UNITS FOR THE DECREASE OF RADIAL FORCES AND VIBRATIONS IN VEHICLE SUB-ASSEMBLES

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1. Introduction

One of the trends of operation of modern machines – increase of speeds – is inseparably bound with the increase of dynamic loads, affecting details of machines and mechanisms, the appearance of vibrations and so on. Simultaneously, an increasing attention is being paid to reliability-related problems, the decrease of vibrating activity and noises as well as to the decrease of sizes and weights of the structures themselves.

To meet the above-mentioned requirements force designers develop new technical decisions or improve the existing ones.

All these problems are especially urgent for high-speed rotor systems that are essential in the structure of a vehicle. In rotating systems radial forces appear; they affect block bearings of shafts and axles and are transferred to their cases causing vibrations and breaks. It is particularly urgent in sub-assemblies where a rotational movement is transferred between radially misaligned shafts. A proper selection of a connecting unit with the corresponding dynamic and exploiting features may cause an essential impact upon the longevity, evenness of run and the decrease of noises of sub-assemblies as well as comfortability of vehicles. So, investigations of structures of connecting units that allow to ensure isolation of sub-assemblies against vibrations and to reduce radial forces as well as their characteristics are particularly urgent.

2. The Object of the Research

In the presented work elastic centrifugal ring couplings are used as connecting units of shafts of sub-assemblies.

Their essential exploitation parameters are considered the size and shape of the deformation caused by an acting moment of rotation. Couplings of lower stiffness distinguish themselves for wider possibilities of suppression of rotational vibrations. Research of these features was the subject of many works [1, 2]. Herein, we shall discuss one feature of these couplings – a possibility of compensation of radial misalignement of shafts to be connected that causes the decrease of arising reactions and suppression of radial vibrations.

As it is known when a rotational movement is transferred between misaligned shafts 1 and 2, reactions R_1, R_2, R_3 and R_4 inevitably appear in block bearings (Fig 1).

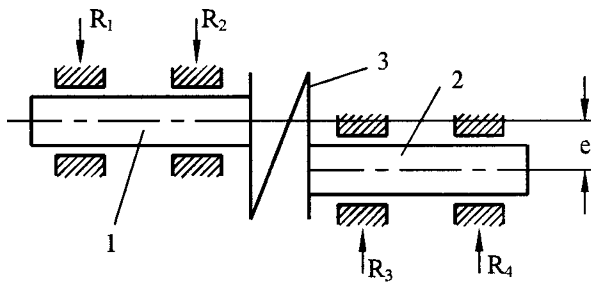

Fig 1. Connection of two misaligned shafts with a coupling

At first sight, the values of these reactions depend on the value of misalignement e, and, besides, they vary within a revolution.

The less is radial stiffness of the connecting coupling 3, the less reactions appear.

In the couplings developed by the authors the radial stiffness also depends on the angular rotational speed: the increase of the speed decreases the stiffness.

3. The Principle of Decreasing Radial Stiffness

Let us discuss the mechanical model presented in Fig 2. There is a rotating shaft 1 with a hole drilled in it and a pivot 2 passing through the said hole. Masses 3 are fixed to the ends of the pivot. Cylindrical springs 4 are inserted between the masses and the shaft.

