BRANIMIR PAVIĆ, D.Sc.
Fakultet strojarstva i brodogradnje
I. Lučića 5, Zagreb
JERKO RADOŠ, D.Sc.
MARIO ŠAFRAN, B.Eng.,
Fakultet prometnih znanosti
Vukelićeva 4, Zagreb

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DURABILITY OF UNIVERSAL JOINT ELEMENTS DEPENDING ON DAMAGES IN THE PHASE OF TRANSITIONAL OCCURRENCES

ABSTRACT

The paper considers the influence of the relationship between the dynamic behaviour of the universal joint and the damage in the period of transitional occurrences. Based on the theoretical study of the shaft system with universal joints, the classification, arrangement and value of dynamic loads have been determined, and the dynamic model described by differential equations has been defined. Laboratory testing have been directed towards checking the safety and durability compared to the structural requirements, precision and quality of production, as well as mechanical and thermal treatment for a characteristic type of a universal joint.

KEY WORDS

universal joint elements, transitional occurrences

1. INTRODUCTION

In order to determine the magnitudes of dynamic loads in drive mechanisms with universal joints, it is necessary to determine differential equations of motion taking into consideration the values of transmission and flexible elements. Describing of dynamic occurrences in such drives is much simplified by the use of Mathieu's differential equations which are derived from the Lagrangian equation of motion of the second order. The friction forces (which act as stabilisers) are neglected as well as the rigidity of universal joints which is in practice much lower compared to the shafts.

During operation a universal joint must function properly over a certain period of time, i.e. it must meet its application requirements. In order to determine its optimal performance, its basic characteristics need to be analysed compared to the specific ones caused by the changes in working conditions. In the process of such comparison, the limitations need to be considered regarding torque as well as the safety in the operation of the spherical mechanism. These limitations are determined by the analysis of exploitation or laboratory testing i.e. by calculation methods using appropriate mathematical models. The best results are obtained from the comparison of calculation parameters and values obtained by laboratory or exploitation testing.

Calculating the damaging process of the universal joint bearings, and especially the cross journals (cross joints) indicates the possibility of improving the durability of the complete joint mechanism. The most economical device for laboratory testing of the four specimens (joints) simultaneously, which operate simultaneously in a closed circle was used also for this testing at the Laboratory for Machine Elements, FSB - Zagreb.

2. DETERMINING THE MAGNITUDE OF CHANGES OF DYNAMIC MOMENT AND TORQUE ON THE ELEMENTS OF THE TESTING SYSTEM OF UNIVERSAL JOINTS

Dynamic changes in the given testing conditions are determined by Mathieu's equation of motion

$$\frac{d^2 y}{dz^2} + (a - 2q\cos 2z) y =$$

$$= \frac{M_3}{I_3 \omega_1^2} - \frac{M_1}{I_1 \omega_1^2} + a\sqrt{A^2 + B^2} \sin 2(\varphi_1 + \delta_e)$$

for the presented dynamic system according to Figure 1.

$$A = \frac{\delta^2}{2} = 0.00746312 \qquad B = 0 \qquad \mathrm{tg}\, 2\delta_e = 0$$

$\delta_e = 0$ angle of drive unit disturbance
$D = 0.000003482$

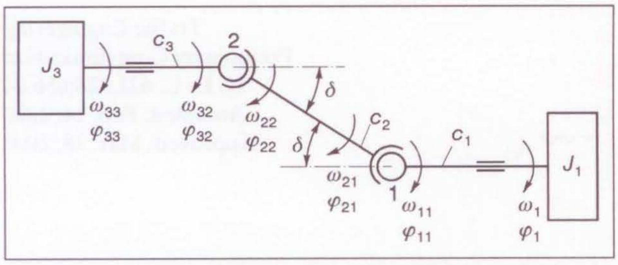

Figure 1 - considered part of the mechanism for testing of universal joints- schematically presented

$$Z=\varphi_1+\delta_e=\varphi_1 \qquad a=\frac{c}{I_1\omega_1^2}=51.597 \qquad 2q=1.71$$

