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Vibration response of the boat composite shafting having constant velocity joint during change of the operation regime

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Abstract : The usage of constant velocity (CV) joint is effective for motorboats on gliding regime of the motion. During transition on the gliding when angle of the CV differs from null on driving and driven composite shafts there are moments of the second order. Excitation of oscillations of the second order moments occurs when driving shafts transmits a variable torque, which generates through CV joint a lateral moment acting on the bearing. As a result of oscillations from a resonating harmonic of a shafting the harmonic with the greater or periodically varying amplitude for power condition trough transferring to nominal power 144kW. Beating conditions coincide with third mode having frequency 45.486 Hz. In that case there is high increasing of the equivalent stresses. The forming of the stiffness of the composite material is concerned to use most orientation of the layer angle in the range of ±60 degrees relatively of shaft axis. Application of that angles for layer orientation gives possibility to avoid high disturbance of the shafting for motorboat transition regime.

Key words : CV joint, Shafting, Second order moment, Transferring regime, Vibrations, Beating, Adjustment

1. Introduction

The shafting having constant velocity joint is effectively applied in motorboats to joint the propeller with the engine. Application of that type of CV joint is most effective for motorboats on gliding motion. In a considered case the shafting consists of two shafts and has one joints of constant velocity rate[1]. CV joints can have different slope angles in one plane. The slope angle β of a CV joint in one plane at setting in a shafting is dependent on distance between element of the shafting and length.

During application of a driving torque when the angle differs from null, on driving and driven shafts there are moments of the second order. The contact elements of the CV joint transmitting

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power and being at the plane of the constant rotation. Pairs of the forces on those contacts create moment of second order. The moments of the second order act as static on bearings and are a function only a driving torque and an angle of the CV joint. They are equal both on driving, and on the driven shafts; and the direction of their action coincides with a normal to a plane of an angle between shafts. At assigned directions of a driving torque and an angle of the CV joint both moments have equal inclinations. The torque transmitted from engine to the drive line and angle of the CV joint in that application are variable and shafting fixed on two positions, and is reason of the variable second order moment, which can excite lateral oscillations with bearing interaction. The moment of the second order represents a bending moment which originates because the joint transmits a driving torque trough balls connecting driving inner race and driven bell housing are located under angle relatively driving shaft axis and balls reaction forces create bending moment dependent on torque and inclination angle [2]. In a shafting the moments of the second order act on support structures, and intermediate with other shafts. They can create effect characteristic for activity near to a critical velocity, acting on bearing parts dependent on engine torque oscillations per revolution. Those oscillations can excite vibration of the shaft with frequency, corresponding to one of the critical value. Action couples of forces on shafts excites oscillations for each revolution and develops with others oscillators (for example, owing unbalance), having the same frequency significant amplitudes [3]. Change of angular velocity of shaft rotation connected by CV joint of variable torque, excites oscillations in a power train of a motorboat. Consideration of the second order moment influence on the shaft behavior is important for motorboat having transitional regime on the gliding. In those conditions the angle of the jointed shafts is changed.

2. Modeling and main equations

On the Fig.1 two positions of the motorboat shafting are shown. The driven shaft has slope position for two motor boat speed conditions. When speed is moderate the position of the driven shaft is horizontal. The slope position with angle β of the driven shaft is applied in the high speed conditions for supporting satisfy efficiency of the propeller on

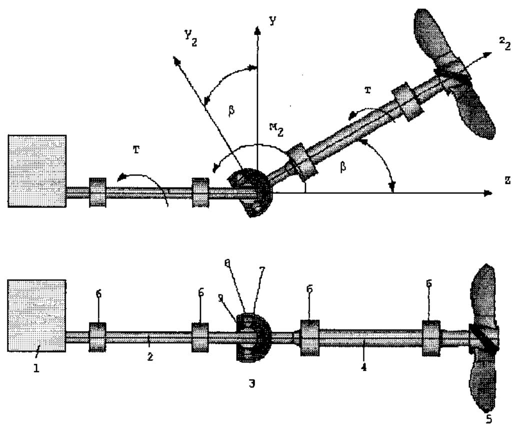

Fig. 1 Motorboat drive line at two positions

gliding motion. Fig.1 shows a shafting, which includes of assigned length driving shaft 2 connected through a CV joint 3 to driven shaft 4 and to the propeller, which spins at angular velocity ω. The torque from the motor 1 is transmitted by driving shafting 2 and driven shaft 4, which installed in the bearings 6 having stiffness and damping performances fixed accordingly to design.

