Vibration Analysis of A 3-Bladed Marine Propeller Shaft for 35000DWT Bulk Carrier

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***Abstract*—One of the most dreaded problems on board ships is high level of vibration. The propeller shafts of Bulk Carriers are subjected to dangerous resonance. This dynamic excitation is usually transmitted through the propeller shaft to the rest of the ship. Some ships have been disposed of for years due to objectionable level of vibrations, subjecting them to precarious operation. This thesis aims at analysing the vibration of a 3-bladed marine propeller shaft for 35000DWT bulk carrier. The objectives of the work were to mathematically design the 3-bladed propeller shaft, carrying out computer aided design of the shaft and numerically performing vibration analysis. The methodology used was the application of Solidworks and Analysis System (ANSYS) softwares to design and calculate the natural frequency of the shaft. The mathematical design gave a Hub (boss) diameter of 0.17m. The hollow shaft had external and internal diameters of 0.10m and 0.09m respectively. Also, torques of 202Nm^2 and 384.72Nm^2 were obtained at the driver and driven shafts respectively. The calculated natural frequency was 249Hz while that of the ANSYS was 280Hz which gave an error of 12%. However, the numerical analysis carried out with ANSYS software also showed that a phase difference of 180^0 occured at the frequency of 280Hz which is a signal of possible misalignment of shaft. At this frequency, the displacement of the shaft had a maximum value of 7.87×10^{-6}m. Reaction forces from the components of the shaft were also observed to have contributed to the vibration of the propeller shaft. These reaction forces, which cause wearing of the stern tube and intermediate bearings due to friction, are represented by phase angles closer to 0^0. Wear due to friction is a major source of shaft misalignment. In view of the aforementioned results, it is pertinent to recommend that Bulk Carrier should never be operated at the natural frequency of the engine and the critical speed arrangements should always be carried out from preliminary design stage to operational stages.**

***Index Terms*—ANSYS, Shaft Misalignment, Ship Propeller, Bulk Carrier Vibration.**

I. Introduction

According to Galloway [1], "Mechanical vibrations are the continuing motion, repetitive and often periodic, of a solid or liquid body within certain spatial limits". Vibration occurs frequently in a variety of natural phenomena which include tidal motion of oceans, rotating and stationary machinery, buildings and automobiles [2]. Vibrations can be desirable or not. For example, mining operations rely on sifting vibrations to sort out different size of particles. In geology, vibrations are employed to simulate earthquakes for geological investigations [3]. It can also be used for drilling of geo-technical wells. In agriculture, vibration is used for harvesting by forced vibrations of fruit bearing trees. On the other hand, vibration can be detrimental to both humans and machines. Many structures, buildings and bridges fail because of vibration. Vibration causes rapid failure of machine parts such as bearings, gears, shafts, etc. Passengers of automobiles become uncomfortable when vibration of the vehicle is in excess. One of the most common and dreaded problems on board ships is high levels of vibration. In the past, there have been ships that have been disposed of for years due to objectionable level of vibrations, subjecting them to precarious operations. The dynamic excitation generated by the propeller is usually transmitted through the propeller shaft to the rest of the ship. Although, over the years, with research by classification societies, a lot of improvement has been achieved in terms of ship vibrations. Design techniques [4] have undergone advancement to amalgamate factors related to minimization of vibration levels during the entire lifetime of a ship.

A propeller shaft (Cardan Shaft or Driving Shaft) is correlated with a mechanical component which is used for transmitting torque and rotation. It is a well-known fact that the propeller shafts of Bulk Carriers which are driven by internal combustion engines are subjected to dangerous resonance phenomena of torsional vibrations. This work aims at designing a 3-bladed propulsion shaft for 35000DWT bulk carrier in order to ascertain the cause of dangerous vibration using computer aided design softwares such as Solidworks and Analysis System (ANSYS) and to proffer solution.

II. Literature Review

Durfy [5] worked on Investigation of damping treatments for propeller shaft vibration in which Finite Element Analysis was used. The software used for this analysis was ANSYS. The material element used was elastic 8 node shell with a thickness of 0.00189mm to create the shaft. The shaft was divided into 40 sections along the z-axis and 6 sections radially and was meshed using quad mapped mesh. Fig. 1 shows the model created with ANSYS.

