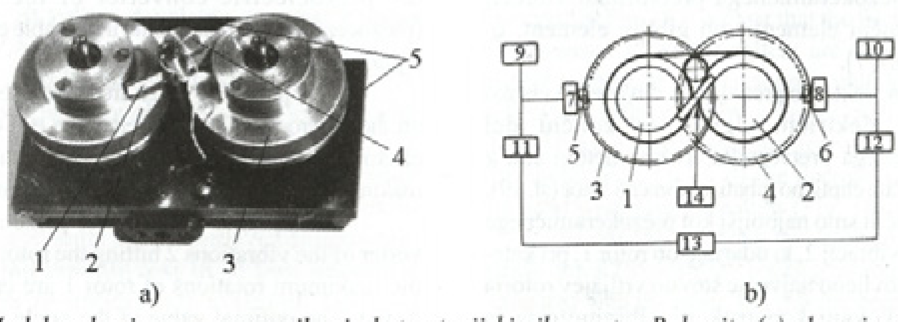
Fig. 19. *Path of the tip of the changer (a – at the moment of providing a high-frequency electrical signal on the electrodes of the converter; b – when the working mode has been stabilized) and a model of the mechanism of paper's band pulling, using a vibromotor: 1 – rotor; 2 – piezoelectric converter of vibrations; 3 – power supply wire of piezoelectric converter that provides a high-frequency voltage; 4 – the paper band*

Pogonsko vozlišče sestoji iz piezoelektričnega pretvornika vibracij 1, ki ustvarja vrtenje valjev - rotorjev 2 in 3. Valja -rotorja 2 in 3, kakor tudi tesno prilegajoči se valj manjšega premera 4, so oviti z neskončnim jermenom 5. Valja 2 in 3 sta ovita z neskončnim jermenom 5 pod kotom, ki je večji od 270°, valj 4 pa je ovit pod kotom, večjim od 180°. Shema meritev je predstavljena na sliki 20 b.

Na gredi rotorjev 1 in 2 sta pritrjena dva enaka rastrska diska 3 in 4 z znakom, ki se odziva na svetlobni zaznavali 5 in 6. Predojačevalnika 7 in 8 sta

The drive node consists of a piezoelectric converter of the vibrations 1, which is providing a rotation of the rollers-rotors 2 and 3. The rollers-rotors 2 and 3, and also a roller of a smaller diameter 4 that is tight, are wrapped by an endless band 5. The rollers 2 and 3 are wrapped by an endless band 5 by an angle of more than 270°, and a roller 4 by an angle of more than 180°. The scheme of measurement is presented in Fig. 20 b.

On the shaft of rotors 1 and 2 two identical raster disks 3 and 4 with a raster mark that interacts with the photo sensors 5 and 6 are fixed. The former-amplifiers 7



Sl. 20. *Model mehanizma prenosa gibanja kot rotacijski vibramotor Rolamite (a): 1 – piezoelektrični pretvornik vibracij; 2 in 3 – valj - rotor;4 – valj manjšega premera; 5 – neskončni jermen in shema eksperimentalnega stojala (b): 1 in 2 - rotorja; 3 in 4 – rastrska diska; 5 in 6 – svetlobni zaznavali; 7 in 8 – predojačevalnika; 9 in 10 – merilnika frekvence; 11 in 12 – razločevalnika frekvence; 13 – blok dvojnega kanala registracije; 14 – napajalnik piezoelektričnega pretvornika vibracij*

Fig. 20. *The model of a mechanism of movement transmission as a rotary Rolamite vibromotor (a): 1 – piezoelectric converter of vibrations; 2 and 3 – roller-rotor; 4 – roller of smaller diameter; 5 – endless band and scheme of experimental stand (b): 1 and 2 - rotors; 3 and 4 - raster disks; 5 and 6 - photosensors; 7 and 8 - former-amplifiers; 9 and 10 - frequency meters; 11 and 12 - frequency discriminators; 13 - block of double channels of registration; 14 – power-supply unit of piezoelectric converter of vibrations*

povezana z merilnikoma frekvence 9 in 10 ter prek enakih razločevalnikov frekvence 11 in 12 tudi na blok dvojnega kanala zapisa 13. Elektrode piezoelektričnega pretvornika vibracij so priključene na napajalnik 14 in zaradi visokofrekvenčnega električnega signala, ki prihaja iz napajalnika 14, začne piezoelektrični pretvornik vibracij vibrirati in vrteti rotorja 1 in 2.

Na ekperimentalnem stojalu (sl. 20 b) smo raziskali dva rotorja RVM, katerih vrtilno gibanje je spodbudil piezoelektrični pretvornik vibracij. Nastavili smo najboljši kot (120° do 130°) stika pretvornika vibracij z vrtečim rotorjem. Najprej smo raziskali vibriranje dveh rotorjev vibramotorja pri prostem (neobremenjenem) teku, ko je bil pretvornik vibracij togo pritrjen na sredino vibramotorja.