The driving inner race 9 and driven bell housing 7 interacted to each other through balls 8. That system is used in the derivation of the equations of motion for the driving and driven shafts. Coordinate axes XYZ are fixed with origin at the center of the driving inner race with Z directed along the driving shaft. Axis X is perpendicular to the driving line at YZ plane and axis Y is oriented perpendicularly to XZ. A rotation angle β about Y corresponds to driving shaft slope during motorboat operation and can be presented by intermediate axes x_2, y_2, z_2 axis x_2 is perpendicular to the x_2z_2 plane. Moving driven bell housing is attached to the shaft on its central axis and is described by Euler angles β. First a rotation β about X leads to the intermediate axes x_2, y_2, z_2 as shown in Fig.1. The moving coordinate system does not spin with the shaft. The angular velocities of the driven bell housing and driving inner race are same and equal to ω and matrix equations of dynamics include kinematical and stiffness performances of the system. The model of a CV driveline shown in Fig.1 consists of two parts that are described by mass moments of inertia (J_x, J_y, J_z)

$$j_x\{\ddot{\theta}_x\}+C_x\{\dot{\theta}\}+K_x\{\theta_x\}=\sum_{k=1}^{m} m_{sk}\ \tan(\beta/2)\sin(k\omega_x t)$$
$$+\sum_{k=1}^{m} m_{ck}\ \tan(\beta/2)\cos(k\omega_x t) \qquad (1)$$

$$J_y\{\ddot{\theta}_y\}+C_y\{\ \dot{\theta}_y\}+K_y\{\theta_y\}=\sum_{k=1}^{m} m_{sk}\ \sin(k\omega_y t)$$
$$+\sum_{k=1}^{m} m_{ck}\ \cos(k\omega_y t) \qquad (2)$$

$$J_z\{\ddot{\theta}_z\}+C_z\{\dot{\theta}_z\}+K_z\{\theta_z\}=\sum_{k=1}^{m} m_{sk}\ \sin(k\omega_z t)$$
$$+\sum_{k=1}^{m} m_{ck}\ \cos(k\omega_z t) \qquad (3)$$

where J_x, J_y, J_z is mass moments of inertia around correspond axis; C_x, C_y, C_z, - damping coefficient of the shafts on the axes X, Y, Z; K_x, K_y, K_z X, Y and Z components of the shaft stiffness; m_{sk} and m_{ck} sin and cos components of the driving torques T harmonics; $\ddot{\theta}$, $\dot{\theta}$, θ - components of the angular acceleration, angular velocity and angular displacements of the shafts around corresponding axis; m is quantity of the harmonics is number of harmonic.

Right side of the equation (1) represents the second order of the moment M_2, which is excited by join due to angle β of the driven shaft slope, which is connected to the driving inner race through ball. The plane of the balls has slope angle β to the axis Z. Damping coefficients C_z are defined at the nodes, which represent the bearing having stiffness and damping properties as part of the fined element model of the system [4]. The coefficients of the terms in cosine and sine are expressed respectively by equations:

$$m_{ck}=\{m_{mk}\ \cos(\phi_{mk}+\delta_k)\}+\omega_z^2\{m_{ik}\ \cos(\phi_{ik}+\delta_k)\} \quad (4)$$

$m_{sk}=\{m_{mk}\sin(\phi_{mk}+\delta_k)\}+\omega_z^2\{m_{ik}\sin(\phi_{ik}+\delta_k)\}$ (5)

where ϕ_{mk} is phase of k-th harmonic of the mass moment of inertia torqueis, δ_k is phase of k-th harmonic of the torque acting on the driving shaft, ϕ_{ik} is phase of k-th harmonic of the mass moment of inertia torque

The amplitude on the s-mode shape

$$\phi(t)=\alpha q_s\sin(k\omega t) \quad (6)$$

where α is coefficient of proportionality, q_s is s-th eigenvector. The coefficient α is amplitude of the deflected shape

$$\alpha=\frac{\sqrt{[\sum_{j=1}^{n}q_{sj}M_{0j}\cos(\delta_j)]^2+[\sum_{j=1}^{n}q_{sj}M_{0j}\sin(\delta_j)]^2}}{k\omega_z\sum_{j=1}^{n}c_{eqj}q_{sj}^2} \quad (7)$$

where M_{oj} is the amplitude of the k-th harmonic of the driving torque applied to the j-th node, δ_j is the phase lag between the moment applied to the j-th node and that applied to the first node obtainable phase-angle correlation.