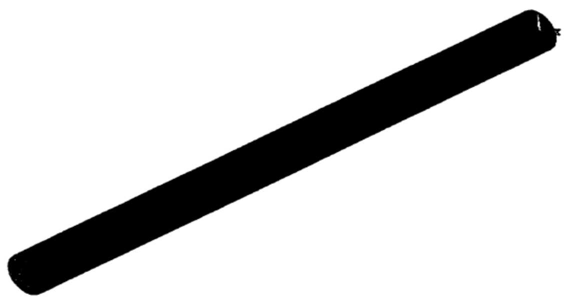

Fig. 1. Shaft model created with ANSYS software

Two more element types were also created; the first being a mass element of 0.691kg and then a 3-D link. The mass

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element was created on a node in the middle of each end of the shaft. A link element was created from the mass element to every circumferential node at the ends shaft. The link was assigned a small cross-section and a large stiffness value. This mass and link combination was an approximation of the end caps on the shaft. No load was applied since the shaft was to be tested in the free-free condition. The modal analysis was performed using Block Lanczos method for the frequency range of 50-800Hz. The results obtained are tabulated as shown in Table I.

In order to expatiate on the effect of longitudinal vibration on marine propulsion system, it was confirmed that axial

vibrations occur in the mass-elastic system comprising the propeller, main propulsion shafting and the propulsion machinery as a result of periodic variation in propeller thrust produced by the non-uniformity of the wake in which the propeller operates [6]. Some forced vibrations exist to some extent in all propeller drives but have required special attention only in recently constructed vessels of relatively high power in which, some cases, large exciting forces and resonance in the shafting system combine to make the vibration critical. The method of fixed end approximation was used to find the modes of the natural frequencies as shown in Fig. 2.

TABLE I: ANSYS RESULTS OF PROPELLER SHAFT VIBRATION ANALYSIS IN FREE-FREE CONDITION

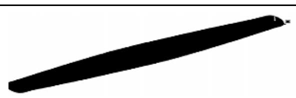
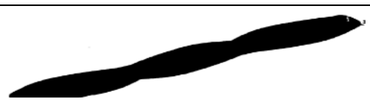
Mode Shapes	Frequency (Hz)	Graphic Results
First Bending (First root)	176	
First Bending (Second root)	176	
First Breathing and First Bending	392	
First Breathing and Second Bending	425	
Second Bending (First root)	523	
Second Bending (Second root)	524	
First Breathing and Third Bending	533	

Fig. 2. Fixed end approximation

The formula of the natural frequency is as shown in (1).

$$f = \frac{F_R}{2\pi}\sqrt{\frac{\bar{k}}{m_p + \frac{\bar{m}}{3}}} \tag{1}$$

where:

f = frequency in Hz

$\bar{k} = \frac{EA}{l}$ kg per m. A being the shafts section area in m^2

l = Equivalent length of shaft from the propeller to bull gear in m.

m_p = Mass of propeller plus virtual mass of surrounding water (the latter may be taken as 60% of propeller mass)

$\bar{m}$ = mass of shaft

F_R = Flexibility factor depending on thrust bearing foundation stiffness

Flexibility factor is the ratio of rotation of a shaft bend to that of a straight shaft of same diameter, wall thickness and length under the action of the same bending moment [7].

Values of flexibility factor required to make the first mode frequencies computed by equation 1 agreed with the observed critical frequency for four propulsion systems of North Carolina (BB-55) and South Dakota (BB-57) ships as shown in Table II.

TABLE II: FLEXIBILITY FACTORS OBTAINED BY FIXED END APPROXIMATION METHOD

Class	Propulsion unit	First mode frequency observed (CPM)	k $\times 10^{-6}$	m_p	M	F_R
BB55	S.O	520	1.72	141	380	0.68
	S.I	600	2.20	144	295	0.66
	P.I	680	2.86	145	228	0.62
	P.O	600	2.09	140	313	0.68
BB57	S.O	520	1.78	161	480	0.73
	S.I	650	2.63	150	333	0.68
	P.I	700	3.43	150	262	0.61
	P.O	600	2.11	161	410	0.75

Fig. 3 shows the graph of flexibility factor against shaft length. It was maintained that the flexibility factors obtained vary somewhat with shaft length since the effect of the thrust flexibility will be greater on a short shaft than on a long one.

