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Research article

A recent approach to the research on axial oscillation of shaft system on ships

Van Tu Hoang, Ngoc Thanh Huynh*, Quoc Toan Tran

ABSTRACT

In the design of ship propulsion system, the study of oscillation is a great importance because they play a decisive role in the operating ability of the shaft system. Among ship shaft oscillations, axial oscillation has been only a particular interest after applying on ships for diesel engines with ratio of piston stroke and its diameter 3-4,4. In this paper, the special features of the computational model are presented when studying the axial oscillations on the ship shaft system. It should be emphasized that the system of differential equations in describing the free and forced oscillations can be obtained directly from the analogous equations of the torsion oscillation. However, unlike the torsional oscillation, the axial oscillation of the ship shaft system directly affects the hull through the thrust bearing. In order to solve with the problem of determining the forced and resonant axial oscillations based on the Runge-Kutta numerical method is done by directly integrating the system of 2nd order differential equations in describing the oscillations. Therefore, the paper presents the research results in proposing a procedure to reduce the order of 2nd order differential equations and providing an algorithm to solve the system of differential equations in describing axial oscillations. From that, it will be considered that the mechanism under which forces of variable value are applied to it, axial oscillation occurs. The accuracy in calculating of the axial oscillation depends not only on the accepted method of determining the free and forced oscillation, but also the proper in giving the parameters of the discrete model. If the masses are easy to calculate, the evaluation of the axial deformation of the crankshaft is not univariate because there are many formulas given on the basis of idealizing the crankshaft under the structure of bar. Hence, it is proposed to build a 3D model and determine the deformation of the engine crankshaft components by the finite element method.[1–9]

Key words: shaft system, axial oscillation, discrete model, numerical method

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INTRODUCTION

The shaft system of main marine diesel engine equipment is an elastic system that is subjected to periodic forces and moments during operation. This action of the load causes torsional, bending and axial oscillations. If torsion and bend oscillations have been solved with a methodology so far that allows to calculate reliably and predict the amplitudes and stresses of forced and resonant oscillation, axial oscillation has been one of particular interests in operating practice in the 1970s last century. Perhaps, the engineers of the Fiat Gugliemotti и Maciotta Company[1] were pioneers to approach the study of axial oscillation on the basis of calculating and measuring the ship propulsion system with slow diesel engines. Their research has shown that axial oscillation is dangerous for ship shaft system and the relationship between axial and torsional oscillation. Axial oscillation is especially dangerous after applying on ships that are equipped with diesel engines with a ratio of piston stroke to its diameter 3,0-4,4[2]. In these diesel engines, without coating the journal of crankshaft, the bending stiffness of the crankshaft is reduced, resulting in the radial force applied to the connecting rod the crankshaft is easily bent. The result of the crankshaft deformation in the axial direction increases, so the probability of a dangerous resonance of axial oscillations also raises up. The risk of axial oscillation depends on the level of proximate resonance. When operating the shaft system gradually to the resonant rotation frequency, the variable deformation of the shafts increases suddenly, which can cause shaft fracture, destroy the thrust bearing, and increase the oscillation of the structure of ship hull.

The main disadvantage of the existing methodologies for calculating axial oscillation is that they are based on simplified methods oriented towards manual calculation. Therefore, it is necessary to add harmonic analysis of the forcing forces and then only take into account the partial harmonic (e.g. if the resonant frequency is 4^{th} order, only the 4^{th} harmonic component is taken). All this leads to a significant error between

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the calculated and measured amplitudes of the resonant axial oscillation. To solve with this disadvantage, we apply numerical methods to solve the system of differential equations directly in describing axial oscillation.

METHODS

Mathematical Models

The calculation of torsional and axial oscillations has many common reviews[1]. When calculating axial oscillation, the shaft system on ship is replaced by a discrete model consisting of a collection of concentrated masses that are related to each other by corresponding deformations and properties of oscillated dispersion. Thus, the differential equations in describing the free and forced axial oscillation can be obtained directly from the analogous equations for torsional oscillation, if the angular displacement of the mass is replaced by a linear displacement, the inertial moment of the mass is simply replaced by the masses, and the torsional deformation is replaced by the axial deformation.

Nevertheless, unlike the torsional oscillation, the axial oscillation of the shaft system directly affects the hull through the thrust bearing. The relationship of axial oscillations with the hull on the computational model is described as a branch consisting of two masses m_k of the thrust collar and the mass of the hull. The mass of the hull is much larger than the total mass of the shaft which is considered to be rigidly mounted. The axial deformation between the two these masses is determined by the deformation of the thrust bearing and its housing (C_{n+2}, Figure 1).

The analogous relation of the shaft system with the hull is made by an axial anti-oscillation device. This relationship is created by the structure of the axial anti-oscillation device, it is idealized in the form of a branch with 3 masses (m_2, m_{n+1} as shown in Figure 1) with the computational model, where the outer mass is rigidly coupled.