For composite shaft the orthotropic configuration to define stiffness properties $K_x\{\theta_x\}$, $K_y\{\theta_y\}$, $K_z\{\theta_z\}$ and properties of the layers is selected to be satisfy to vibration load on shafting with accounting second order moment influence. The stiffness coefficients are based on the reduced stiffness definition corresponding to axis of the shafting orientation. The reduced stiffness Q_{ij} of the layers are selected in terms of the independent engineering material constants in principal material direction as

$$Q_{11}=\frac{E_1}{1-\nu_{12}\nu_{21}},\quad Q_{22}=\frac{E_2}{1-\nu_{12}\nu_{21}},\quad Q_{12}=\frac{\nu_{21}E_1}{1-\nu_{12}\nu_{21}},\quad Q_{66}=G_{12} \quad (8)$$

Two of the four independent material constants are the elastic modulii in the first and second principal directions E_1 and E_2. Main two material constants are the shear modulus in the 1-2 plane G_{12} and v_1 and v_2 the major Poissons ratios is defined as the negative ratio of the strain in the 2 directions. For selection of the of the layer stacking angle the influence of Poisson coupling effect on the natural frequencies and damping results was accepted mostly at the range near to ±60℃. The strain in one direction gives rise to strain in perpendicular directions due to Poisson's effect [5]. The lateral strain is produced without any external stress in the lateral direction; no strain energy is stored due to Poisson distortion in the lateral direction. In order to make lateral strain zero an external stress in the lateral direction has to be applied for preventing instability at the XY and XZ planes.

A multi-layered scheme was considered with seven angle rates of fiber layers orientation 0°, 15°, 30°, 45°, 60°, 75°, 90°. The flexural frequencies have been obtained for symmetric combinations. As it considers the laminating of the all thin layers has assumed effect to act at the mean radius [6]. This assumption is quite valid for symmetric configurations. For symmetric configurations, the frequencies values are predicted by layer theory accepted for shafts properties construction. Even though for the two symmetric configurations, the flexural frequency does have a slight dependence on the relative positioning +60° and -60° of the layers, and this orientation which assumed due to difference in flexural stiffness, will increase with wall thickness. For construction of the composite shaft properties to avoid instability an vibration exciting is accounted that the fundamental frequencies of the thick wall shafts is decreased monotonically at a constant rate when the rotating speed increases. The eigenfrequency and damping properties of the shafts are selected in function due to geometry, laminate structure and fiber angle because of the variety of possible combinations of these variables for response selected EP/GF(epoxy/glassfiber) composite shafts (composite characteristics see table) having non beating behavior on transitional conditions of the motorboat. The influence of different geometrical parameters and their interaction are presented for cylindrical shells of different lengths and thicknesses having the constant outer radius R_1=0.025 m and inside radius r_1=0.08m of the driving shaft: outer radius R_2=0.035 m and inner radius r_2=0.018 m for driven shaft. The eigenfrequency characteristics of the first and second flexural modes for shafting dependent on the length-to-radius ratio and is estimated in that case with the outer radius/thickness ratios R/h = 1.6 and length /outer radius l/R = 12. For that region of the relative parameters an eigenfrequency and damping performances have not strong dependency from thickness. In that case the shaft is considered having large deformation resistance compared to the flexural deformation resistance of the cross section to vary resulting natural frequency. For assigned geometrical performances and material structure the shear deformation has dominating influence on the damping properties of the shafts, because resistance to cross section deformation grows and shaft can increasingly executes flexural vibration.

	Material	EP/GF
Components	Fiber	ERS2310FW
	Matrix	EP
System	Composite	UD
Fiber volume content	P_F(%)	61±2.0
Young's modulus(axial)	E_x(GPa)	47.77
Young's modulus(transverse)	E_y(GPa)	5.23
Shear modulus	G_{xy}(GPa)	4.72
Axial damping	H(%)	0.155
Shear damping	t(%)	0.736
Density	d(%)	1.42
	ρ(kg/m³)	1810

