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INFLUENCE OF VIBRATIONS ON SPHERICAL JOINT ENDURANCE IN SERVICE

ABSTRACT

The paper deals with the causes and consequences of vibrations occurring during service on spherical joints in vehicles. It explains the main causes of vibrations and their effects on the most fragile joint elements, and shows the damages on the joint surface. On the basis of these considerations, conclusions are drawn regarding their influence on the lifetime and loading capability of the joint assembly. Consequently, recommendations for optimum conditions for fitting into vehicles and service can be given. The paper also deals with the problem of determining vibrations as one of the element showing the state of joints in service, as well as using the results in order to detect early damage on spherical joints in vehicles.

KEY WORDS

influence of vibrations, endurance, spherical joint, exploitation

1. INTRODUCTION

The lifetime of spherical (universal) joints is indicated in the manufacturers' catalogue data obtained by laboratory testing at constant parameters: loading, installation angle and rotation speed. These data are used as orientation data for design, and under the conditions of real service regime there occur deviations in the form of momentary impact stochastic loadings or vibrations. Service with variable parameters has a substantially different effect on the lifetime of the joints, especially if these are installed on heavy vehicles. The lifetime of joints depends, apart from the loading parameters, also on the quality of manufacture and lubrication of bearing elements, whose durability in the majority of cases represents the durability of the whole joint.

Since joints are compact assemblies of articulated mechanisms, and operate within them as closed tribo-systems, it is difficult to notice the changes on the rolling surfaces and the beginnings of damages on the bearing elements without disassembling the whole joint, which is a big and expensive undertaking. The mentioned joint disassembling is not usually done since this means putting the vehicle out of service for a long period of time, and the checking of damages does not give any visible results without a good laboratory testing.

In testing the joints on heavy vehicles, cargo vehicle of KAMAZ type and crane lorries, two comparison methods were used to detect the origin of damage in bearing elements of the joint:

a) measuring the change in temperature growth rate as indicator of increased wear i.e. operating friction,
b) measuring of vibrations in spherical joints i.e. vibrations that are transferred to the bearing supports of their telescopic shafts, as consequence of wear of the joint bearing elements.

Measuring of the average temperature value was done on the covering part of the bushing i.e. covers of needle bearings in spherical joints, rather than on the contact points of rolling elements in bearings. Therefore, these data, due to heat dissipation do not provide actual reflection of tribological occurrences in joint bearings, i.e. real phases of their wear. Previous laboratory testing, proven also during testing on vehicles, have shown that the temperature rise gradient of 3°C/min on the covering parts of the bearings indicates intensified wear, in contact surfaces of bearing elements in normally loaded and in overloaded and obsolete joints during their progressive damaging (which leads to destruction).

The mentioned reasons have led us to supplement the diagnosing of the state of spherical joints in vehicles by measuring vibrations occurring in joints, which are transferred to their bearing support in the vehicle. Increased wear of spherical joint elements affects increase of clearance which additionally increases vibrations on contact surfaces in the already complex service conditions. This may be used for rough determination of single states of wear and estimate of the durability of bearing elements, and thus of the joint as a whole as well.

2. THEORETICAL CONSIDERATIONS

Theoretical considerations show that non-resonance service regimes, unlike resonance ones, do not cause vibrations indicated by disturbances in movement, i.e. loading. Vibrations of resonance state of joint mechanisms during operation influence damaging of bearing elements and their propagation. Causes of these vibrations are described by the usual differential equations of movement of multi-joint mechanism propelled by internal combustion engine. The driving torque changes in that case according to the laws of harmonious movement, and methodics and application are the properties of any joint mechanism (with a certain number of joints) which provides periodical movement.