Dynamic properties of the shaft system – the object of axial oscillation are determined by the time-shifted displacements of the masses that are related to the static balance position under the action of forcing external forces. The most suitable expression of these displacements is to use Lagrange equations of the 2^{nd} type as shown:

$$\frac{d}{dt}\left(\frac{\partial K}{\partial \dot{x}_i}\right)-\frac{\partial K}{\partial x_i}+\frac{\partial \Pi}{\partial x_i}+\frac{\partial \phi}{\partial \dot{x}_i}=T_i\left(t\right), \qquad (1)$$

where K – Kinetic energy of translational moving masses

$$K=\frac{1}{2}\left(m_1\dot{x}_1^2+m_2\dot{x}_2^2+...+m_n\dot{x}_n^2+m_{n+1}\dot{x}_{n+1}^2\right);$$

Π – Potential energy of elastic links between discrete masses

$$\begin{aligned}\Pi=&\tfrac{1}{2}\textstyle\sum_{i=1}^{n-1}C_{i,i+1}\left(x_{i+1}-x_i\right)^2\\&+C_{2,n+1}\left(x_{n+1}-x_2\right)^2+C_{n+1}x_{n+1}^2\\&+C_{n+2}x_k^2;\end{aligned}$$

Φ – The Rayleigh energy dispersion function is expressed as:

$$\phi=\frac{1}{2}\left(B_1\dot{x}_1^2+B_2\dot{x}_2^2+...+B_n\dot{x}_n^2+B_{n+1}\dot{x}_{n+1}^2\right);$$

$T_i(t)$ – Force causing forced axial oscillation, which is a function of time;

m_i, $C_{i,i+1}$, $C_{2,n+1}$, C_{n+1}, C_{n+2} và B_i – The respective masses, stiffness (strain) and dispersion coefficients;

x_i và $\dot{x}_i$ – displacement and velocity of discrete masses.

After some uncomplicated transformations, it is able to get a system of 2^{nd} order differential equations with constant coefficients.

$$\begin{cases} m_1\ddot{x}_1+B_1\dot{x}_1+C_{1,2}\left(x_1-x_2\right)=T_1\left(t\right);\\ m_2\ddot{x}_2+B_2\dot{x}_2+C_{1,2}\left(x_2-x_1\right)+\\ C_{2,3}\left(x_2-x_3\right)+C_{2,n+1}\left(x_2-x_{n+1}\right)=T_2\left(t\right);\\ \dots\\ m_k\ddot{x}_k+B_k\dot{x}_k+C_{k-1,k}\left(x_k-x_{k-1}\right)+\\ C_{k,k+1}\left(x_k-x_{k+1}\right)+C_{n+2}x_k=T_k\left(t\right);\\ \dots\\ m_{n-1}\ddot{x}_{n-1}+B_{n-1}\dot{x}_{n-1}+C_{n-2,n-1}\times\\ \left(x_{n-1}-x_{n-2}\right)++C_{n-1,n}\left(x_{n-1}-x_n\right)=T_{n-1}(t);\\ m_n\ddot{x}_n+B_n\dot{x}_n+C_{n-1,n}\left(x_n-x_{n-1}\right)=T_n\left(t\right);\\ m_{n+1}\ddot{x}_{n+1}+B_{n+1}\dot{x}_{n+1}+C_{2,n+1}\left(x_{n+1}-x_2\right)\\ +C_{n+1}x_{n+1}=T_{n+1}(t). \qquad (2)\end{cases}$$

The system of equations (1) can be briefly written in the form of a matrix:

$$[m]\left\{\ddot{x}\right\}+[B]\left\{\dot{x}\right\}+[C]\left\{x\right\}=\left\{T\right\} \qquad (3)$$

where:

$[m]$ and $[B]$ - diagonal matrix of masses and dispersal coefficients;

$[C]$ - square matrix of the coefficient of stiffness (axial deformation);

$\{x\}$, $\{\dot{x}\}$, and $\{\ddot{x}\}$ - column matrix of extrapolated coordinates, velocity and acceleration;

$\{T\}$ - the column matrix of periodic changed forces.

Starting from (2), getting the differential equations in describing the free oscillations in matrix form as follows:

$$[m]\left\{\ddot{x}\right\}+[C]\left\{x\right\}=0. \qquad (4)$$

The stress in the shaft due to axial oscillation is quite small to use as an evaluation parameter in the research[3]. Hence, it is considered that the only risk of axial oscillation through amplitude, for example, in terms of the free-end amplitude of the crankshaft.