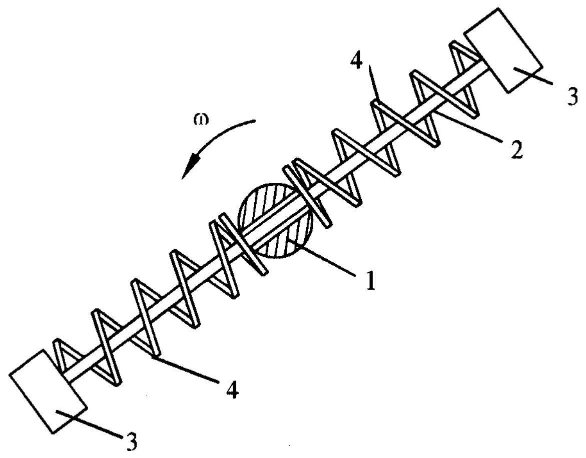

Fig 2. The Model for the illustration of the decrease of radial stiffness

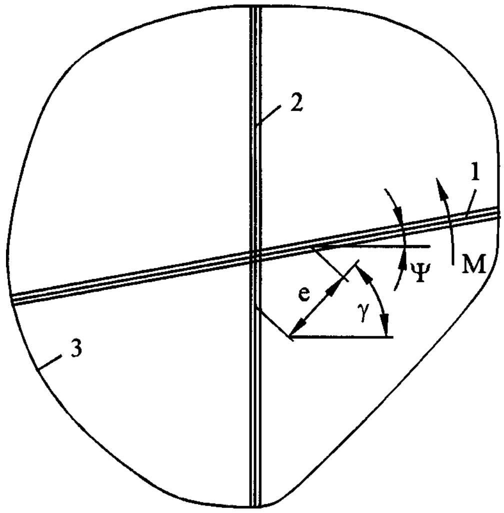

Fig 3. The scheme of elastic centrifugal ring coupling connecting misaligned shafts

The radial stiffness of this rotating model will be found as the second derivation of its kinetic potential on the shift, i.e.

$$c_R = \frac{\partial^2(\Pi - T)}{\partial X_R^2}, \tag{1}$$

where c_R – radial stiffness, Π – potential energy, T – kinetic energy, X_R – the shift of the masses 3 in radial direction. After an insertion of the expressions of kinetic potential and the performance of differentiation, we shall find from the expression (1) that

$$c_R = 2c_S - 2m\omega^2, \tag{2}$$

where c_S – the stiffness of springs, m – the mass of the masses 3, ω – angular rotational speed.

It may be concluded from the expression (2) that when $c_S = m\omega^2$, the pivot 2 may be pushed in radial direction practicably without a force, i.e. the system has the "zero" radial stiffness.

4. The Structure and the Principle of Operation of Elastic Centrifugal Ring Coupling

The effect of the decrease of radial stiffness on an increase of angular rotational speed may be achieved by the use of so called elastic centrifugal ring coupling. The simplified principal scheme of its structure is presented in Fig 3. The half coupling 1 is connected with the driving shaft, the half coupling 2 – with the output shaft and a steel ring 3 situated between the half couplings connects their ends.

As it may be seen, the coupling connects the misaligned shafts with the distance e between their axes of rotation and because of this, the elastic ring 3 is a little deformed.

The potential energy of the coupling Π is accumulated in the deformed elastic ring and depends on:

$$\Pi = \Pi(e, \gamma, \Psi), \tag{3}$$

where e – the value of misalignement, γ – the angle of its direction, Ψ – the angle of deformation of the coupling.

When the coupling starts to rotate kinetic energy may be expressed by the following dependence:

$$T = T_0 + \Delta T(e, \gamma, \Psi, R, \gamma_n, \omega), \tag{4}$$

where T_0 – kinetic energy of the non-deformed ring, ΔT – the change of this energy depending on: R – the radius of the curvature of the ring, γ_n – the apportioned mass of the ring, ω – angular speed. Other symbols are the same as above-mentioned.

Thus, it is evident that when

$$\omega^2 = \frac{\dfrac{\partial \Pi(e, \gamma, \Psi)}{\partial e}}{\dfrac{\partial T(e, \gamma, \Psi, R, \gamma_n)}{\partial e}}, \tag{5}$$

radial stiffness of the coupling is equal to zero and the additional reactions R_1, R_2, R_3 and R_4 (Fig 1) do not arise.