Provided that:

$\varphi_{11}=\varphi_i$; $\omega_{11}=\omega_1$; output moment $T_{iz} = T_k =$ const.; angular speed (crankshaft) driving engine ω_p, relation $\xi=\frac{\omega_P}{\omega_1}$ and $j=\sqrt{-1}$ then the change of driving torque at every joint with k = 0,5; 1; 1,5; 2;...;n can be expressed in the form $T_P=T_0+\sum_k T_k \sin(k\xi\varphi_1+\Theta_k)$, i.e. a differential equation of movement is obtained for spherical joints mechanism in the form

$$\frac{d^2y}{dz^2}+(a-2q\cos 2z)y=\frac{T_{iz}}{I_5\omega_1^2}-\frac{T_0}{I_1\omega_1^2}+$$
$$+\frac{a\sqrt{A^2+B^2}}{2j}\left(e^{2j\delta_1}e^{2j\varphi_1}-e^{-2j\delta_1}e^{-2j\varphi_1}\right)+$$
$$+\sum_k\frac{T_k}{2I_1\omega_1^2 j}\left(e^{-kj\theta_k}e^{-kj\varphi_1}-e^{kj\theta_k}e^{kj\varphi_1}\right)$$

which is reduced to Mathieu's non-homogeneous differential equation, thus simplifying the theoretical analysis of the system.

The majority of theoretical considerations performed in transmissions with two or more spherical joints do not take into consideration the transmission elements masses, i.e. the resulting forces of inertia. In designing the exact transmissions with low masses of transmission elements, the forces of inertia are neglected out of practical reasons, in relation to the magnitude of active driving and reactive forces (from which they are several times smaller). Otherwise, they need to be taken into account.

If the driveshaft rotates at high constant speed, the driven shafts rotate at periodically variable angular speeds, the change periods being very frequent, and resulting in periodical angular accelerations. In such cases, apart from moments of reaction forces, also the moments of inertia of masses at output shafts need to be added

$$T_i=-\varepsilon_i I_i$$

In cases of greater clearance in kinematic couples of the system, variable moments caused by forces of inertia can cause impact loading that result in damage at shaft bearing supports and cross-shafts of the system. This tends to be avoided, and therefore the following condition must be fulfilled in transmission:

$$T_i-\varepsilon_{\max}I_i\geq 0$$

In exploitation of cars and other vehicles, angles between spherical joint shaft axes during service constantly or frequently change (i.e. $\alpha_{i;i+1}\neq$ const.) thus not fulfilling the conditions of homokinetic transmission, which can be the basis for the start and support for a resonance state. Apart from forces and moments as consequences of in-service loading, the action is also exerted by forces and moments that are caused by changes in angle $\alpha_{i;i+1}$ (its size and speed i.e. the mode of change), so that the axial, gyroscopic and inertia forces act in addition on the elements of the transmission mechanism.

If the transmission is performed by spherical (universal) joints with intermediate shaft, where one or more bearing supports (i.e. spherical joints) are movable, the spacing between the centres of spherical joints changes during transmission, and that change of spacing is permitted by the application of split grooved shafts, that apart from this change transmit also the torque. By changing the length of the intermediate shaft in grooves, friction forces appear that are transmitted over the grooves onto the shaft bearing support, and it is possible to determine them by the following expression:

$$F_{tr}=\pm 2\mu_u\frac{T_i}{d_{sr}}$$

T_i – moment at the output shaft,

μ_u – friction coefficient in grooves of the intermediate shaft,

d_{sr} – mean diameter of the grooved shaft.

The friction forces in intermediate shafts are transmitted as axial loading onto the bearings of driveshafts and onto the bearings of cross shafts of spherical joints on them. By acting in the centre of the joint, they can be disassembled into components in the direction of the space co-ordinate system. The components determined by friction forces in the grooved parts of intermediate shafts are added to the loading components of the mechanisms with spherical joints.