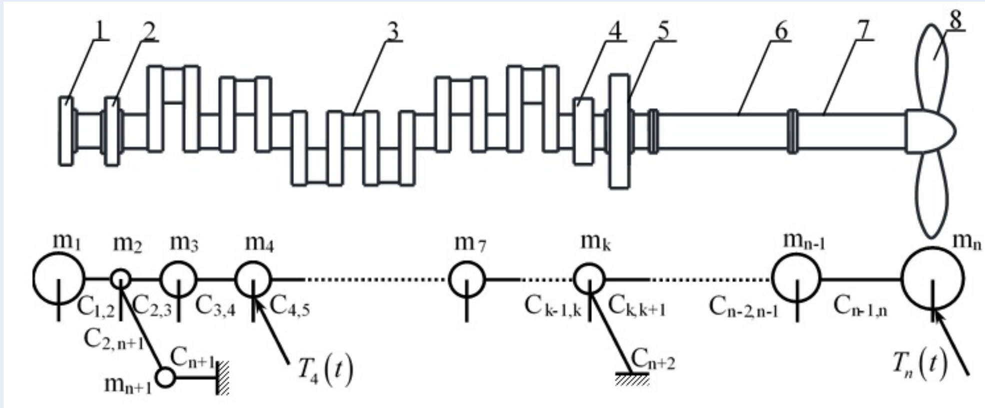

Figure 1: Discrete model of ship shaft system. 1-Torsional anti-oscillation device, 2-Axial anti-oscillation device, 3-Crankshaft, 4-Thrust bearing, 5-Flywheel, 6-Intermediate shaft, 7-Propeller shaft, 8-Propeller.

Determine The Forced External Force Applied To The Shaft System

So far, it has been considered that the structure in which axial oscillation occurs. As mentioned above, the variable deformation of the crankshaft is the cause of the axial oscillation. This deformation is caused by the radial force Z applied to the journal of connecting rod as shown in Figure 2. Bending deformation accompanied by the extension or contraction in the axial direction of the crankshaft, causing reciprocating motion of the shaft elements. The equivalent axial force T of this motion is expressed through the transfer function k_e by the simple relation

$$T = k_e.Z \tag{5}$$

where Z is the force that changes periodically according to the rotation angle of crankshaft determined on the basis of force analysis of the crank mechanism and connecting rod[4].

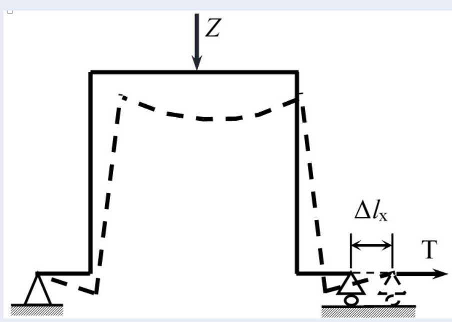

Figure 2: Deformation of crankshaft.

From theoretical viewpoint, the transfer function is calculated using the conventional method. The essence of this method is based on the idealization of the crankshaft in the form of a bar, which means that it does not take into account particularly the structure of the crankshaft, the level of coating the shaft journal, the influence of crank webs to a deformed condition[5]. The finite element method allows to deal with the above drawbacks and solve the problem of determining the axial deformation and the coefficient of transfer function for the crankshaft. To further clarify this proposal, it is considered the crankshaft of diesel 6S50MC-C[3] with the structural parameters shown in Figure 3 and Table 1.

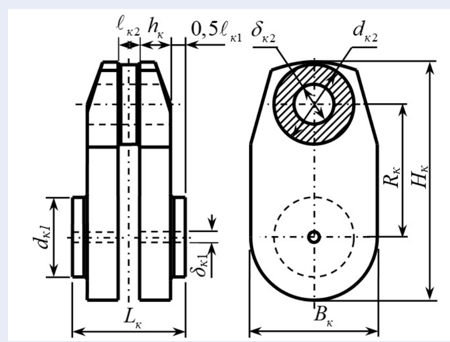

Figure 3: Crankshaft structure of diesel 6S50MC-C[6]

The calculation of the crankshaft deformation on the basis of the finite element method is done, firstly building a 3D model of the crankshaft, and then exporting it to the ANSYS Workbench environment to generate the element mesh and set up the forces as shown in Figure 4a. On the basis of meshing procedure, the obtained finite element model has a type of quadrilateral grid with an average size of 35 mm and a