The propeller is placed in the end of the shaft with assigned stiffness at the outside location of the bearing span and having viscous external damping the coefficient of which is assigned from CFD calculation results. Propeller hub increases the stiffness of the driven shafts. Assuming is made that the thrust bearing accepts all thrust loads. The every shafts are mounted on the elastic bearings due to plastic hull and having small damping coefficient and increase system stiffness. It is considered that the restoring force due to the bearing performance vary with the deformation of them elements. Applications of the rolling element ball bearings have been considered in this work. Though in reality rolling element bearings offer restoring force varying nonlinearly with deformation of the elements, yet, the effect of non-linearity in such cases is not much pronounced particularly in the presence of external damping due to the damping materials in this case. The effect of non-linear restoring force is neglected. Bearings on both sides are supported on identical supports without accounting of deformation behavior of the motorboat hull, i.e., the number, material and position of driving shaft on two bearings are identical relatively of distance from the ends.

3. Discussions

The resonance amplitude of oscillations was estimated at acceleration of the motorboat on transitional conditions as a result of coincidence of frequency own oscillations of a power train of any form with frequency of a harmonic of the shafting and the properties of the composite material is considered to decrease vibration displacement on the operation regimes. Beside that an exciter of such resonance can be of the coinciding order of the CV joint as it has the same oscillation period, as well as a harmonic of the shafting. The level of the oscillation amplitudes is raised and the resonating harmonic of the engine should be defined in zones of a resonance. Hence, in a zone of resonance torsion oscillation from the engine is considered for inclined shafting. On the Fig.2 the resonant zone of the shafting is shown for third mode, which corresponds to acceleration regime of the motorboat on the engine rpm 2729.

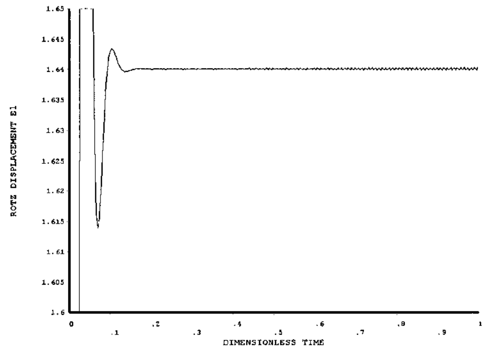

Fig. 2 Torsion oscillations of the driving of node located in the root of the driving inner race.

On the vertical axis the displacements are fixed by degree increased ten times (E1=10), at horizontal axis dimensionless time is fixed represented acceleration time of the boat on considered engine regime.

A shafting having joint of variable torque with different slope angles is consider as the excitation source of torsion vibrations with several harmonics (by angle slope of CV joints). If joints of a shafting have the slope angles in one plane all harmonics of oscillations having the equal period, will have an equal vibration phase. Hence, the common law of oscillation change for all shafting presents a harmonic too. The amplitude of a summarized harmonic is sum of harmonics amplitudes of separate shafts connected in the CV joint. Therefore increasing slope angles of CV joints and torque will increase amplitude of a summarized harmonic of torsion and lateral oscillations. If joint of a shafting have slope angles β in plane, non-uniformity of rotation and consequently amplitude of a summarized harmonic will be increased, because common slope angle β of the CV joint is increased. At motion of a motorboat the angle of the CV joints is varied during transitional regime. It has influence on value of amplitude of a summarized harmonic. Influence of each exciter is especially appreciably in a resonance zone of forced vibrations. For an eigenfrequency and for range of a frequency alternation of an exciting shafting known harmonic, we will define possibility of the resonance creation of the lateral and torsion oscillations. Frequency of a jointed shafts harmonic, which can coincide on all range with some shafting eigenfrequency of the multi-nodal form, will be defined for torsion and lateral oscillations.

As a result of a superposition of oscillations from a resonating harmonic of the engine and harmonics of a shafting the harmonic with the greater or periodically varying amplitude was defined. A phase angle changes character of amplitudes of a summarized harmonic a little, but does not influence on the law of change of oscillations. The amplitude of a summarized curve at a superposition of oscillations from harmonics of second order of the CV joint and a shafting will be increased, as both harmonics will have the equal vibration period. At the superposition of oscillations from a harmonic of the shafting and harmonics of other orders of the engine the summarized harmonic will have periodically varying amplitude. The oscillations of the shafting have less amplitude influence on torsion oscillations in zones of a resonance and, when the oscillation amplitudes from harmonic of the engine are large. If oscillation amplitudes of the engine harmonic have low level and oscillation amplitudes of the shafting have high level, the influence of the shafting oscillations is higher. Beside that the harmonic amplitude values of CV joint are increased proportionally to angle β of the shaft inclination.

