ANALYSIS OF TORSIONAL VIBRATION OF A MARINE PROPULSION SYSTEM

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Abstract*-This paper deals with the problem of torsional vibration which causes fatigue of shaft line and ,consequently, fall of the propulsion system. For initializing marine propulsion system, there should be a simple method for calculating vibration. In this paper, a simplified method of determining the torsional natural frequency of the propulsion system has been described considering propeller, flywheel and engine as three individual rotor system that are interconnected by torsional spring , which can be used for preliminary and quick estimation of natural frequency and its relation with other parameters like diameter and shaft material . Natural frequency has also been calculated by FEM method. The calculated natural frequencies has then validated by measurements from real ships which shows promising result. The critical speed and barred speed range has been estimated , and conversion between undercritical and overcritical running system has also been discussed.*

Keywords: Torsional vibration, Natural frequency, Critical speed, Undercritical, Overcritical.

1. INTRODUCTION

Torsional vibration problems are very common in the marine propulsion system which particularly increases with the increasing use of mechanical engine for marine propulsion. The stories about ship shafts snapping became regularly printed on the pages of the newspapers since 1870. A list of ship with a snapped shaft simultaneously raised as the transoceanic shipping of steamer became popular.

Since 1912 when the first ocean-going diesel motor ship SELANDIA (East Asiatic Company) was launched the number of casualties increased very fast. At the beginning of the 20th century, a lot of facts were accumulated to start scientific research on the problem. The main reason for shaft snapping is torsional vibration. So it is very important to consider the torsional vibration for the safety of navigation in the sea. On a typical merchant ship, mainly two-stroke slow speed marine engine has been used in which engines are connected to the propeller by relative short shaft line. For the development of engine in recent years' engines are comprised of less cylinder and smaller engine room but the main problem lies in added vibration due to propulsion system especially for container carrier which is investigated by Andersen and Jensen (2014). Sometimes vibration may produce which is very dangerous for the comfort of the crew and in many ship structures. Fatigue may occur in the shaft line and consequently, fall of the propulsion system. A small failure in the shaft line may need huge money to repair and loss of voyage time. Designing safe marine propulsion system is very important for designer's point of view so many authors have analyzed ship vibration for many complex shapes such as Lin et. al. (2009).

Torsional vibrations of the marine power transmission system are usually most dangerous for the shaft line and crankshaft (DNV Rule Book -2013). So every ship equipped with a reciprocating engine have to examine for torsional vibration especially in the resonance condition. In the case of modeling torsional vibration, the propulsion system can be assumed to be isolated from the hull for the sake of calculation. So the model has no boundary condition. A typical model for the marine propulsion system has been illustrated in Figure 1. Each reciprocating crank is modeled by a single polar mass moment of inertia as well as the intermediate shaft, propeller shaft, and propeller.

Murawski and Charchalis (2014) presented a simplified method for the calculation of the natural frequency and stress generated as there is no easy way to calculate this. They assumed the whole propulsion system to be two-rotor system converting all the components into propeller and engine. In this paper, the whole propulsion system has been considered 3 rotor system considering propeller, flywheel and engine with different shaft diameters. Converting the non-uniform shaft diameter into uniform shaft diameter, a more simplified way of determining natural frequency has been discussed. All the calculated data has been validated by measured data presented by Murawski and Charchalis (2014). This paper also presents some applications of the theory that has used to calculate the natural frequency. The changes in the natural frequency with different shaft diameter and shaft material has been presented. The process of under critical to over critical and overcritical to under critical transformation has also discussed.

2. MODELING METHOD

A typical propulsion system of a merchant ship is made up of slow-speed marine engine connected directly to the propeller by relatively short shaft-line. In case of torsional vibration analysis, all connections between ship and crankshaft – shaft line system are omitted for the sake of simplicity. So, there is no boundary condition. Usually the reciprocating and rotating masses of the engine including crankshaft, intermediate shaft, propeller shaft and propeller are modeled as a system of rotating masses connected by torsional spring. (Engine selection guide two-stroke MC/MC-C engines-2000). The power transmission system model with one degree of freedom in each node is usually sufficient, Murawski and Charchalis (2014). As the first one node mode of torsional vibration is dominant for two-stroke low-speed marine diesel engine, so it has been considered the first one node mode frequency as system natural frequency (Engine selection guide two-stroke MC/MC-C engines-2000).