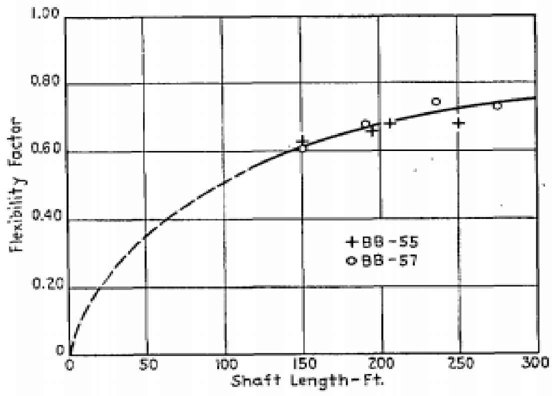


Fig. 3. Flexibility factor versus shaft length

The dominating uncertainty with regard to longitudinal vibration is the stiffness of the main thrust bearing and its foundation [8]. The thrust bearing foundation bends in response to the thrust transferred through the thrust bearing. Fig. 4 shows a three-mass model which can be used to estimate the first two shafting system modes. Three masses

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are considered the minimum number needed for estimating the first two system modes with reasonable accuracy.

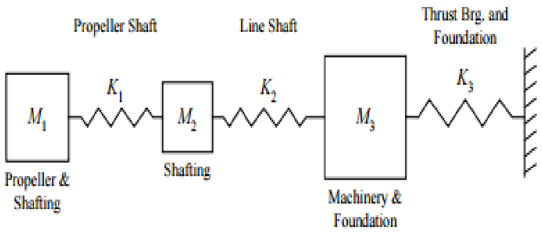

Fig. 4. 3-Mass longitudinal model of main propulsion system

M_1 = Lumped mass at the propeller in Kg, composed of propeller mass, increased 60% for hydrodynamic added mass, one half of the propeller shaft weight

K_1 = Stiffness of propeller shaft in N/m, from propeller to coupling with line-shaft = $\frac{AE}{l}$

A = Shaft cross-sectional area in m^2

M_2 = lumped mass at propeller shaft or line shaft coupling in Kg, composed of one-half of propeller shaft mass plus one half of line-shaft mass (For very shot propeller shafts, M_2 could be located in the line-shaft with mass and stiffness contributions appropriately adjusted).

K_2 = Stiffness of line-shaft in N/m, from coupling to thrust bearing (thrust bearing assumed to be at aft end of engine casing

M_3 = lumped mass at thrust bearing in Kg, composed of one-half of the line-shaft mass, the engine, including the thrust bearing, plus a thrust bearing or engine foundation structural weight allowance of 25%

K_3 = Stiffness of thrust bearing elements and engine foundation in N/m

It was convenient to view the foundation stiffness K_3, as an unknown, which is to be determined so that the two natural frequencies lie at appropriate levels with respect to the blade-rate excitation frequency. The coupled equations of motion lead to the Eigen value problem as shown in (2).

$$\begin{bmatrix} -\omega^2{}_n M_1 + K_1 & -K_1 & 0 \\ -K_1 & -\omega^2{}_n M_2 + K_1 + K_2 & -K_2 \\ 0 & -K_2 & -\omega^2{}_n M_3 + K_2 + K_3 \end{bmatrix} \begin{bmatrix} \psi_{n1} \\ \psi_{n2} \\ \psi_{n3} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \quad (2)$$

Where ψ is the mode shape vector. It is necessary to expand the determinant of the coefficient matrix to form the characteristic equation whose roots are the three natural frequencies. First defining the following for convenience of notation:

$$\Omega_n = \omega^2{}_n, \quad \Omega_{11} = \frac{K_1}{M_1}, \quad \Omega_{12} = \frac{K_1}{M_2}, \quad \Omega_{22} = \frac{K_2}{M_2}, \quad \Omega_{23} = \frac{K_2}{M_3}, \quad \Omega_{33} = \frac{K_3}{M_3}$$