The performed research showed that the value of kinetic energy can be rather precisely approximated with the following expression:

$$T=T_0+0,131R^2\gamma_n\omega^2 e+35,85R\gamma_n\omega^2 e^2 - 0,815R^2\gamma_n\omega^2 e\Psi - 0,619\, R^3\gamma_n\omega^2\Psi-5,763\, R^3\gamma_n\omega^2\Psi^2. \tag{6}$$

In Fig 4 the dependence of radial stiffness of the coupling on various parameters is shown. The calculations were performed for the case when the radius of the curvature of the ring R = 0,15 m, the modulus of elasticity $E = 2 \cdot 10^{11}$ N/m^2, the moment of inertia of the cross-section of the ring $I = 133 \cdot 10^{-12}$ m^4, γ_n = 0,2 kg/m, ω = 0…400 s^{-1}.

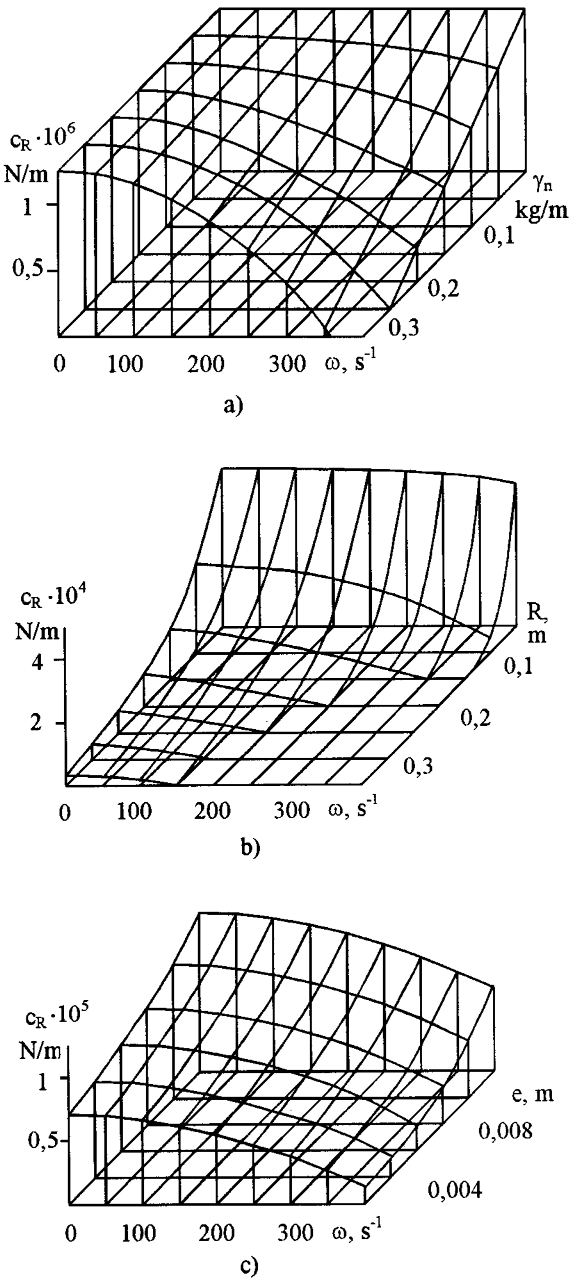

Fig 4. Dependence of radial stiffness of the coupling on various factors

5. Research of Radial Vibrations

The dependence of radial stiffness of the cling on the angular rotational speed considerably hardens the research of its dynamic characteristics. The decrease of stiffness impacts the values of reactions of the block bearings and, in addition, forms new conditions for radial vibrations.

Frequency of eigen radial vibrations of the coupling can be expressed as follows:

$$p = \sqrt{\frac{-\dfrac{\partial^2 T(\delta)}{\partial \delta^2} - \dfrac{\partial^2 \Pi(\delta)}{\partial \delta^2}}{m_R}}, \tag{7}$$

where m_R – the reduced mass of the elastic element, δ – the position coordinate, describing the configuration of the rotating coupling and specifying the shift of the center of gravity of the elastic elements from the axis of rotation.