It has been assumed the marine propulsion system to be equivalent to the three-rotor system considering propeller, flywheel and engine act as three individual rotors, which is demonstrated in Fig. 1(a). The influence of bearing in torsional vibration is insignificant and, consequently, it has been omitted from the shaft-line which is investigated by Batrak (2009), and it has been shown in Fig. 1(b). Torsional stiffness of each crank of the crankshaft is more than ten times greater than the shaft-line stiffness and, therefore, an infinitely stiff crankshaft has been assumed. For simplification, it has been assumed that the material of the propeller shaft and intermediate shaft is the same.

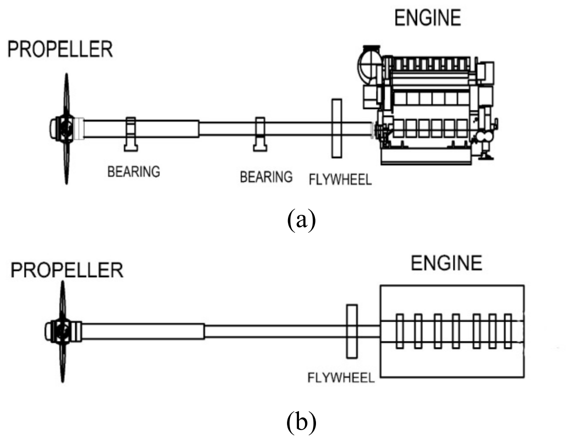

Fig.1: Modeling Marine propulsion system :(a) With bearing, (b) Without bearing.

2.1 Equivalent Uniform Shaft

Figure 1 illustrates a stepped shaft considering engine, flywheel and propeller as three individual disk. In such cases an un-stepped equivalent shaft ought to be used instead of the actual shaft for the purpose of the analysis as shown in Fig.2 as area polar moment of inertia is constant along the whole shaft length in un-stepped shaft. The equivalent shaft must have the same torsional stiffness as the real shaft. Since the torsional springs are connected in series, the equivalent torsional spring can be written as

$$\frac{1}{k_{eq}} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \qquad (1)$$

And using the equivalent torsional stiffness equivalent length and equivalent diameter can be calculated.

$$l_{eq} = l_{eq_1} + l_{eq_2} + l_{eq_3} \ ; \ d_{eq} = \frac{1}{l_{eq}} \sum_{i=1}^{3} \{ l'_i d_i \} \qquad (2)$$

l_{eq_1}, l_{eq_2} *and* l_{eq_3} are the equivalent lengths of shaft segments having equivalent shaft diameter d_{eq} ,and l_{eq} is the total equivalent length of un-stepped shaft having diameter d_{eq} as shown in Fig. 2.

2.2 Torsional Natural Frequency of the Propulsion System

Uniform Propeller Shaft with engine, flywheel and propeller as disc 1, disc 2 and disc 3 respectively with their corresponding polar mass moment of inertia which is demonstrated in Fig. 2(a). Here, l_1 the length of shaft segments between disc 1 (propeller) and disc 2 (flywheel), l_2 the length of shaft segments between disc 2 (flywheel) and disc 3 (engine) and I_{pi} the polar mass moment of inertia of a disc (with *i*= 1, 2, 3). It is a three disc rotor system .So, three natural modes with corresponding three natural frequencies would be occurred according to rotor dynamics.

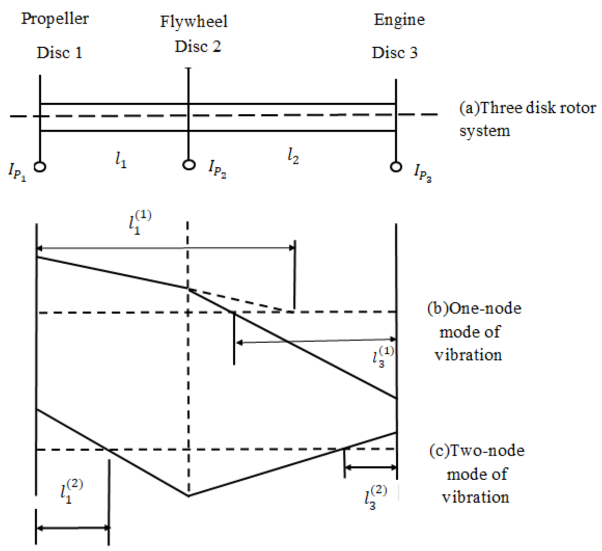

Fig.2: (a) Three disk rotor system , (b) One-node mode of vibration, (c) Two-node mode of vibration