Table 1: Geometrical parameters and material of crankshaft

Symbol	Unit	Value
d_1	mm	600
δ_1	mm	80
d_2	mm	600
δ_2	mm	300
L	mm	850
h	mm	233
l_2	mm	162
l_1	mm	222
B	mm	950
H	mm	1785
R	mm	1000
E	Pa	$2{,}1.10^{11}$
ρ	Kg/m^3	7850

number of 158075 elements. Then, the axial deformation of the crankshaft under the force value P = 4000 N is determined by the following formula:

$$e_o = \frac{\triangle l_p}{P} \quad (6)$$

where: P - axial force applied to the crankshaft, N; Δlp – deformation caused by axial force P, m.
Deriving from the finite element model, determining the equivalent axial deformation Δl_x caused by the radial force Z = 6000 N applied to the connecting rod journal as shown in Figure 4b. Therefore, the transfer function is determined by the following equation:

$$k_e = \frac{\triangle l_x}{e_0.Z} \quad (7)$$

The variable force is caused by propeller working in uneven water flow that depends on many factors. Among them, the most special feature is the shape of the stern, the arrangement of the propellers, the angle of the shaft, the shape of the blades, the draught, and pitch and roll of the ship. According to [7], propeller thrust is determined by the following equation:

$$T_{cv} = \frac{2T_{tb}\left[1 - V_t.tg\varphi \cos z\alpha / (\omega.R_0)\right]^2}{2 + \left[V_t.tg\varphi / (\omega.R_0)\right]^2} \quad (8)$$

where:
T_{tb} – average thrust of propeller [N];
V_t – ship speed, [knot];
φ - inclination angle of the flow, $\varphi < 20$ radial;
α - rotation angle of propeller;
ω - Angular velocity of propeller;
z - Number of propeller blades;
R_0 - calculated radius of the blade, [m], $R_0 = (0.6 \div 0.65)\, D/2$.
D: Diameter of propeller

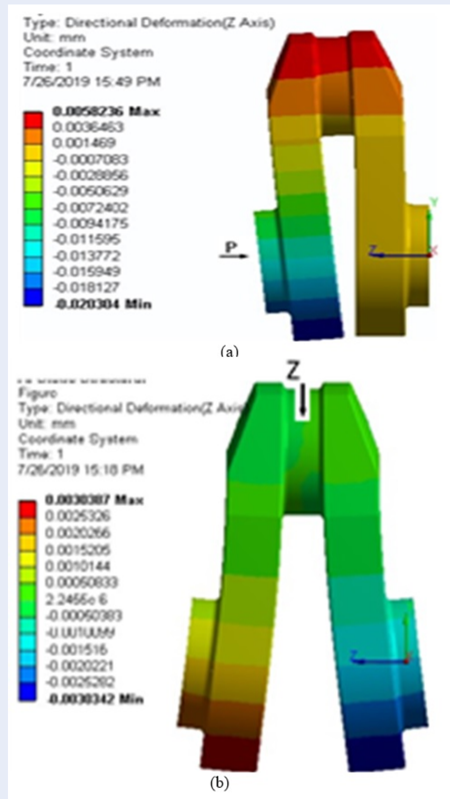

Figure 4: Deformation of crankshaft in ANSYS simulation program.

Calculation Of Axial Oscillation Using The Numerical Method Runge-Kutta

As known, the task of determining the forced and resonant oscillations of ship shafting's is carried out by the classical method – energy method. To implement this method, it is firstly necessary to analyze the forcing functions on the right-hand side of equation system (2) in the form of harmonics, and then take only one harmonic component corresponding to the resonant order, which can be determined by calculating free oscillations. It is clear that the calculation of oscillations by the energy method does not take into account the influence of all other harmonic components, which leads to a significant discrepancy in the calculation results.

The numerical integration method allows directly solve the system of differential equations (2) without the need for harmonic analysis of the forcing forces. In engineering calculations, the most widespread is the fourth-order numerical Runge-Kutta's method that allows us to determine the values of the desired partial functions $y_j = f_j(t)$ at discrete moments of time (space) using recurrent formulas for numerical integration. In this case, each time interval is set equal to the integration step h

$$t_0 = a,\ t_1 = t_0 + h,\ t_2 = t_1 + h, ..., t_{i+1} = (i+1)h \qquad (9)$$

Integrating the differential equations of the Runge-Kutta method are well developed and described in sufficient detail in the reference material[8]. For the one first-order equation, these methods can be used without any corrective changes. And regarding to the second-order differential equations of the numerical method, it was impossible to find the canonical formulas for their solution in the mathematical books. To solve the system of equations (2), it is clear to transform it into a system of the first-order equations (10) using the procedure in lowering the order of the differential equation. The essence of the procedure in lowering the order is as follows, if a second-order differential equation is given:

$$m\ddot{x} + B\dot{x} + Cx = T(t)$$

then using the substitution $x = y_1$ and $\dot{x} = y_2$ it can be reduced to a equation system consisting of two first-order equations:

$$\begin{cases} \dot{y}_1 = y_2; \\ m\dot{y}_2 + By_2 + Cy_1 = T(t). \end{cases}$$

In normalized form, this system of equations will look like this

$$\begin{cases} \dot{y}_1 = F_1(t, y_1, y_2); \\ \dot{y}_2 = F_2(t, y_2, y_2). \end{cases}$$

where F_1 and F_2 - the right-hand sides of the equations, respectively equal to

$$F_1(t, y_1, y_2) = y_2;$$
$$F_2(t, y_1, y_2) = \frac{T(t)}{m} - \frac{C}{m}y_1 - \frac{B}{m}y_2.$$