During the shift of one of the spherical joints of intermediate shafts that rotate around the two mutually perpendicular axes, at angular velocity of ω_{i+1} around the longitudinal axis and at an angular speed of $\dot{\alpha}=\Omega_{i+1}$ around the joint centre. Because of that, the additional gyroscopic moments act additionally on the intermediate shafts.

$$\vec{T}_G=I_{i+1}\left(\vec{\omega}_{i+1}\, x\, \vec{\Omega}_{i+1}\right)$$

I_{i+1} – moment of inertia of intermediate shaft mass,

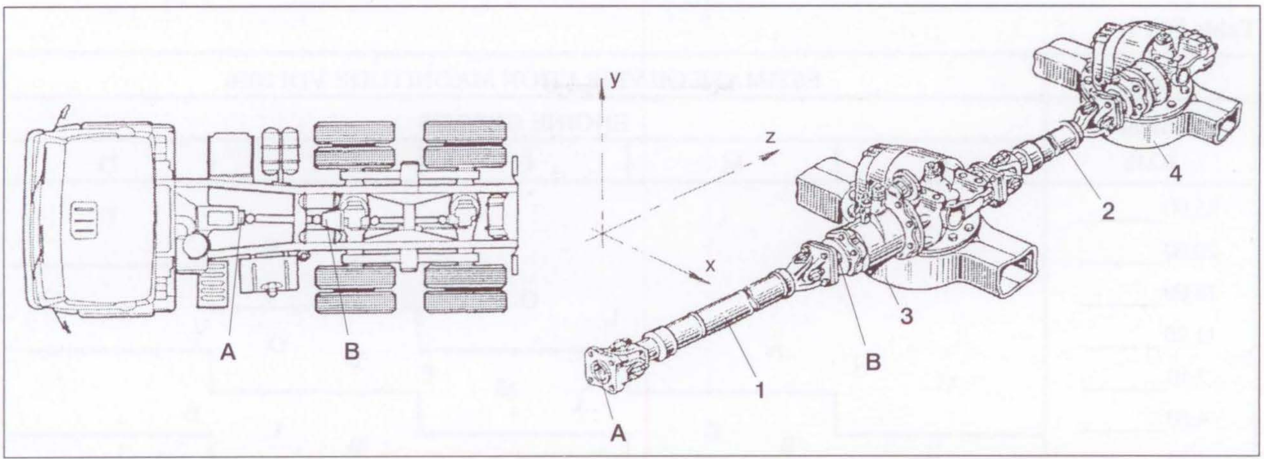


Figure 1 – Measuring points and directions of measuring vibrations on the KAMAZ 55 111 lorry

ω_{i+1} – angular speed of the intermediate shaft around the longitudinal axis,

Ω_{i+1} – angular speed of the intermediate shaft around the spherical joint centre

Gyroscopic moment achieves the influencing value even when it concerns a low moment of inertia of the intermediate shaft masses I_{i+1}, and high angular speeds ω_{i+1} i Ω_{i+1}.

3. TESTING METHODS

From the previous theoretical considerations it is difficult to estimate the durability of bearing elements of joints i.e. joints as assemblies. The greatest problem of such considerations is the estimate of the real loading model in exploitation conditions, which cannot be defined theoretically due to frequent stochastic occurrences. Precisely for these reasons, and aided by theoretical experience we have insisted, if possible, on diagnosing real and momentary indicators of the condition of spherical joints in vehicles. The problems that arise refer to the relatively inaccessible measuring points on vehicles, and the lack of possibility of constant measuring under full load during exploitation. In order to estimate the condition and wear of certain joints, measurements were performed on load-free vehicles (with a determined number of clocked up operating kilometres) in stationary condition. The results of vibration measurements during testing served as comparison data for the analysis of condition of the joint bearing points, performed parallel with the measuring of the joint temperature increase rate over shorter time intervals as orientation indicators of wear.

Measurements of vibrations during in-service testing were performed on the spherical joints of the telescopic shaft (1), which propel the middle axle (3) of the KAMAZ 55 111 type lorry over the intermediate shaft differential gears, i.e. on their bearing supports A and B presented in Figure 1. In crane lorries the driving moment is transferred from the gear unit to the differential gears by means of three spherical joints connected to the telescopic shafts (presented in Figure 2). The points of measurements include the bearing of the gear unit joint and the bearing of the middle joint connected with the vehicle chassis (indicated by equal symbols as in Figure 1). Vibrations were recorded at these measuring points in two mutually perpendicular directions X and Y, that are perpendicular to the rotation axis of the spherical joints bearing support.