Let $l_1^{(2)}$ be the distance of one node from disc1, and $l_3^{(2)}$ the distance of the other node from disc 3 for two-node vibration (Fig. 2 (c)). The three-disc free-free rotor system could be considered as three single-DOF rotor systems in the particular node system. The torsional natural frequency of the single-DOF cantilever system which consist of disc 1 and shaft length $l_1^{(2)}$ is given as

$$\omega_{nf_1}^{(2)} = \sqrt{\frac{GJ}{l_1^{(2)} I_{p1}}} \qquad (3)$$

Similarly, for another single-DOF cantilever system (Fig. 2(c)) is given as

$$\omega_{nf_3}^{(2)} = \sqrt{\frac{GJ}{l_3^{(2)} I_{p3}}} \tag{4}$$

For the single-DOF fixed-fixed rotor system (Fig. 2(c)) which consist of disc 2 and shaft lengths at either sides of the disc as $(l_1 - l_1^{(2)})$ *and* $(l_2 - l_3^{(2)})$, the torsional natural frequency is

$$\omega_{nf_2}^{(2)} = \sqrt{\frac{k_{t_2}^{(2)}}{I_{p2}}} = \sqrt{\frac{\mathrm{GJ}(l_1 - l_1^{(2)} + l_2 - l_3^{(2)})}{(l_1 - l_1^{(2)})(l_2 - l_3^{(2)}) I_{p2}}} \tag{5}$$

As for a particular mode all frequencies $\omega_{nf_1}^{(2)}$, $\omega_{nf_2}^{(2)}$ and $\omega_{nf_3}^{(2)}$ must be equal, the natural frequency of the two-node or one-node mode could be calculated after knowing the node positions. On equating equations (3) and (4),(4) and (5)

$$l_1^{(2)} = \frac{l_3^{(2)} I_{p3}}{I_{p1}} \tag{6}$$

$$\frac{1}{l_1^{(2)} I_{p1}} = \frac{(l_1 - l_1^{(2)} + l_2 - l_3^{(2)})}{(l_1 - l_1^{(2)})(l_2 - l_3^{(2)}) I_{p2}} \tag{7}$$

Equation (6) can be used to eliminate $l_1^{(2)}$ from equation (7), and it get simplified to

$$\left\{\frac{I_{p2} I_{p3}}{I_{p1}} + \frac{{I_{p3}}^2}{I_{p1}} + I_{p3}\right\} (l_3^{(2)})^2 - \left\{\frac{I_{p2} I_{p3} l_2}{I_{p1}} + I_{p2} l_1 + I_{p3}(l_1 + l_2)\right\} l_3^{(2)} + \{I_{p2} l_1 l_2\} = 0 \tag{8}$$

The quadratic equation (8) gives two values of $l_3^{(2)}$, and the values give the position of nodes for both one-node and two-node torsional vibration. Then the actual frequencies can be calculated by using the two values in equation (6). Only one of these two values of $l_3^{(2)}$ may give the position of a real node, while the other gives the point at which the elastic line between discs 1 and 2, when produced, cuts the axis of the shaft (as shown in Fig. 2 (b) by the dotted line).

The moment of inertia of the vibrating propeller in the water (I_{p1}) is comprised of the mass moment of inertia of the propeller in air (I_a) and added moment of inertia of the entrained water (I_w). I_a Is generally determined by designer. Otherwise it may be estimated by well-known empirical equation that showed in equation (9). Several empirical formulas are used to describe I_w value. According to Dien and Schwanecke, that showed in equation 10.

$$I_a = 4D^5 \tag{9}$$

$$I_w = D^5 \rho \left[CJ_1 + CJ_2 \frac{A_e}{A_0} + CJ_3 \frac{P}{D} + CJ_4 (\frac{A_e}{A_0})^2 + CJ_5 (\frac{P}{D})^2 + CJ_6 \frac{A_e}{A_0} \frac{P}{D}\right] \tag{10}$$

Where : I_w the inertia of entrained water(kgm^2), D the propeller diameter (m), ρ the specific mass of sea water(usually 1025 $\mathrm{kg/m}^3$),
$\frac{A_e}{A_0}$ the expanded blade area ratio,
$\frac{P}{D}$ the propeller pitch ratio ,
Z the blade number, $\mathrm{CJ_i}$ the Coefficients given in Table 1

Table 1: Coefficients for propeller inertia of entrained water.

CJ_i	Z= 4	Z= 5	Z=6
CJ_1	0.00303	0.00278	0.00237
CJ_2	-0.00808	-0.00716	-0.00629
CJ_3	-0.00407	-0.00373	-0.00306
CJ_4	0.00341	0.00305	0.00275
CJ_5	0.00043	0.00046	0.00023
CJ_6	0.00997	0.00853	0.00736

Polar moment of inertia of flywheel $I_{p2} = \mathrm{M}K^2$ (11)

Where: M the mass of flywheel, K the radius of gyration (r) .