$I_3=0.01873\,[\mathrm{Nms^2}]$ $I_0=0.008895\,[\mathrm{Nms^2}]$	moment of inertia of single members i.e. system as a whole
$n_1=1400\left[\frac{\mathrm{o}}{\mathrm{min}}\right]$ $(\omega=146.607\,[s^{-1}])$	number of revolutions i.e. angular velocity of testing
$M_1=60\,[\mathrm{Nm}]$ $M_2=60\,[\mathrm{Nm}]$	set pre-stress moment
$\delta=7^\circ\,(0.122173[\,\mathrm{rad}])$	set angle of universal joints installation
$C_{11}=C_{12}=C_{13}=$ $=0.00373156\,[\mathrm{Nm/rad}]$	rigidity coefficient of the drive unit members
$C=18742.452\,[\mathrm{Nm/rad}]$	drive unit rigidity coefficient
$I_1=0.01694\,[\mathrm{Nms^2}]$	
$I_2=0$	negligible regarding the system disturbance

By introducing the system parameters, the differential equation of the difference of the angles of fluctuation in joint 2 is obtained;

$$\frac{d^2(\varphi_2-\varphi_1)}{d\varphi_1^2}+(51.94896-1.71\cos\varphi_1)(\varphi_2-\varphi_1)=$$
$$=-0.16479+0.38062\sin 2\varphi_1$$

i.e. for the case that mass is not bonded further in the system of differences of angles of shaft fluctuations at the input and the output, with changes of values and magnitudes

$$a=97.94896$$
$$2q=1.73688$$

$$\frac{d^2(\varphi_3-\varphi_1)}{d\varphi_1^2}+(97.94896-1.73688\cos\varphi_1)(\varphi_3-\varphi_1)=$$
$$=0.01575+0.731\sin 2(\varphi_1+0.12217)$$

If we use $\varphi_1=\omega_1 t$ after introducing into the above expressions, with application of double integration carried out by a computer, graphical presentation (plotter) of the change in angular velocities is also obtained.

From the expression regarding change of moment and by using the developed Mathieu's differential equation, after rewriting it and neglecting the small values, in the similar way the following is obtained:

$$M_3=-3.0109+139.7583\sin 2\varphi_1$$

that is,

$$M_2=-60+15.778\sin 2\varphi_1$$

which is presented in the enclosed graphs.

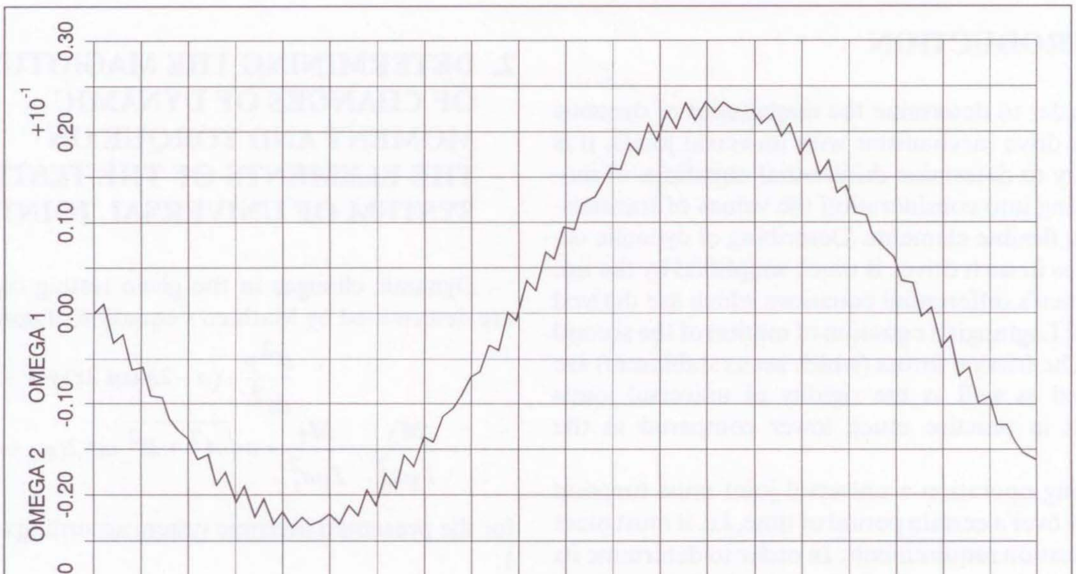

Figure 2 - Functional dependence of the difference of angular velocities ω on the change in time t