The following measuring equipment was used: two-channel measuring instrument with vibration analyser, dynamic vibration transformer and microcomputer for numerical analysis with attached equip-

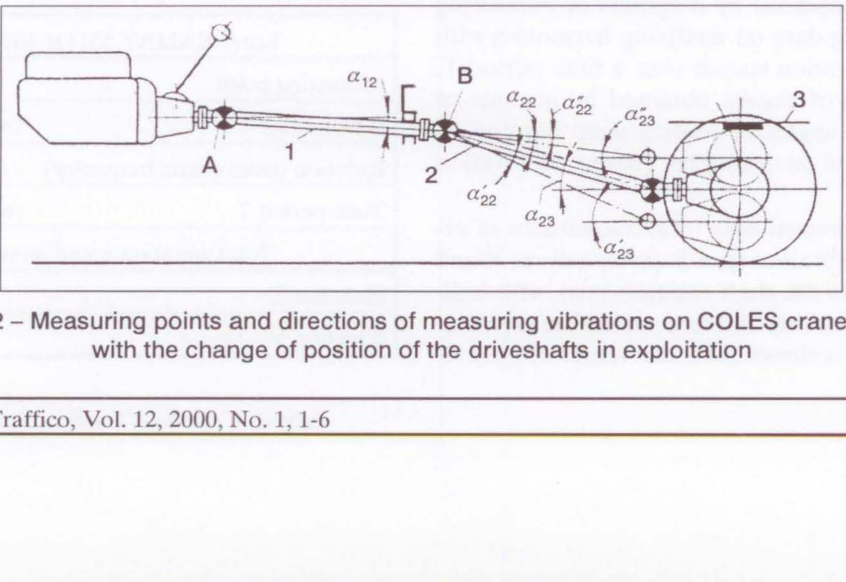
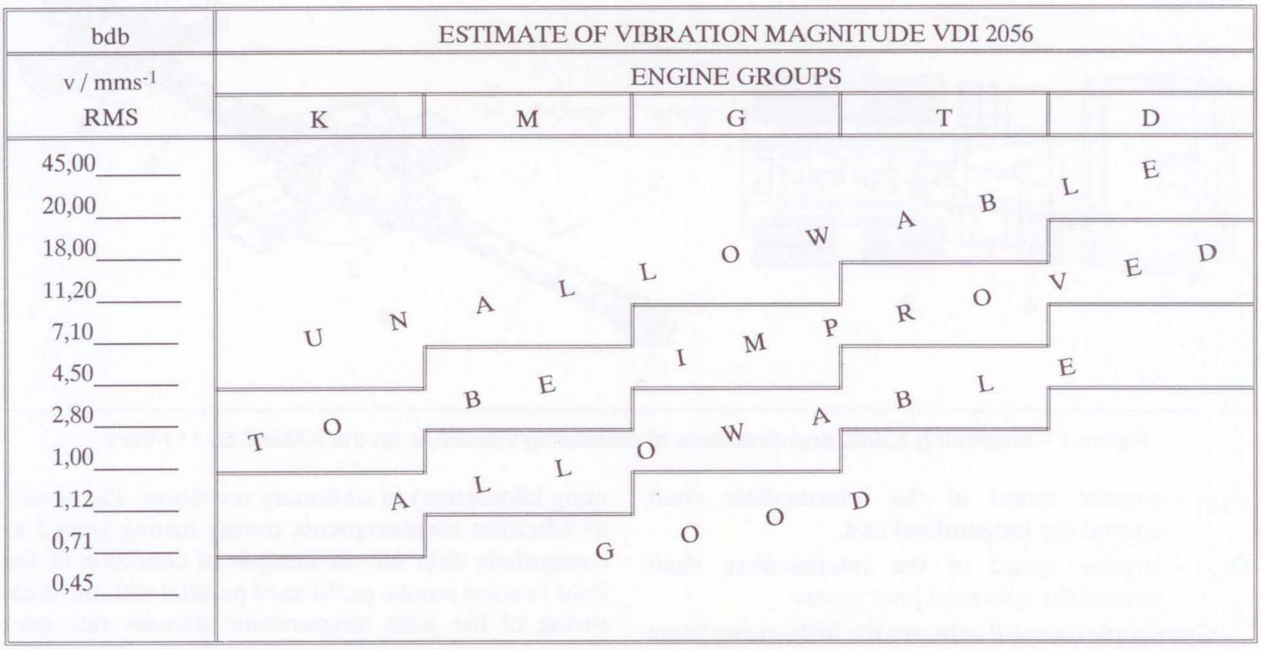