2.3 Natural Frequency by FEM Method

In order to find the natural frequency by Finite Element Equation (12) has been used which is the elemental equation and derived by Galerkin method. If there is a rigid disc in the rotor system then generally we put a node at the disc and the mass (or the polar mass moment of inertia) of the disc is considered to be lumped at that node and the lumped mass appears in the diagonal of the mass matrix corresponding to that nodal variable at which it is lumped.

$$[M]^{(e)}\{\ddot{\theta}\}^{(ne)} + [K]^{(e)}\{\theta\}^{(ne)} = \{T_R\}^{(ne)} + \{T_E\}^{(ne)} \quad (12)$$

Where , $[M]^{(e)} = \frac{\rho J l}{6}\begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix}$

$[K]^{(e)} = \frac{GJ}{l}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}$

Where $[M]^{(e)}$ is the element mass matrix , $[K]^{(e)}$ is the element stiffness matrix , $\{T_R\}^{(ne)}$ is the nodal reaction torque vector of the element and $\{T_E\}$ is the nodal external torque vector of the element.

Natural frequencies could be obtained by using solution of the Eigen value problem using equation (13).

$$([A] - \lambda\{I\}\{\theta\} = 0) \quad (13)$$

$$[A] = [K]^{-1}[M] = \left(\frac{GJ}{l}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}\right)^{-1} \frac{\rho J l}{6}\begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix}$$

$$\lambda = \frac{1}{\omega_{nf}^2}$$

Equation (13) has been used in the MATLAB code to determine the natural frequency.

3. RESULTS AND DISCUSSION

The methodology presented has been verified by a 2600 TEU container ship power transmission system. Propulsion system of the ship is equipped with a slow-speed, two-stroke, seven-cylinder main engine: MAN B&W7 S70 MC-C type. The engine main parameters are as follows : Power- 21,735 kW , nominal speed -91 rpm , Flywheel -14,455. kg m^2 . The propulsion system has five-blade propeller having 7.42 m diameter, the mass in air 33,700 kg and polar moment of inertia of air 107,200 kgm^2. The diameters of the intermediate shaft and the propeller shaft are 595 mm and 675mm respectively. The total polar moment of analyzed crankshaft is 176063 kgm^2 . Intermediate shaft length 7.5 m (flywheel is 2m away from engine) and propeller shaft length 8.5 m , shaft material is Aqualoy 19.

Propeller added water mass moment of inertia is 36262 kgm^2 according to equation (10).Then after using other equations one-node natural frequency becomes 30.7 rad/s, and two-node natural frequency becomes 202.62 rad/s. Natural frequency has been calculated both by FEM and with infinite crankshaft, which is demonstrated in Table 2 ,and later comparison with the actual measured data and Independent design office data has been made.

Table 2: Natural frequency

Independent design office	4.858 Hz
Measurements	5.03±0.1 Hz
Natural frequency with infinite crankshaft	4.93 Hz
FEM	4.91 Hz

The error relative to measurements is 1.99% and relative to independent design office is 1.48 % when infinite crankshaft is used. The error relative to measurements is 2.39% and relative to independent design office is 1.07% when FEM is used.

4. APPLICATION

The propulsion system that have been discussed before is a overcritical system as in overcritcal condition one-node resonance vibration with the main critical order should occur about 30-70%(54% in our case) below the nominal engine speed. In undercritical condition, it remains about 35-45% above the nominal engine speed. If diameter increase , natural frequency will also increase and after increasing a significant amount of diameter ,the natural frequency increase to such amount that it causes undercritical condition, which has been shown in Fig.3. In 1.10m diameter, the critical oreder is 38% above the nominal engine speed and it becomes undercritical condition .

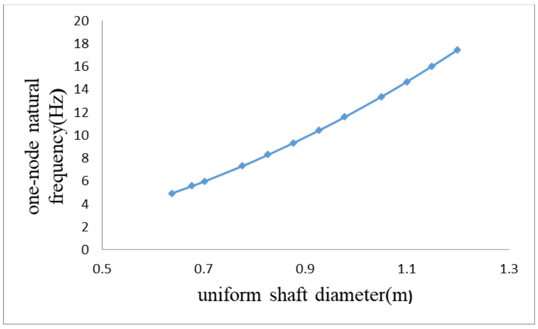

Fig.3: Response of natural frequency with shaft diameter

Natural frequency due to torsional vibration is proportional to modulus of rigidity. The relationship between them has been illustrated in the propulsion system and showed in Fig.4 .Various shaft materials with their natural frequencies have been calculated and demonstrated in Table 3.