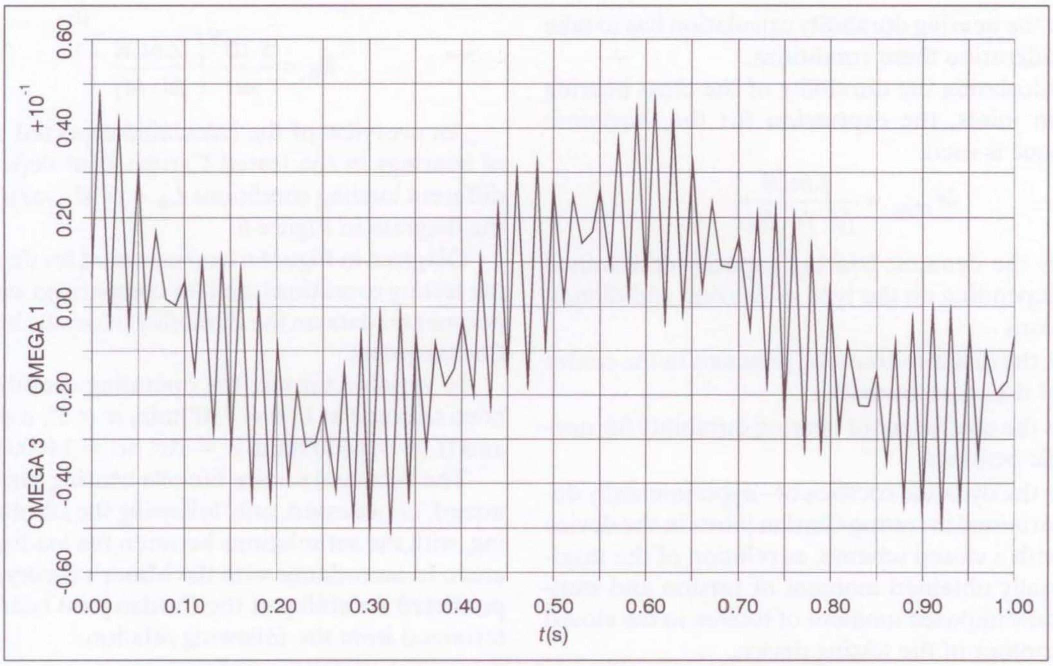

Figure 3 - Differences of angular velocities ω as function of time t

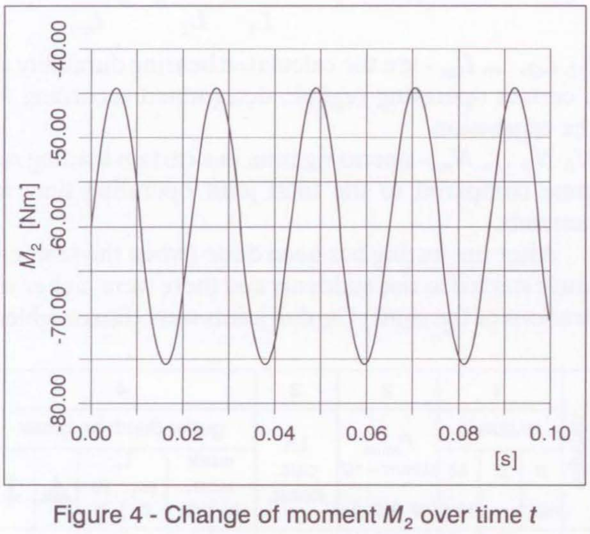

Figure 4 - Change of moment M_2 over time t

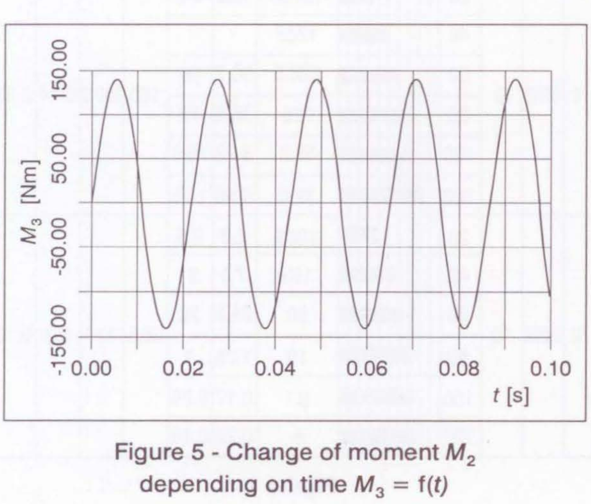

Figure 5 - Change of moment M_2 depending on time M_3 = f(t)

3. ANALYTICAL METHOD OF DETERMINING THE DURABILITY OF UNIVERSAL JOINT BEARINGS

In the literature, mainly by foreign authors, several mathematical models have been suggested for determining the durability of universal joints, which have been used for comparison analysis with laboratory testing of Cardan joint with roller needle bearings on the cross journals. The tested Cardan joints are used in transmission of power in machine tools, and with small dimensions they provide high level of performance and loading capacity.