Figure 2 – Measuring points and directions of measuring vibrations on COLES crane lorries, with the change of position of the driveshafts in exploitation

Table No. 1



ment. The measurement results are presented in the form of diagrams and related tables containing data of effective and summary values of vibration speeds V_{ef} /mms^{-1} in the indicated directions of measurements, revolutions i.e. rotation speed n/min^{-1} or s^{-1} and time period T /ms, which corresponds to the time of one revolution of a joint, i.e. shaft.

4. TESTING RESULTS AND ANALYSIS

The tables indicating results with values of speeds and phase angles of harmonics for certain rows (as multiple of the rotation speed) are used to register the total vibration speeds. Based on their values, the vibrations magnitude is estimated and their influence on the dynamic state at certain measuring points according to empirical recommendations on the permitted vibration levels VDI 2056 group M, which is in accordance with the ISO 2372 – 1974 standards (appendix in Table 1).

Due to the conciseness of this work, tabular data have been replaced by diagrams of measuring records containing data on analysing harmonics with the change in vibration speeds over a time period T, i.e. presentations of results obtained by analysis of speed and phase angles of several main harmonics, that are contained as multiples (rows) of rotation speeds.

The tables represent only total magnitudes of effective values of vibration speeds (in directions X and Y perpendicular to the shaft rotation axis), with indicated testing parameters and data on vehicle exploitation. The interest is shown for those values of total vibration speeds that according to VDI and ISO standards (and according to Table No. 1) indicate a problematic or extremely excessive magnitude, and which, on the other hand, with comparative temperature measurement results indicate serious damage of the spherical joints bearing elements.

The highest results of measuring vibrations have been noticed on KAMAZ 55111 lorries that travel about 160 000 km, on the shaft bearing of the spherical joint A. It shows a noticeable increase in vibrations with the increase of rotation speed, especially in the direction of Y axis (presented in Table 2 and in the presentation of vibration speeds in Figure 3). The second row of the measured rotation speed has the dominant influence on the total magnitude of vibrations, resulting from the analysis of contained harmonics. With the increase in rotation speed rises also the influence of vibrations caused by the dynamic characteristics of engine performance, whose upper harmonics contribute to the increase of total vibrations.

Table No. 2

Lorry KAMAZ 55111(160 000 km)		
Measuring point		A
Revolutions	/min^{-1}	621,60
Rotation speed (basic frequency)	/ s^{-1}	10,36
Time period T	/ms	96,50
Total vibration speed /mm s^{-1} RMS		
direction X		6,20
direction Y		12,38

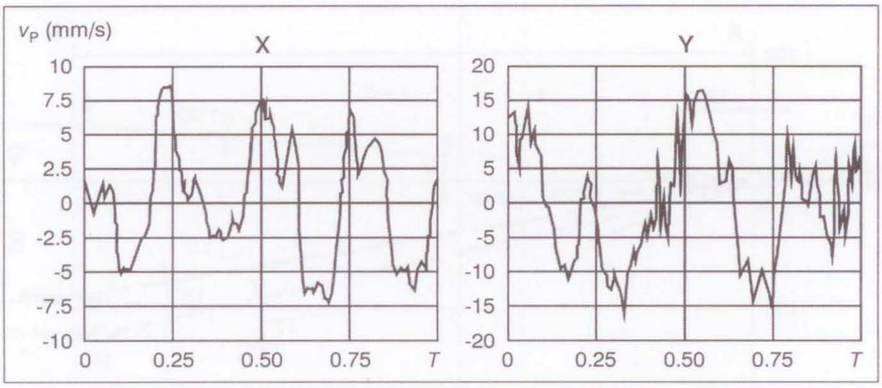


Figure 3 – Vibration speeds in the directions x and y for the measuring point A on the KAMAZ lorry with travelled 160 000 km