On the basis of the procedure in lowering the order, the system of equations (1) is lowered to a system consisting of $(2n+2)$ of the first order differential equations in normalized form.

$$\begin{cases} \dot{y}_1 = F_1(t, y_1, y_2, ..., y_{m+2}); \\ \dot{y}_2 = F_2(t, y_1, y_2, ..., y_{m+2}); \\ \dot{y}_{k-1} = F_{k-1}(t, y_1, y_2, ..., y_{m+2}); \\ \dot{y}_k = F_k(t, y_1, y_2, ..., y_{m+2}); \\ \quad ... \\ \dot{y}_{m+1} = F_{+1}(t, y_1, y_2, ..., y_{m+2}); \\ \dot{y}_{m+2} = F_{m+2}(t, y_1, y_2, ..., y_{m+2}); \end{cases} \qquad (10)$$

with initial conditions $t = t_0,\ y_1(t_0) = y_{10},\ y_2(t_0) = y_{20},\ ...,\ y_{m+2}(t_0) = y_{m+2,0};$

where $F_1,\ F_2, ...,\ F_{m+2}$ - the generalized right-hand sides of each equation of system (10), respectively equal to

$$F_1(t, y_1, y_2, ..., y_{m+2}) = y_2;$$
$$F_2(t, y_1, y_2, ..., y_{m+2}) = \frac{T_1(t)}{m_1} - \frac{C_{1,2}}{m_1}y_1 - \frac{B_1}{m_1}y_2 + \frac{C_{1,2}}{m_1}y_3;$$
$$F_3(t, y_1, y_2, ..., y_{m+2}) = y_4;$$
$$F_4(t, y_1, y_2, ..., y_{m+2}) = \frac{T_2(t)}{m_2} + \frac{C_{1,2}}{m_2}y_1 - \frac{C_{1,2}+C_{2,3}+C_{2,n-1}}{m_2}y_3; - \frac{B_2}{m_2}y_4 + \frac{C_{2,3}}{m_2}y_5 + \frac{C_{2,n+1}}{m_2}y_{m+1}$$
$$...;$$
$$F_{k-1}(t, y_1, y_2, ..., y_{m+2}) = y_k;$$
$$F_k(t, y_1, y_2, ..., y_{m+2}) = \frac{T_k(t)}{m_k} + \frac{C_{k-1,k}}{m_k}y_{k-3} - \frac{C_{k-1,k}+C_{k,k+1}+C_{n+2}}{m_k}y_{k-1}; - \frac{B_k}{m_k}y_k + \frac{C_{k,k+1}}{m_k}y_{k+1}$$
$$...;$$
$$F_{m-3}(t, y_1, y_2, ..., y_{m+2}) = y_{m-2};$$
$$F_{m-2}(t, y_1, y_2, ..., y_{m+2}) = \frac{T_{n-1}(t)}{m_{n-1}} + \frac{C_{n-2,n-1}}{m_{n-1}}y_{m-5} - \frac{C_{n-2,n-1}+C_{n-1,n}}{m_{n-1}}y_{m-3}; - \frac{B_{n-1}}{m_{n-1}}y_{m-2} + \frac{C_{n-1,n}}{m_{n-1}}y_{m-1}$$
$$F_{m-1}(t, y_1, y_2, ..., y_{m+2}) = y_m;$$
$$F_m(t, y_1, y_2, ..., y_{m+2}) = \frac{T_n(t)}{m_{n-1}} + \frac{C_{n-1,n}}{m_n}y_{m-3} - \frac{C_{n-1,n}}{m_n}y_{m-1} - \frac{B_n}{m_n}y_m;$$
$$F_{m+1}(t, y_1, y_2, ..., y_{m+2}) = y_{m+1};$$
$$F_{m+2}(t, y_1, y_2, ..., y_{m+2}) = \frac{T_{n+1}(t)}{m_{n+1}} + \frac{C_{2,n+1}}{m_{n+1}}y_3 - \frac{C_{2,n+1}+C_{n+1}}{m_{n+1}}y_{m+2};$$

The implementation algorithm of the 4th-order Runge-Kutta's method is lowered to periodical calculations of the value of the function at each step $i+1$, when the i-th step is known, according to the following recurrent formulas

$$y_{j,i+1} = y_{j,i} + \frac{1}{6}\left(K_{1j} + 2K_{2j} + 2K_{3j} + K_{4j}\right) \qquad (11)$$

where:

i - number of integration step, i = 0,1,2,...;

j - number of equations, j = 1,2,...,m = $2n$;

n - number of mass on the main branch of the discrete model;