The design of cross bearing must provide transmission of power with a certain durability at different torque M_t, rotational velocities n and joint inclination angles α. In designing bearing elements (journal, needles, bushing) all the factors influencing strain concentration need to be reduced. The lengths of roller surfaces of the journal and inner part of the bushing should not depart too much from the effective length of the needle. The needle length is limited by the deflection of the journal of the cross and fork, and by clearances in the bearing, which may lead to delay and this causes additional loading with consequently unevenly distributed loading and premature damage.

The operating characteristic of roller bearings of the Cardan joint is that, as result of variable transmission ratio of this mechanism, the rolling bodies operate in the regime of rolling motion with low oscillations. This results in specific conditions of lubrication of loaded bodies in contact and rolling paths, and therefore the bearing durability calculation has to take into consideration these conditions.

For calculating the durability of the cross bearing in Cardan joints, the expression for the maximum given torque is used:

$$M_{\max} = \frac{2.6CR}{fd \cdot fn \cdot fn\alpha}$$

C N – is the dynamic bearing capacity determined depending on the type of bearing and dimensions

R m – is the distance from the joint axis to the centre of the cross journal

fn – is the coefficient of bearing durability for needle bearings

fd – is the dynamic coefficient - experimentally determined in testing Cardan joints in the device with a closed scheme, as relation of the maximally obtained moment of torsion and statically imposed moment of torsion in the closed contour of the testing device.

fnα – is the coefficient which calculates the magnitude of the product na for the needle bearing

n °/min – is the number of Cardan joint shaft revolutions

α° – angle between the Cardan joint shaft axes

The expression for reference determination of the durability of Cardan joint needle bearing for different loading parameters has the following form:

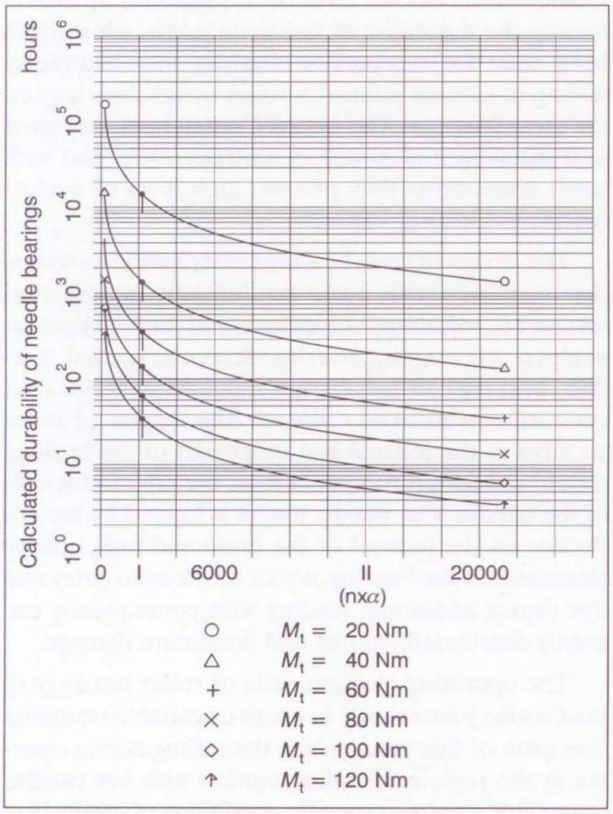

Figure 6

$$L_h = \frac{5 \cdot 10^5}{n\alpha} \left(\frac{2.6CR}{fd \cdot M_t} \right)^{\frac{10}{3}}$$

An overview of the calculated expected durability of bearings in the tested Cardan joint depending on different loading conditions $L_h = f(M_t, n\alpha)$ is given in the diagram in Figure 6.

Diagram in Figure 6 has been used for determining the testing conditions, and for comparison with the experimental data on the durability of needle bearings in Cardan joints.