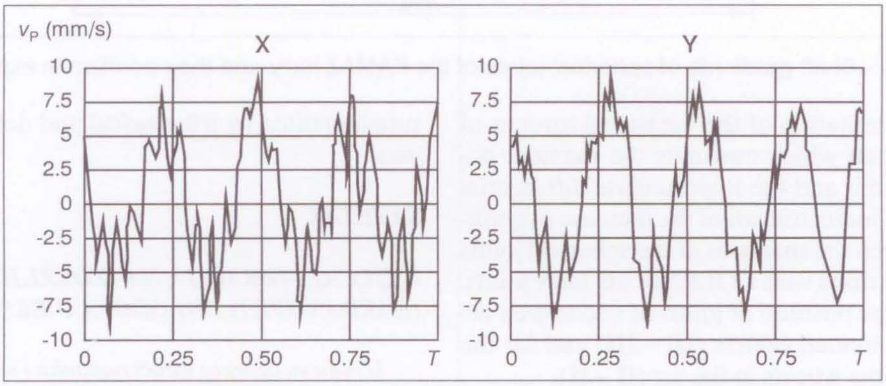


Figure 4 – Vibration speeds in the directions x and y for the measuring point B on COLES vehicle with travelled 70 000 km

Table No. 3

Crane vehicle COLES (70 000 km)	
Measuring point	B
Revolutions /min^{-1}	403,40
Rotation speed (basic frequency) / s^{-1}	6,72
Time period T /ms	148,74
Total vibration speed / mm s^{-1} RMS	
direction X	6,27
direction Y	6,94

Data in Table 3 and the accompanying recordings of vibration speeds in the directions X and Y (Figure 4) represent the measured total values of vibration speeds at bearing point B, for crane vehicles COLES that have travelled about 70000 km.

In relation to KAMAZ type lorries, there are extremely high total vibration speeds in both measuring directions, that are somewhat more pronounced in the direction Y. More intensive and greater changes of gradient angle of the intermediate shaft 2 compared to joint B (α_{22} in Figure 2), that occur during driving, influence faster the occurrence of resonance states in bearing parts of the joint and transmission system. The mentioned change of the gradient angle is not caused by slight changes in vehicle loading, as is the case in a lorry, but only by geodetic levels of certain sections of roads and roadway condition.

It has been noticed that greater damage of these joints occurs in a relatively short exploitation interval, in cases of driving along roads or construction sites that have a variable geodetic level. In such cases, the angles between intermediate driveshafts and joints change during the drive. The design planned mutual relation of angles changes, and the gradient angle of the middle joint α_{22} can be greater than 30°, resulting in non-homokinetic transmission and the occurrence of resonance states and additional loadings due to mass inertia, i.e. gyroscopic moments. The damage that occurs in the form of greater deviation from the circle on the edges of rolling surfaces of journals (in the contact zone of bearing rolling elements) turns with the progression of the damage into partial or total plastic deformations on the bearing journals.