$K_{1,j},\ K_{2,j},\ K_{3,j},\ K_{4,j}$ - coefficients, calculated by the formulas

$$K_{1,j} = hF_j\left(t_i, y_{1,i}, y_{2,i}, ..., y_{m+2,i}\right)$$

$$K_{2,j} = hF_j \times \left(t_i + \frac{h}{2},\ y_{1,i} + \frac{K_{1,1}}{2},\ y_{2,i} + \frac{K_{1,2}}{2},\ ...,\ y_{m+2,i} + \frac{K_{1,m+2}}{2}\right)$$

$$K_{3,j} = hF_j \times (t_i + \frac{h}{2},\ y_{1,i} + \frac{K_{2,1}}{2},\ y_{2,i} + \frac{K_{2,2}}{2},\ \ldots,\ y_{m+2,i} + \frac{K_{2,m+2}}{2})$$
$$K_{4,j} = hF_j \times (t_i + \frac{h}{2},\ y_{1,i} + \frac{K_{3,1}}{2},\ y_{2,i} + \frac{K_{3,2}}{2},\ \ldots,\ y_{m+2,i} + \frac{K_{3,m+2}}{2})$$

A computer program is written on the basis of the described algorithm that allows to implement a step-by-step cycle according to the 4^{th}-order Runge-Kutta's formula (11). It should be noted that lowering the integration step, the accuracy of solving the system of differential equations increases. In practical calculations, the correction of the choice of the integration step is determined by the half-step method[8], for the task specified below, the integration step is $h = 10^{-4}$.

CALCULATED RESULTS AND DISCUSSION

To clarify the above statements, we will consider the shaft system of an Oil Tanker with a displacement of 47,400 tons and a speed of 15.1 knots. The single-shaft propulsion system directly drives the torque to the propeller as shown in Table 2[5]. It consists of a 2-stroke engine 6S50MC-C[6] as shown in Table 3, a thrust bearing arranged in the engine, an intermediate shaft and propeller shaft, and a fixed pitch 4-blade propeller. There is a device to prevent axial and torsional oscillation on the crankshaft of the free end of the crankshaft. The flywheel is situated on the driving side of the crankshaft.

The discrete model of the shaft system is shown in Figure 5. It consists of 12 masses in the main branch and two rigidly connected branches: the 1^{st} mass idealizes the disc of the torsional anti-oscillation device, the block the 2^{nd} - the toothed disc of the axial anti-oscillation device, the masses 3^{th} to 8^{th} - the crankshaft mass, the 9^{th} - the mass of the rim of the thrust bearing and the driving end of the crankshaft, 10^{th} mass – flywheel, 11^{th} mass - intermediate shaft and propeller shaft, 12^{th} mass - propeller, 13^{th} mass - axial anti-oscillation housing. All masses are related to each other by their respective axial deformation. The parameters of the discrete model are shown in Table 4.

The calculation of the free axial oscillation is carried out by solving the system of equations (3) on the basis of the Cholesky procedure[9] with the aim of determining the resonant frequencies. In this paper, the calculation of the resonant oscillation will be performed in two cases: firstly, there is no anti-axial oscillation device, and then there is this device. The obtained results show that, in all operating modes (Figure 6, curve 2), the resonant axial oscillations are dangerous for the shaft system. The most dangerous mode of the engine is corresponding to the frequency of 114 and 76 rpm (resonance order 4 and 6 respectively; resonance amplitude 6.459mm and 6,112mm respectively) (Figure 7a and Figure 8a). This means that the exploitation of a ship when the axial anti-oscillation device is damaged is not allowed, because in this case the amplitudes of oscillations will exceed the permissible value (the given value of 0.93 mm by the manufacturer) is 6-7 times. When the axial anti-oscillation device works normally, the oscillation amplitude can reach 0.638 mm (Figure 7b and Figure 8b), lower than the permissible value.

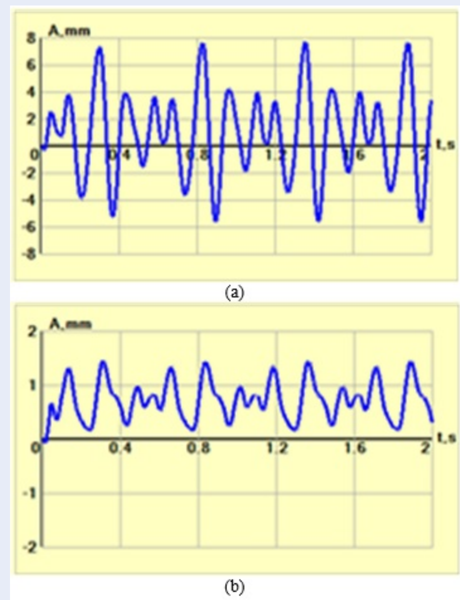

Figure 7: Axial oscillation of the crankshaft free end of 6S50MC-C engine at rotation speed 114 rpm without (a) and with (b) anti-oscillation device.