Let us analyze the vibrations of the ring details in radial direction when a misalignement of shafts takes place. On a rotation the misalignement causes an exciting force proportional to the value of misalignement e that depends on the angular rotational speed as well. The equation describing the vibrations may be expressed as follows:

$$m_R\ddot{x} + c_R(\omega)x = F(e)\sin(k\omega t + \gamma_0), \tag{8}$$

where x – the shift of details of the construction in radial direction, m_R – the reduced mass of the elastic elements, $c_R(\omega)$ – the radial stiffness of the coupling as a function of angular rotational speed, F(e) – the amplitude of the exciting force proportional to misalignement e, k – the coefficient depending on the number of points of fixation of the elastic ring to half couplings (k = 4, when the half couplings are with two ends and k = 6, when they are with three ends each), t – time, γ_0 – the angle of the direction of misalignement with one of the half couplings on the initial moment.

If it is considered that on the initial moment $\gamma_0 = 0$, the solution of the equation (8) is the following:

$$x = \frac{F(e)}{m_R\left(p^2 - k^2\omega^2\right)}\sin\left(k\omega t - \frac{k\omega}{p}\sin pt\right), \tag{9}$$

and the frequency of eigen vibrations:

$$p^2 = \frac{c_R(\omega)}{m_R}. \tag{10}$$

Equations of this type are widely used and well-investigated in engineering [3].

In general we are interested in the angular speed ranges where the coupling may operate as a damper of vibrations and in the ranges where excited vibrations will be amplified. In other words, it is necessary to determine the ratio of the amplitudes of the engine side half coupling and the pump side half coupling on various rotational speeds. Such frequency response may be found from the equation using the method of finite elements [4]:

$$[M][\ddot{u}]+[h][\dot{u}]+[c][u]=\{R(t,e,\Psi)\}, \qquad (11)$$

where [M]– the matrix of masses, [h], [c] – the matrices of coefficients of damping and stiffness, respectively, [u], $[\dot{u}]$, [ü] – the matrices of shifts, speeds and accelerations of corresponding points, $\{R(t,e,\Psi)\}-$ the vector of exciting forces.

Let us consider that the engine side half coupling is affected by radial vibrations of the shaft. In this case the exciting force shall be expressed as follows:

$$\{R(t,e,\Psi)\}=-[M]\{\ddot{u}_v\}, \qquad (12)$$

where $\ddot{u}_v$ – accelerations of radial shifts of the shaft.

Then the expression (11) turns into the following:

$$[M][\ddot{u}_r]+[h][\dot{u}_r]+[c][u_r]=-[M]\{\ddot{u}_v\}, \qquad (13)$$

where $[u_r]$ – relative shift of details of the construction.

$$u_{ri}=u_i-u_v. \qquad (14)$$

This task may be solved in two ways.

In the first method a stepped integration of the equation (13) is carried out. In the second method the following fact is used: the reaction of a dynamic construction to an exciting force may be described by several lower eigen frequencies and shapes of vibrations [5].

Using the transformation

$$[u]=[\Phi][x], \qquad (15)$$

where $[\Phi]$– the normalized shapes of eigen vibrations, [x] – shifts of details caused by the impact of the exciting force, the expression (13) may be written as follows:

$$[\ddot{x}]+[\Delta][\dot{x}]+[\Omega]^2[x]=-[\Phi]^T[M]\{\ddot{u}_v\}, \qquad (16)$$

where [Δ] - matrix – column

$$[\Delta]=diag(2\omega_i\xi_i), \qquad (17)$$

where ω_1 – the value of the corresponding eigen frequency), ξ_i – the coefficient of damping of the construction, [Ω] – the matrix, corresponding to the following expression

$$[\Omega]^2=diag\left(\omega_i^2\right), \qquad (18)$$

$[\Phi]^T$– the transformed matrix of the shapes of the normalized eigen vibrations.

The solution of the equation (16) enables to determine a dynamic reaction to any excitation.