The same cases of damage, manifested by the increased total vibration speeds in the indicated measuring directions, occur during the exploitation of KAMAZ lorries between 140 000 and 160 000 km. The design solution of the transmission of moment by means of two driving axles is compensated by the increase in gradient angles between the spherical joint shaft axes which realise the transmission to the driving

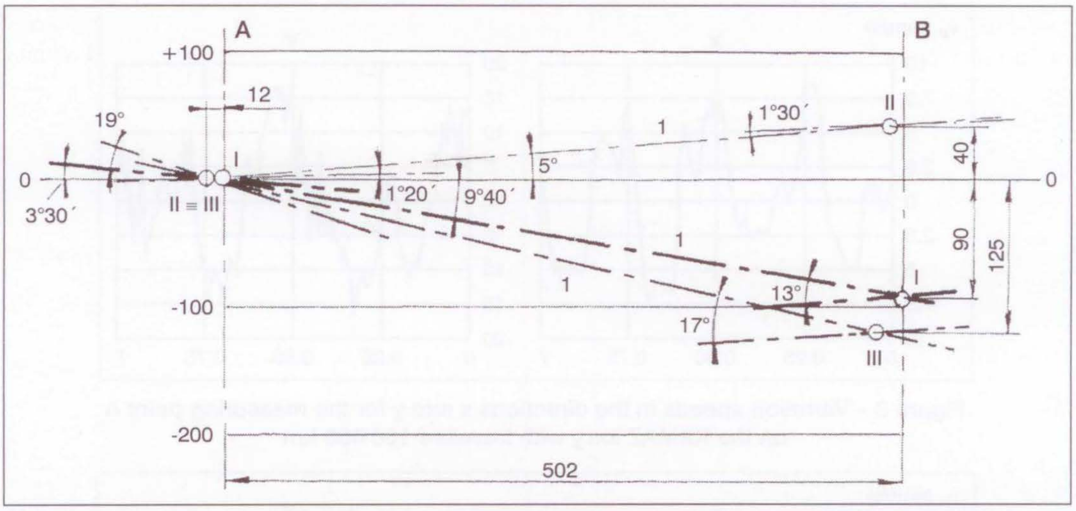


Figure 5 – Shaft gradients of spherical joints of the KAMAZ lorry and their position in exploitation

wheels. The presentation of the measured torques of the telescopic shaft which transmits the moment between the gear unit and the intermediate differential gears (Figure 5), indicates half of the increase in gradient angles between the shaft axis of the spherical joints compared to the cases with COLES crane lorry joints. This is true for the position of joints of a load-free vehicle (I- I), fully loaded vehicle (III – III) and for the lifted body with the wheels in the air (II – II).

5. CONCLUSION

The study has confirmed the efficiency of measuring vibrations on the driving units in vehicles. In the described cases of application on the spherical (universal) joints it is necessary to use the results obtained from these measurements along with monitoring the change in temperature and a series of defectoscopic analyses, exclusively as comparative indicators. In standing vehicles and those in service, various sources of disturbance are present, and their origin may influence the magnitude of the measured total vibration speeds. Apart from the effects of unbalanced operation of the driving engine whose torque changes according to the laws of harmonic movement, there are also the installation inaccuracies, too sudden and uncontrolled wear of certain machine parts and a number of other influences which cannot be predicted in advance. According to experience regarding various types of vehicles and recommendations through standards VDI 2056 and ISO 2372, the level of total vibration speeds supplements the image of dynamic condition in certain spherical joints, that needs to be supplemented by tribological and defectoscopic analyses.

SAŽETAK

UTJECAJ VIBRACIJA NA IZDRŽLJIVOST SFERNIH (KARDANOVIH) ZGLOBOVA U EKSPLOATACIJI

U radu su izneseni uzroci nastanka i posljedice djelovanja vibracija u procesu eksploatacije sfernih zglobova na vozilima. Objašnjeni su osnovni razlozi nastanka i posljedice na najosjetljivijim elementima zglobova kao i prikaz oštećenja na njihovim površinama. Iz navedenih razmatranja izvedeni su zaključci o njihovom utjecaju na trajnost i opteretivost zgloba kao cjeline, te su na temelju toga proizašle preporuke za optimalne uvjete ugradnje i korištenja na vozilima. U radu je dat kratki osvrt na registriranje vibracija kao jednog od pokazatelja stanja zglobova u eksploataciji, te njihovog korištenja u svrhu ranijih otkrivanja oštećenja na sfernim zglobovima vozila.

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