Instability of CV joint performances changes values of angular rate and a driving torque, which excites oscillations both in the power shafting and supporting bearers. These oscillations can be shown by principal views; each of them has particular performances and creates only to its generic effect. Disturbance of normal operation is estimated on the specified range of rpm, and maximum values of a vibration level which are determined by results designs selected dependent on assigned part of transitional time. Instability of velocity, which originates by activity of a CV joint elements results in appearance of bending (Fig.3) oscillations.

It is necessary to provide best angles values of the CV joint in all the range of assigned transitional time to suppress amplitude, which can be considered as maximum permissible for the given system. The effect originating by activity in shafting joint is possible to summarize and express through the angle equivalent gyrating ROTX during dimensionless time of the transitional conditions. The amplitude of rotary oscillations, which can be adopted as admissible, i.e. not triggering unacceptable disturbance of optimum performance, depends on operational angular rate and conditions of fixing parts of a power shafting. The specified oscillations at the maximum angular acceleration, which arise during a continued operation with constant velocity such power shafting is quite satisfactory on motorboats. The lower level corresponds to oscillations of the system with CV joint having same angle of inclination at rotary speed to 6000 rpm. A motorboat intended for maintenance at variable conditions and precipitations. Shifting torsion oscillations are stipulated by change of the basic reaction at the supports from a driving torque, vibrations

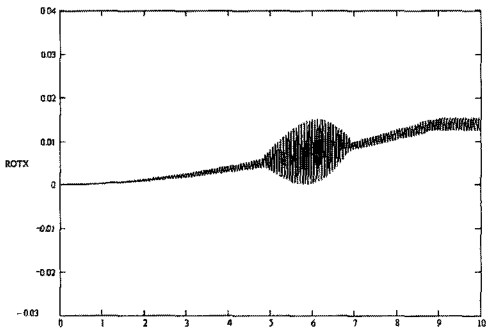

Fig. 3 Beating behavior of the shafting at the root point of the outer driven housing on the transferring regime.

On the vertical axis the displacements are fixed by degree, at horizontal axis time (s) is fixed represent acceleration of the boat on considered engine regime of the engine fastening, power train, and also bearing parts of the shaft train, second order moments of the joint, especially on frequencies, close to own resonance frequencies of these designs.

Increasing of the second order moments oscillations occurs, because driving shafts slope angle became larger on the frequencies close to third mode. On large slope angle the lateral moment is increased through CV joint pieces acting on the bearing. The couples of forces which affect fastening of the shaft create moment by increased distance between centers of the pieces. The supporting bearer placed in a mean part of a customary motorboats CV power shafting is sensitive to these oscillations as it usually has a strongly pronounced resonance frequency. Forces from the moments of the second order will excite oscillations in a supporting bearer on a shaft speed, an equal part of its resonance frequency. It is stipulated by condition, that oscillatory forces for one rotation of the shaft are changed.

Deviations from normal operation significantly depend on a rotary speed. The driving torque deviates in rather narrow range of the rotation speed, preset service conditions, at motion with a top speed or at an inertial motion when angular velocities are below of nominal rpm and the bending of support structures provokes to vibrations is significant. The shifting torsion and lateral oscillations originating in bearing parts of shafts can reduce in a variable operation period of rotating parts with variable loading. Most unfavorably a case when frequency of vibration corresponds to one cycle per revolution within the limits of the effective range. The oscillations (Fig.3) of this type originate amplitudes to change driving torque which time delays of a system and inevitable speedups grow out at a variable rotation speed. Such conditions exist in any power shaft working at certain point of the joint. In transferring time the intermediate element of a power drive train generates oscillations of the moment the loadings stipulated by frequency drifts of rotation. These oscillations depend on an angle of the joint and slugged parameters of a system. Application of the ability to select layer angle stacking allows to avoid beating on transferring conditions.

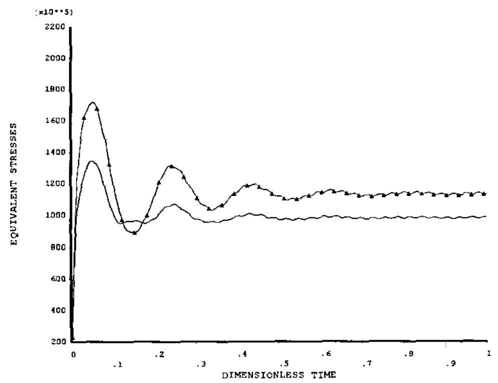

Fig. 4 Equivalent stresses oscillations at the opposite points of the CV joint during end part of beating

On the vertical axis the equivalent stresses (MPa), at horizontal axis dimensionless time is fixed represented acceleration time of the boat on considered engine regime.