Ko se amplituda dovedene električne napetosti U zveča, se poveča tudi število rotorjevih vrtljajev n, vendar pa nenatančno pritrjen piezoelektrični pretvornik povzroči nesinhrono vrtenje rotorjev (sl. 21, krivulji *1* in *2*). Če sta rotorja vibramotorja ovita z neskončnim jermenom v obliki mehanizma RTM in se oba rotorja vrtita, nesinhronost vrtenja dveh rotorjev izgine, kar pomeni, da Δn=0 (sl. 21, krivulja *3*).

Ko se poveča zunanja obremenitev enega od rotorjev, se močno poveča tudi nesinhronost vrtenja obeh rotorjev (sl. 22, krivulja *1*).

Povezava dveh rotorjev z neskončnim jermenom v obliki mehanizma RTM zmanjša nesinhronost njunega vrtenja na najmanjšo vrednost (sl. 22, krivulja *3*). S slike 22 tudi izhaja, da je nesinhronost vrtenja rotorjev vibramotorja Rolamite, ki sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki vrti oba rotorja, dosti manjša od nesinhronosti vrtenja rotorjev vibramotorja Rolamite, ki sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki vrti le enega od rotorjev (sl. 22, krivulja 2).

Iz navedenega je mogoče sklepati, da največjo sinhronost vrtenja rotorjev zagotavlja vibramotor Rolamite, ki se sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki hkrati vrti oba rotorja.

Pomanjkljivost vibramotorja, ki uporablja diagonalno udarjanje piezo električnega pretvornika vibracij (piezokeramični elementi) ob rotor, je njegova nezmožnost spremeniti smer vrtenja rotorja.

and 8 are connected to the frequency meters 9 and 10, and through identical frequency discriminators 11 and 12 to a block of double channels of registration 13. The electrodes of the piezoelectric converter of vibrations are connected to a power-supply unit 14, and during a high-frequency electrical signal from a power-supply unit 14, the piezoelectric converter of the vibrations starts to vibrate and rotates the rotors 1 and 2.

On an experimental stand (Fig. 20 b) the two RVM rotors were investigated, where the rotors rotational movement was obtained from one piezoelectric converter of vibrations. The optimal angle (120° to 130°) of contact for the converter of vibrations with a rotor rotated by it was used. First, the vibration mode of the two rotors, with the vibration's converter rigidly fixed to the middle of it, is investigated in the empty (unloaded) working mode.

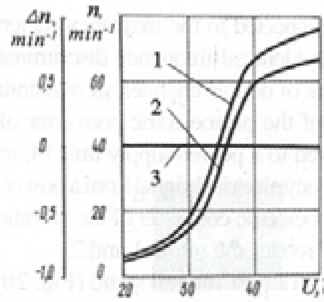
When the amplitude of the feeding voltage U is increased, then the number of turns n of rotors increases; however, inexact fastening of the piezoelectric converter leads to non-synchronous rotations of the rotors (Fig. 21, curves *1* and *2*). If the two VM rotors are wrapped by an endless band in the form of RTM and both rotors are rotating, the nonsynchronicity of the rotors' rotations disappears, i.e., Δn=0 (Fig. 21, the curve 3).

When the external loading incereases on one of the rotors, the nonsynchronicity of the rotations of both rotors sharply increases (Fig. 22, the curve *1*).

The connection of the two rotors by an endless band in the form of an RTM reduces to a minimum the nonsynchronicity of their rotations (Fig. 22, the curve *3*). From Fig. 22 it also follows that the magnitude of the nonsynchronicity of the rotation of the rotors of the Rolamite vibromotor of two rotors with a piezoelectric converter of vibrations, which rotates both rotors, is much less than the magnitude of the nonsynchronicity of the rotation of the rotors of the Rolamite vibromotor of two rotors with a piezoelectric converter of vibrations, which rotates only one rotor (Fig. 22, the curve 2).

From this it is necessary to draw a conclusion that the highest synchronicity of rotation of the rotors is provided by the Rolamite vibromotor of two rotors with a piezoelectric converter of vibrations that simultaneously rotates both rotors.

A weakness of the vibromotor that used diagonal hits of the piezoelectric converter of vibrations (piezoceramic elements) onto a rotor is the impossibility of reversing the rotation of a rotor.