CONCLUSION

Calculation results show that the resonant axial oscillation of the ship shaft system can be the cause of various damages, causing unpredictable consequences as mentioned above, and especially dangerous for the shaft system which are equipped with slow-speed diesel on ships. Testing the technical condition of the axial anti-oscillation device is a necessary to the normal operation of the energy equipment.

Calculating resonant and forced axial oscillation by the numerical method will solve the limitation of the energy method, since this conventional method only considers a harmonic component of the forcing

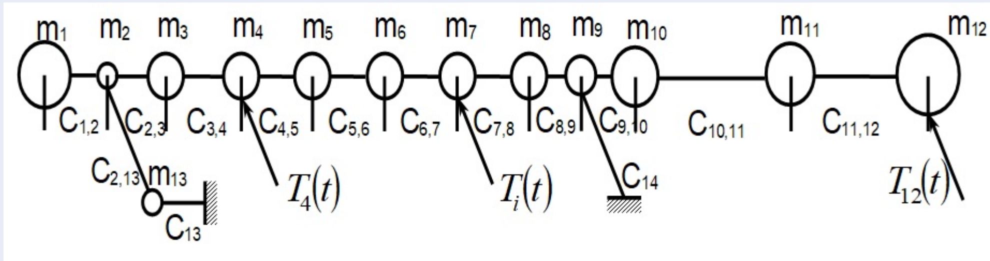

Figure 5: Calculation model of ship shaft system of Oil Tanker

Table 2: Basic parameters of shaft system [7]

Designation	Symbol	Unit	Value
Diameter of crankshaft journal	d_1	mm	600
Diameter of intermediate shaft	d	mm	437
Diameter of propeller shaft	d_{cv}	mm	530
Diameter of propeller	D	mm	5800
Weight of propeller	m_{cv}	kg	13554
Weight of flywheel	m_{bd}	kg	2977
Weight of axial anti-oscillation device	m_{dd}	kg	1000

Table 3: Basic parameters of Diesel Engine 6S50MC [9]

Designation	Symbol	Unit	Value
Rated power	N_{dm}	kW	8310
Engine Speed	n_{dm}	rpm	123
Bore of cylinder	D_{xl}	mm	500
Piston Stroke	S	mm	2000
Number of cylinders	V	-	6
Ratio of Crank radius to Connecting rod Length	λ	-	0,4
Air turbocharged pressure	p_k	MPa	0,365
Pressure at the end of compression stroke	p_c	MPa	13,2
Maximum combustion pressure	p_z	MPa	15,1
Compression ratio	ε	-	15,5
Expansion ratio	ρ	-	1,7
Multivariable compression ratio	n_1	-	1,3
Multivariable expansion ratio	n_2	-	1,34
Crank Radius	R	mm	1000
Mass of reciprocating parts	m_{tt}	kg	3500
Cylinder order	-	1-5-3-4-2-6	

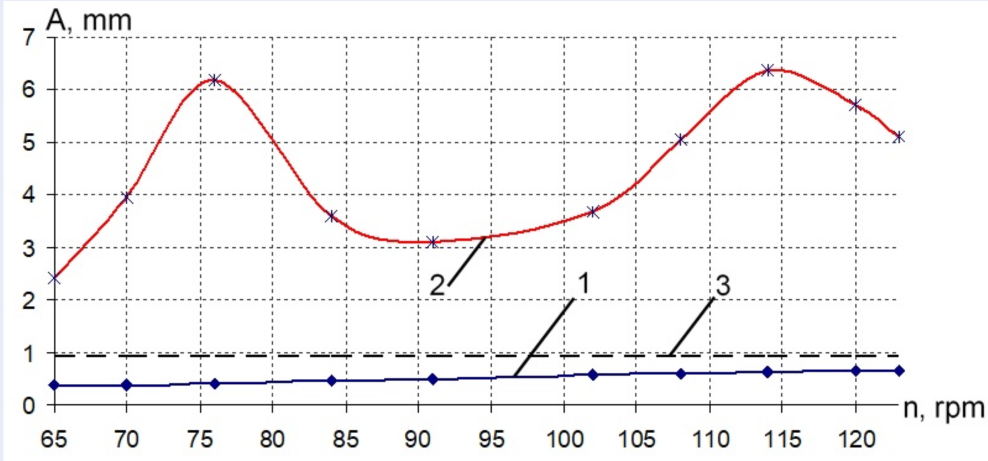

Figure 6: Variable graph of the maximum amplitudes of the axial oscillation for the crankshaft without (2) and with anti-oscillation device (1), (3) – permissible amplitude.