Herein, we present (as an example) the dependence of the ratio of the amplitudes of the engine side half coupling and pump side half coupling of the coupling on the angular rotational speed (Fig 5). It was considered that the exciting force was sinusoidal and the geometrical parameters of the coupling are the following: R = 0,1 m, the radius of the ring – 2 mm.

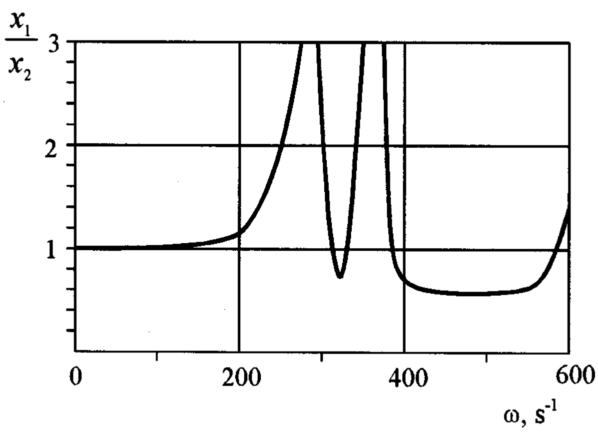

Fig 5. The dependence of the ratio of amplitudes of radial vibrations on angular rotational speed of the coupling

6. Conclusions

The performed research allows to present the following conclusions:

1. If rotating masses are located in a certain way, it is possible to develop constructions of couplings with radial stiffness equal to zero.

2. Radial stiffness of an elastic centrifugal ring coupling depends on the value of the apportioned mass of the ring and angular rotational speed.

3. The found expression of kinetic energy of rotating system enables to find the dynamic component of radial stiffness.

4. The value and direction of misalignement as well as the angle of deformation of the coupling, depending on the transferred moment of rotation slightly impact radial stiffness.

5. The increase of the apportioned mass of the ring reduces the eigen frequencies of vibration of the coupling.

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ĮTAISAI RADIALINĖMS JĖGOMS IR VIRPESIAMS MAŽINTI AUTOMOBILIŲ AGREGATUOSE

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S a n t r a u k a

Straipsnyje pateikti originalių įtaisų – tampriųjų išcentrinių žiedinių movų – taikymo mechaninėse rotorinėse sistemose atliktų tyrimų rezultatai. Nagrinėjamas šių movų taikymas radialiai nesąašiems besisukantiems velenams sujungti. Naudojant paprastą mechaninį modelį parodyta, kaip tinkamai išdėsčius mases išcentriniame jėgų lauke galima gauti jungiamosios konstrukcijos radialinio standumo mažėjimą, didinant kampinį jos sukimosi greitį. Šis principas pritaikytas tampriųjų išcentrinių žiedinių movų konstrukcijose.

Nagrinėjant rotorinės sistemos su tokia mova kinetinį potencialą, parodyta, kad kinetinė besisukančio deformuoto žiedo energija duoda radialinio standumo dedamąją, mažinančią šį standumą. Ištyrus nemaža movų konstrukcijų, išvesta įvairiais veiksniais deformuoto tampraus žiedo kinetinę energiją aproksimuojanti formulė. Parodyta, kad radialinio standumo mažėjimas daug priklauso nuo tampraus žiedo paskirstytosios masės dydžio ir kampinio sistemos sukimosi greičio.

Tiriant movos virpesius radialine kryptimi, taikytas baigtinių elementų metodas. Nustatant dažninę movos charakteristiką, atsižvelgta į tą aplinkybę, kad dinaminė konstrukcijos reakcija gali būti aprašyta keletu žemesniųjų savųjų virpesių dažniais ir formomis. Gauta virpesių lygties išraiška, leidžianti nustatyti movos dinaminę reakciją į bet kokį žadinimą.

Pavyzdžiais iliustruoti tirtų movų radialinės standuminės ir radialinių virpesių slopinimo charakteristikos. Parodyta, kad tampraus žiedo paskirstytosios masės didinimas mažina movos savuosius virpesių dažnius.

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