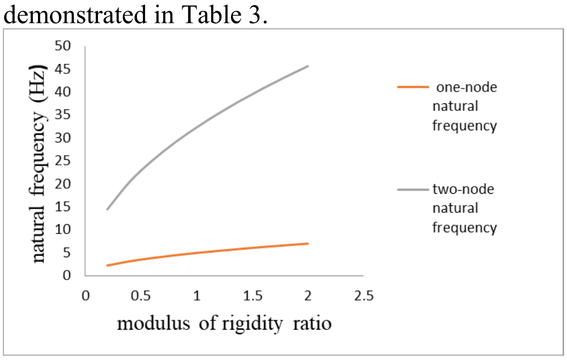

Fig.4: Response of natural frequency with modulus of rigidity

Table 3: Natural frequencies for different shaft materials

Material Name	Modulus of Rigidity(GPa)	Natural Frequency(Hz)
Monel k-500	66	4.58
Aqualoy22	72.9	4.82
Aqualoy19	76.14	4.93
Duplex 2205	76.9	4.95
Aqualoy17	77.2	4.96
Steel C1018	80	5.05

From the natural frequency, critical speed can be calculated which is 41.88 rpm. Engine should not be run in this speed as maximum torsional stress occur at this speed. Propeller damping is dominant in the marine propulsion system. MAN B&W assumed that undimensioned damping factor (Q_R) is equal to 18 for fixed pitch propellers.

Damped Magnification factor, $$Q_\omega = \frac{1}{\sqrt{\left(1-\left(\frac{\omega_e}{\omega_n}\right)^2\right)^2 + \frac{1}{Q_R^2}\left(\frac{\omega_e}{\omega_n}\right)^2}}$$

The vibration magnifier has been described corresponding to different engine speed in Fig.5.

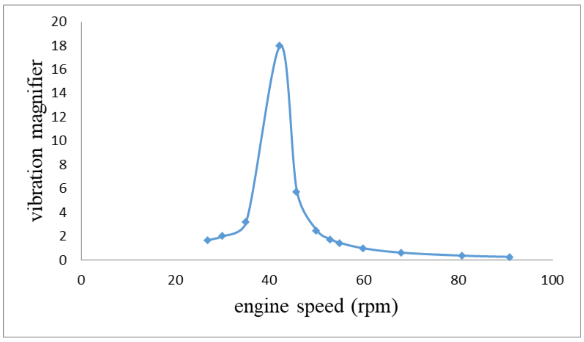

Fig.5: Damped vibration magnifier

5. CONCLUSIONS

There is a lack of simplified method for calculating torsional vibration. Hence, a simplified calculation of torsional vibration has been presented and later to reveal torsional vibration resonances, excitation frequencies are to be compared with propulsion system torsional vibration natural frequencies. Here, the calculated natural frequency is up to 2.39 % deviated from the measured natural frequency. It may be occurred due to neglecting shaft moment of inertia, assumption of the infinite crankshaft, which was done for simplification. The maximum torsional stress occurs at critical speed, which can be determined from natural frequency, and the barred speed range also can be estimated. The natural frequency was 4.93 Hz, and for this natural frequency, the critical speed is 42.24 rpm, which is 54 % below the nominal engine speed. Therefore, it is an overcritical case. After that, a process has been discussed for the transformation of the overcritical system to the undercritical system by increasing the shaft diameter to a considerable amount.

However, this research can further be developed by using the natural frequency to find out torsional stress behavior, especially in the resonance condition. To find out the natural frequency meticulously, finite stiffness crankshaft can be considered. Selection of shaft material and shaft diameter for a specific propulsion system, considering the stress and diameter criteria by DNV Rule book, can also be discussed in further research.

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6. NOMENCLATURE

Symbol	Meaning	Unit
k	Torsional stiffness	(Nm/rad)

G	Shear modulus	(Pa)
J	Polar moment of inertia	(m^4)
I_{p_1}	Propeller polar mass moment of inertia	(kgm^2)
I_{p_2}	Flywheel polar mass moment of inertia	(kgm^2)
I_{p_3}	Engine polar mass moment of inertia	(kgm^2)
I_a	Polar mass moment of inertia of the propeller in air	(kgm^2)
I_w	The added moment of inertia of the entrained water	(kgm^2)