Sl. 21. *Krivulji odvisnosti za nesinhrono vrtenje rotorjev vibramotorja pri prostem (neobremenjenem) teku: krivulji 1 in 2 - ko je n=f(U) pri dveh rotorjih, ki nista povezana z neskončnim jermenom in čigar vrtenje ustvarja splošni piezoelektrični pretvornik; krivulja 3 - ko je Δn=f(U) pri vibramotorju dveh rotorjev s splošnim piezoelektričnim pretvornikom vibracij potrebnim za vrtenje obeh rotorjev*

Fig. 21. *Dependency curves for non-synchronous rotations of rotors VM in the empty (unloaded) mode of working: curves 1 and 2 – when n=f(U) for two rotors unconnected by an endless band and getting the rotation from the general piezoelectric converter of vibrations; the curve 3 – when Δn=f(U) for RVM of two rotors with general piezoelectric converter of vibrations for rotating both rotors*

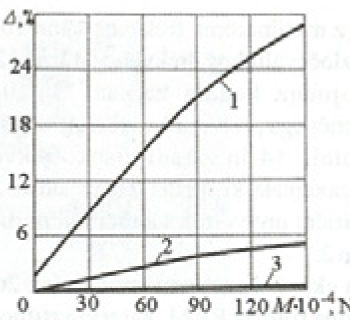
7 SKLEPNE OPOMBE

S teoretičnim delom raziskave smo dokazali naslednje:

1. Povečanje koeficienta suhega trenja omogoči hitrejše oblikovanje nezdrsnega območja med telesi mehanizma RTM.
2. Parametri vibracij mehanizma RTM se spremenijo takoj, ko elementi sistema RTM dosežejo nezdrsno območje, kar pomeni, da lahko na nezdrsno področje vplivamo z vibracijskim vzbujanjem.
3. Če povečamo amplitudo vibracij, se sila trenja med telesi sistema zmanjša; če se poveča zdrsno področje med telesi sistema, se zmanjša izkoristek.

Z eksperimentalnim delom raziskave smo dokazali naslednje:

1. Povečanje amplitude električne napetosti, zmanjšanje sile obremenitve gibkega jermena in



Sl. 22. *Krivulje odvisnosti vrtilne nesinhronosti rotorjev od moči zunanje obremenitve enega od rotorjev Δ = f(M), ko je U = konst. = 50 V: 1 – za primer dveh rotorjev, ki nista povezana z neskončnim jermenom in ju vrti splošni piezoelektrični pretvornik vibracij; 2 – za primer vibramotorja Rolamite, ki se sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki vrti le en rotor; 3 – za primer vibramotorja Rolamite, ki se sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki vrti oba rotorja*

Fig. 22. *Curves of the dependencies of the rotations nonsynchronicity of the rotors on the magnitude of the external loading on one of the rotors Δ=f(M) when U=const=50V: 1 - for two rotors not connected by an endless band and both getting rotation from the general piezoelectric converter of vibrations; 2 – for the Rolamite vibromotor of two rotors with a piezoelectric converter of vibrations, which rotates only one rotor; 3 - for the Rolamite vibromotor of two rotors with a piezoelectric converter of vibrations that rotates both rotors*

7 CONCLUDING REMARKS

Theoretical research proved:

1. The increasing magnitude of the dry-friction coefficient makes it faster to achieve a non-slipping zone between the bodies of the "roller-band" system of the RTM.
2. From the "roller-band" system of the RTM the vibration parameters depend as quickly as bodies of the system achieve a non-slipping zone, and this means that the non-slipping zone can be operated by the excitation of vibrations.
3. If the amplitude of vibrations is increasing, then the force of friction between the bodies of the system is decreasing; if the zone of slipping between the bodies of the system is increasing, the magnitude of the efficiency is decreasing.

Experimental research proved:

1. The increase of the supply-voltage amplitude, the decrease of the force load magnitude of the flexible

zmanjšanje kota, pod katerim gibki jermen ovija valj, povzročijo povečanje amplitude vibracij valja.

2. Mehanizem RTM z vrtečima se, vibrirajočima valjema, ima v primerjavi z vibrirajočima valjema drugega tipa boljšo ležajno zmogljivost in je bolj občutljiv. Zato je priporočljivo, da ga uporabljamo v izredno občutljivih sistemih.
3. Če primerjamo vibramotor Rolamite, ki se sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki sočasno vrti oba rotorja, ter vibramotor Rolamite, ki se sestoji iz dveh rotorjev in piezoelektričnega pretvornika vibracij, ki vrti le enega od rotorjev, prva možnost omogoči večjo sinhronost vrtenja rotorjev.

band and the decrease of the angle wrapping of the roller by the flexible band cause the increase in the amplitude of the roller vibrations.

2. The RTM with rotating vibrating rollers, compared with vibrating rollers of another type, have a better bearing capacity and are much more sensitive. Therefore, it is advisable to use them in exceptionally sensitive systems.
3. If we compare the Rolamite vibromotor of two rotors with the piezoelectric converter of vibrations, which simultaneously rotates both rotors, and the Rolamite vibromotor of two rotors with a piezoelectric converter of vibrations, which rotates only one rotor, the first variant provides the highest synchronicity of the rotation of rotors.

8 VIRI
8 REFERENCES

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