Then the characteristic equation from the determinant expression is as shown in (3) in cubic Ω_n:

$$\Omega^3{}_n + \Omega^2{}_n(\Omega_{11} + \Omega_{12} + \Omega_{22} + \Omega_{23} + \Omega_{33}) - \Omega_n[\Omega_{11}(\Omega_{22} + \Omega_{23} + \Omega_{33}) + \Omega_{12}(\Omega_{23} + \Omega_{33}) + \Omega_{22} + \Omega_{33}] + \Omega_{11}\Omega_{22}\Omega_{33} = 0 \quad (3)$$

The unknown K_3 in Ω_{33} can be calculated for specified distribution of Ω_n in the ranges of interest as shown in (4).

$$\Omega_{33} = \frac{\Omega_n[\Omega^2{}_n + \Omega_n(\Omega_{11} + \Omega_{12} + \Omega_{22} + \Omega_{23}) - (\Omega_{11}\Omega_{22} + \Omega_{11}\Omega_{23} + \Omega_{21}\Omega_{23})]}{-\Omega^2{}_n + \Omega_n(\Omega_{11} + \Omega_{12} + \Omega_{22}) - \Omega_{11}\Omega_{22}} \quad (4)$$

The corresponding value to the required foundation spring constant is given in (5):

$$K_3 = M_3\Omega_{33} \quad (5)$$

K_3 can then be plotted as a function of the arbitrary Ω_n to decide the stiffness of the structure that the shipyard is to design and build to provide the proper support stiffness for the system. On increasing the Ω_n from low values, K_3 will increase and then will become negative as the second mode is reached. The stiffness subsequently turns back to positive again with further increasing frequency in the second mode. The relevant ranges are the ranges of positive K_3 only.

A plot of main machine longitudinal vibration Natural Frequency (FN) is plotted against Foundation Stiffness (Kf) as shown in Fig. 5.

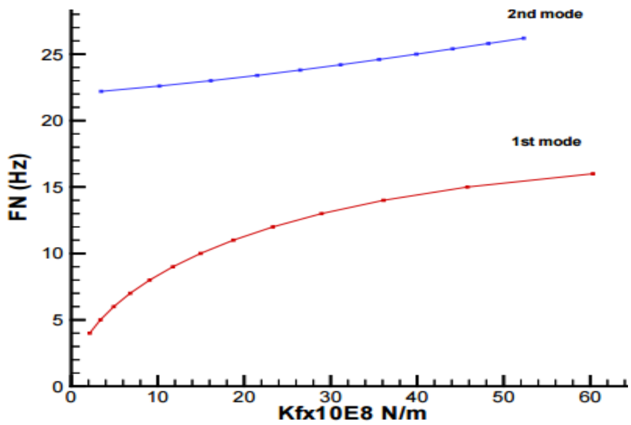

Fig. 5. Foundation Frequency versus Foundation Stiffness.

Any elastically coupled shafts or other system will have one or more natural frequencies. If the shaft is excited the frequency can build up to an amplitude that is perfectly capable to break the shaft. Elastic here is used to refer to a displacement of a twist from rest which creates a force or torque tending to return the system to its position of rest and which is proportional to the displacement.

The equation of frequency of torsional vibration [9] of a single mass is given in equation 6 while that of transverse or axial vibration is given in (6).

$$f = \frac{1}{2\pi}\sqrt{\frac{q}{J}} \quad (6)$$

Where **q** is the stiffness in Nm/rad and **J** is the moment of inertia of the attached mass in Kg/m^2:

$$f = \frac{1}{2\pi}\sqrt{\frac{s}{m}} \quad (7)$$

Where **s** is the stiffness in Newton per meter of deflection and **m** is the mass attached in kg. The essence of control is to adjust these two parameters, **q** and **J** (or **s** and **m**) to achieve a frequency which does not coincide with any of the forcing frequencies.

Two degree of freedom approximation was also used to determine the natural frequency of the propulsion system.