Average values of an angular motion and velocity are same for both jointed shafts. It means that driving inner race makes one revolution, and driven bell housing is pivoted on one revolution too. During this revolution current value of the torque and instant values of the force and angular deformations are transmitted through the joint are not stable on. Assuming, that rotation of the driving inner race is pulsative we shall receive, that the driven bell housing will have maximum deviations of output angular deformation concerning driving inner race when driven bell housing has maximum torque subjection in a plane of joint angle, and when it is perpendicular to angular velocity. Angular deformations increased when angle of the joint is increased, and with the much greater engine torque.

Drive shaftings of motorboats are sensitive to excitation of oscillations of this type, and also to disturbance of normal operation generated by it. In that case the motorboat time for transferring on gliding regime should be shorted; power and rpm of the shaft should be more close to nominal 6000rpm. It limits excitation of oscillations of rotative and unrotative elements generated by third mode, and is equivalent to the angle which is not below of limited level at a rotary speed 3600 rpm. The power shafts with small a diameter, also suppose this heightened for value estimation of speedup that is stipulated them concerning small time of transferring. Propulsive plants of plastic motorboat hull cannot work at that vibration levels.

To prevent instability of shafts at the corresponding setting of the driving inner race and driven bell housing it is important to select regime conditions for transferring on the gliding motion to change angle of joint which will provide speedup of each shafts in phase opposition in relation to speedup dependent rpm. In designs of such type the big angles joint on critical conditions of their application the specified suppressing of the beating may be not reached. Selection of the angles of the joint on the transferring is necessary to eliminate or reduce up to a minimum action by supporting bearers of the shaft through moments of the second order. Excitation of oscillations can generate abnormal operation at motion with heavy loading or before gliding. Effects escorting it in many respects are similar to action of coupled oscillations. Most frequently these oscillations originate in a time of operation near critical zone of the third mode, when equivalent stresses is strong arose (Fig.4). At such loadings the elements of the power shaft have a capability for conveyances and effect noise because of beating. Disturbance of normal operation in case of slugged excitation during transferring regime can be significant. As dynamic characteristics of a motorboat depend on resonance frequencies, disturbance of operation to slugged excitation of oscillations is sensitive to an alteration of speed.

In this case driven bell housing under action a couple of forces are reciprocated on the driving inner race of a CV with a circular frequency corresponding to close natural frequency. Reactions at the supports, which usually act in a certain frequency range and will increase to proportionally transmit driving torque, can create disturbance of a normal operation of shafts. Supporting bearers of the shafts of driving shafts of motorboats are sensitive to this kind of disturbances because of plastic hull. The moments of the second order stipulate a resonance of a supporting bearer when the power shaft is twirled from often its natural frequency of bending oscillations corresponding mode shape. At natural frequency damping of a supporting bearer which stable runs on high frequencies during transferring regime is required, and should maintain resonance loadings of the moments of the second order at non coincide rotary speeds. Shafts with supporting bearers of this type have third critical or a natural frequency near 45.486 Hz , hence they are subjected to beating at range of a rotary speed. The transverse vibration of a rotor system driven by a CV joint is analyzed and the effect of the transmitted torque on the dynamic stability of the system evaluated. As a result of the analysis, the following facts are proved: when the driving shaft and driven shaft are included, both parametric and self-excited vibrations arise due to torque.

4. Conclusions

The beating on third mode (45.485) of

the composite shafting connected by constant velocity joint and having assigned geometrical and mechanical properties. The created beating is result of the excitation by second order bending moment when driving shaft is inclined of angle around thirty degrees and overlapped frequencies of the shafting third mode. The beating of the shafting is not created on the transferring regimes if layer angle stacking is directed on the ±60 degrees forming elastic properties to avoid overlapping of the second order moment frequency and third mode of the shafting inside wide band.

Transferring on gliding of the motor boat having composite shafting with angle stacking of the layers ±45 degrees is accompanied by beating on the second order moment frequency closed to the third mode and to avoid that circumstance it is necessary to execute transferring on rpm close to nominal.

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