In variable torque, the operating conditions have been selected at I (n = 710°/min, α = 3°, n α = 2130) and II (n = 1400°/min, α = 10°, $n\alpha$ = 14000).

The expected service life of a bearing can be determined, i.e. assessed, only following the laboratory testing, with the set relations between the loading parameters. In accordance with the Miner's theory, the total predicted durability of the Cardan joint bearing is determined from the following relation

$$L_{h_{EK}} = \left(\sum_{j=1}^{m} \frac{N_j}{L_j} \right)^{-1} = \frac{1}{\frac{N_1}{L_1} + \frac{N_2}{L_2} + \ldots + \frac{N_m}{L_m}}$$

$L_1, L_2, \ldots, L_m$ - are the calculated bearing durability at a certain operating regime, determined according to the expression

$N_1, N_2, \ldots, N_m$ - operating time in a certain loading regime compared to the total joint operating time in percents.

After the testing has been done (when the temperature started to rise suddenly and there were higher vibrations of the shaft) Cardan joints were disassembled

experim.	1			2	3	4					
	w.cond.			P_{Nmax} N/mm²=10³	Lh calc. durat.	gutter depth on cross					
	n min⁻¹	α	M Nm	4 3 2 1		work u.co. h	%	L_h eq h	re h	▲	↕
I	710	3	20		12796	1.32	2.5	160	189.2	0.14	0.07
			40		1269	-	-				
			60		328.5	50.7	96				
			80		126	38.6	73				
			100		59.8	8.72	16.5				
			120		32.6	0.66	1.25				
II	1400	10	20		1946	2.2	2.5	98.5	114	0.2	0.13
			40		193	7.1	81				
			60		50	24.5	28				
			80		19	1.75	1				
			100		9.1	0.17	0.25				
			120		5	0.22	0.16				

Figure 7

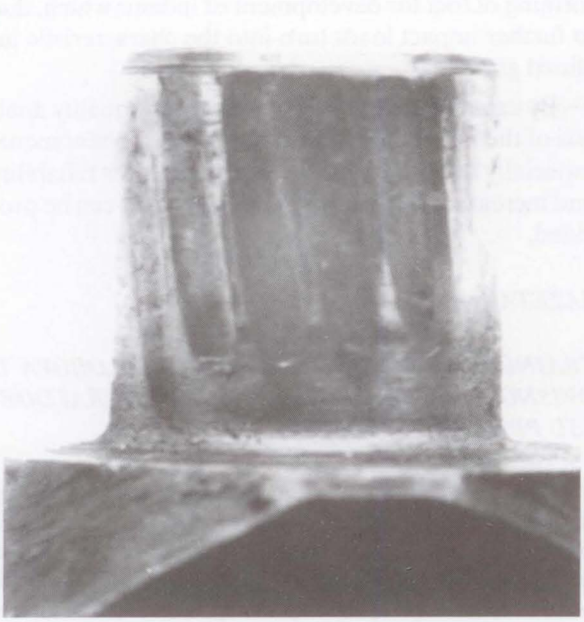

Figure 8

and degreased. In order to determine the level of cross journal surface damage, the depth of the grooves that occur on the cross journals were measured.

The results of measuring the groove depth as well as the actually obtained durability of certain cross specimens were compared with the theoretical values presented in Figure 7. The table also shows the values of calculated maximum contact stresses (Hertz pressures) for the measuring conditions.

4. ANALYSIS OF THE DAMAGING PROCESS OF CONTACT SURFACES OF THE CARDAN JOINT CROSS JOURNALS

The durability of the needle bearing - cross journal couple is determined by the durability of the cross journal surface based on the characteristic form of wear which is indicated by the forming of bevel grooves as in Figure 8.

The grooves are bevelled at an angle of approximately 2-6° to the journal axis, which corresponds to the angle of twist of the needles in the bearing. The bevel grooves obtained by the previously mentioned operating conditions may be explained as a consequence of:

- operation of the rollers at high contact pressures,
- intensive corrosion oxidation,
- possible plastic deformations of the contact surfaces,
- insufficient lubrication,
- variable ratio of the motion transmission.

The process of damaging the journal surface based on the considered occurrences during testing can be divided into two phases. At the beginning of the first phase which is running-in of the surfaces, there are still no indents. At the end of the first phase mechanical damages start to appear (abrasive wearing, as consequence of bonded or tearing activity of the rigid particles of the taken away material, which can be seen on the journal surface as separate cracks. The damage intensity along the length and along the flange of the journal lacks uniformity which can be explained by the inaccuracies in the geometry of the journals and the needles, and by the different value of stresses of certain parts of the contact surface.