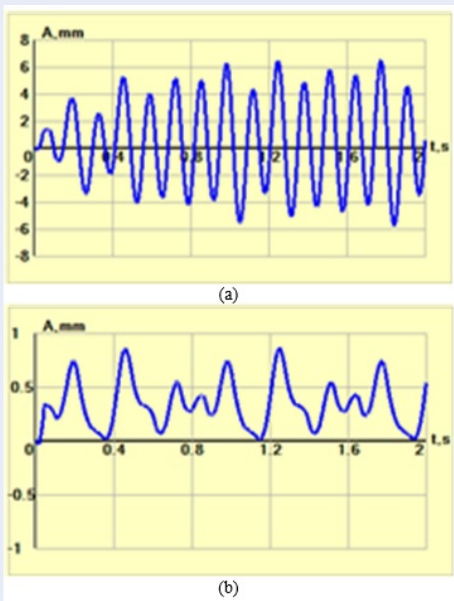

Figure 8: Axial oscillation of the crankshaft free end of 6S50MC-C engine at rotation speed 76 rpm without (a) and with (b) anti-oscillation device.

force. Performing calculations using the Runge-Kutta method allows not only to determine the amplitude of the oscillation, but also to reconstruct the oscillation process over time.

The proposed method of the calculation of axial oscillation can find out a more effective orientation in studying torsional oscillations of the ship shaft system.

CONFLICT OF INTEREST

The authors declare that there is no conflict of whatsoever involved in publishing this research.

AUTHORS' CONTRIBUTION

Van Tu Hoang is the supervisor, contributes ideas for the proposed method and also take part in the work of gathering data and checking the numerical results.
Ngoc Thanh Huynh works as the manuscript editor and contributes the work of gathering data.
Quoc Toan Tran works as the developer of the method and also a manuscript editor.

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Table 4: Parameters of the computational model

Mass	Axial deformation e_j (10^{-9}, $m.N^{-1}$)	Discrete Mass m_j (kg)	Dispersal Coefficient B_j ($kg.s^{-1}$)
1	0,0044	11144,55	-
2		944,18	2600000
	1,4545		
3		5877,8	43356,33
	2,90		
4		5713,2	43356,33
	2,90		
5		5713,2	43356,33
	2,90		
6		5713,2	43356,33
	2,90		
7		5713,2	43356,33
	2,90		
8		6370,82	43356,33
	1,4565		
9		5212,21	4359946
	0,0113		
10		8590,14	-
	0,2023		
11		9129,55	420000
	0,1889		
12		29137,81	7265000
9-14	0,5863	-	-
2-13	100000	-	-
13	5,56	1000	-

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Một cách tiếp cận mới về nghiên cứu dao động dọc của hệ trục tàu thủy

Hoàng Văn Tư, Huỳnh Ngọc Thanh*, Trần Quốc Toản

TÓM TẮT

Trong thiết kế thiết bị đẩy tàu thủy, việc nghiên cứu các dao động có tầm quan trọng lớn vì chúng có vai trò quyết định đến khả năng làm việc của hệ trục. Trong số các dao động hệ trục tàu thủy, dao động dọc chỉ được quan tâm một cách đặc biệt sau khi ứng dụng trên tàu thủy các động cơ diesel có tỷ số giữa hành trình của piston với đường kính của nó bằng 3-4,4. Trong bài báo trình bày tính đặc biệt của mô hình tính toán khi nghiên cứu dao động dọc hệ trục chân vịt tàu thủy. Cần nhấn mạnh rằng, hệ phương trình vi phân diễn tả dao động tự do và dao động cưỡng bức có thể nhận được trực tiếp từ các phương trình tương tự của dao động xoắn. Tuy nhiên, khác với dao động xoắn, dao động dọc của hệ trục tàu thủy tác động trực tiếp đến vỏ tàu thông qua ổ đỡ chặn. Việc giải quyết bài toán xác định dao động dọc cưỡng bức và cộng hưởng dựa trên cơ sở phương pháp số Runge-Kutta được thực hiện bằng cách tích phân trực tiếp hệ phương trình vi phân cấp 2 mô tả dao động. Dẫn ra thủ tục giảm bậc phương trình vi phân cấp 2 và đưa ra thuật toán giải hệ phương trình vi phân mô tả dao động dọc. Xem xét cơ cấu mà dưới tác dụng các lực cưỡng bức có giá trị biến thiên đặt lên nó, dao động dọc xuất hiện. Tính chính xác của tính toán dao động dọc không chỉ phụ thuộc vào phương pháp được thừa nhận xác định dao động tự do và cưỡng bức, mà còn phụ thuộc vào tính đúng đắn của việc đưa ra các tham số của mô hình rời rạc. Nếu với các khối lượng dễ dàng tính toán thì việc đánh giá biến dạng dọc của khuỷu trục không phải đơn trị vì có nhiều công thức đưa ra trên cơ sở lý tưởng hóa khuỷu trục dưới kết cấu dạng thanh. Do đó, đề xuất xây dựng mô hình 3D và xác định biến dạng các thành phần của trục khuỷu động cơ bằng phương pháp phần tử hữu hạn.

Từ khoá: Hệ trục chân vịt, dao động dọc, mô hình rời rạc, phương pháp số

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