Indents start to appear in the second phase, and it can be noticed that they propagate.

In order to explain the process of damaging the cross journal, the character of structural changes, formed in the surface layer in the first phase of the joint operation has to be explained. Therefore, surfaces were tested which had indents of insignificant depths. By measuring hardness on these surfaces, there has been reduction in hardness compared to the initial values. It is assumed that this reduction in hardness occurs because of the high local temperatures leading to yielding and thus reduction in surface hardness. This reduction in hardness occurred unevenly when measured along the length and along the flange of the cross journal. This phenomenon in the first period might be considered as thermal damaging of the journal. The measured values of hardness at some points of the journal surface were about 100 HV less than the initial hardness. This reduction in surface hardness strongly influences the occurrence of cross journal surface deformation.

After the indented foci have been formed, the kinematics of needle movement changes significantly. The increased radial clearance due to the journal surface wear depending on the location of the indent focus enables twisting of needles and keeping them thus twisted. The reduction of the contact area causes increase in edge pressures and leads to further development of indent, thus making the needles still more firm in their position. The increase in the depth of such grooves creates knocking in the bearing and increases vibrations in the joint shaft.

After the measurements, the hardness on the journal surface was controlled, so that hardness in the grooves and on the contact surfaces between the grooves was measured. It was noted that hardness in the groove depth is by about 150HV greater than hardness of the surface between grooves. This could be explained by the fact that in the first phase of surface damaging, there came to yielding in the surface layer which hardened at the indent foci locations, i.e. the material hardens due to microplastic deformations in pressing the surface layers of the needles and the cross journal. Such hardening did not take place at locations between the grooves, and hence such difference in hardness. Thus determined surfaces of the groove bottoms are likely to become brittle after a certain number of stress cycles, which causes pitting within the groove.

5. CONCLUSION

These considerations lead to the conclusion that in case of testing as well as applying of universal joints, it is necessary to know the elements which influence the stability of their system. By changing the influencing parameters, it is possible to optimise the structures in order to avoid the resonance zones. With the use of computers, solving of the basic differential equations of motion is made simpler, and the possibility of varying and simulating changes is greater, especially in simulating friction that has a stabilising character.

The mathematical model applied for theoretical determination of the durability of universal joints showed satisfactory matching with the results obtained from laboratory testing.

The influence of geometry of universal joint elements on the durability of the observed crosses was not noticed. The measured values of the vertical character of the cross journal axis, as well as the fork axis, were within the allowed limits of up to 1°.

Thermal treatment after carbonisation significantly influences the durability of crosses. The reduced hardness of the cross journal surface layer as consequence of greater quantities of residuary austenite is one of the basic reasons for premature damage of the contact surfaces. Higher values of Hertz pressures at the roller contact surfaces of the needles and the cross journals, increase the possibility of such damaging. The values of the measured depths of the inclined grooves, as a characteristic occurrence of damages in cross journals of universal joints, indicate the significant influence of hardness of the journal surfaces. Every reduction in hardness offered the opportunity of forming of foci for development of indents which, due to further impact loads turn into the characteristic inclined grooves.

By expanding the experiments and by quality analysis of the insufficiently studied observed phenomena, especially the occurrence of pitting, greater reliability and increased durability of universal joints can be provided.

SAŽETAK

TRAJNOST ELEMENATA SFERNIH ZGLOBOVA U OVISNOSTI O POJAVAMA OŠTEĆENJA U RAZDOBLJU PRIJELAZNIH POJAVA

U sklopu rada razmatra se utjecaj veze između dinamičkog ponašanja sfernog zgloba i pojava oštećenja u razdoblju prijelaznih pojava. Na osnovu teoretskog proučavanja sustava vratila sa sfernim zglobovima utvrdila se klasifikacija, raspored i veličina dinamičkih opterećenja te definirao dinamički model opisan diferencijalnim jednadžbama. Laboratorijska ispitivanja su usmjerena na provjeru sigurnosti i trajnosti u komparaciji s konstruktivnim zahtjevima, točnosti i kvalitete izrade, te mehaničke i toplinske obrade za jedan karakterističan tip sfernog zgloba.

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