The system is represented as shown in Fig. 6.

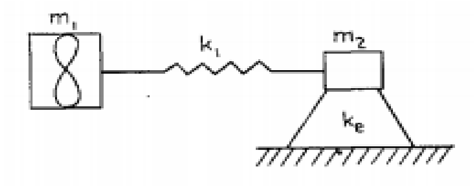

Fig. 6. Two degree of system approximation

The four quantities used in the frequency calculation are defined as follows

m_1 = Mass of the propeller plus the virtual mass of the entrained water plus one-half the mass of the shaft between the propeller and the bull gear. The entrained water may be taken as 60 percent of the propeller mass.

m_2 = Effective mass of the entire propulsion machinery at the level of the shat. It includes the mass of gears, turbines, wet condenser, one-fourth of the mass of machinery foundations and one-half of the mass of shaft between the propeller and the bull gear

k_1 = Static spring constant of the shaft and is equal to $\frac{El}{A}$ where A is the average section area and l = Equivalent length from propeller to bull gear (kg per m).

k_e = Effective fore-and-aft spring constant of the combined turbine, gear and thrust bearing foundations referred to as the shaft level. The natural frequencies were obtained by solving for the positive roots of ω [3].

Equation (7) is quadratic in ω^2. When values are substituted, it can be solved for ω^2 by ordinary quadratic formula and then the two positive square roots of ω^2 can be found.

$$\omega^4 - \omega^2\left(\frac{k_1}{m_1} + \frac{k_1 k_e}{m_2}\right) + \frac{k_1 k_e}{m_1 m_2} = 0 \quad (8)$$

Some possible causes and consequences of torsional vibration problems include the following.

i. Non-uniform operation of dissimilar cylinders as a result of wrong injection timing, improper operation of turbocharger, incorrect valve timing or maybe excessive wear of the piston or its liners; which may induce torsional stresses.
ii. Damage to propeller blading can also causes vibrations being set up in the line shafting thereby causing overheating of the shaft bearings and main thrust.
iii. Lack of engine performance checking and erroneous application of viscous dampers can cause vibration.
iv. Wear of sleeve bearings or a defective rolling element in a bearing can also result in vibrations.
v. Misalignment of shaft and propeller imbalance can cause forces at a frequency equal to the shaft revolutions
vi. Inertia forces on moving parts such as propeller, gears and crankshafts, etc.

The consequences of torsional vibration include but not limited to the following:

i. Torsional oscillation generates high shear stresses leading to fatigue failure of shafts.
ii. Loosening of foundation can damage supporting structure.
iii. Increased stresses on propulsion shaft components particularly with increased load on shaft bearings and this can result in reduced bearing life.
iv. Continued exposure to vibration leads to fatigue and decreased efficiency.

To solve the problem of torsional vibration the following steps were taken [10].

i. To analytically determine the torsional vibration response, one needs to calculate the torsional natural frequencies of the propulsion system. This requires knowledge of the stiffness and polar mass inertia of the shaft and components under investigation.
ii. Dampers such as the viscous type (Houdaille type) are usually utilized in reciprocating engines to help reduce torsional vibration and stresses. These dampers are usually designed to protect the engine crankshaft and not necessarily the driven machinery. To be efficient, dampers are to be located at a point with high angular velocity, usually near the anti-node of the crankshaft where the amplitude is high.
iii. To contribute to the solution of torsional vibration problems, [11] classification society chosen by the owners will invariably make its own assessment of the conditions presented by the vessel's owners and will judge by criteria based on experience. Contemporary designers can carefully adjust the resonance frequency, forcing impulses and resultant stresses. This is done by reconfiguring shaft sizes, number of propeller blades, crankshaft balance weights and firing orders of the engine cylinders.
iv. The use of elastomer-based mounting systems to eliminate noise and vibration. Rubber-to-metal bounded systems are the most commonly applied anti-vibration mountings.

III. Methodology

The methodology involves mathematical design of a three-bladed bulk carrier propeller shaft and calculation of polar mass moment of inertia of the shaft. The natural frequency of the designed shaft is calculated from the polar moment of inertia obtained. Solidworks Computer Aided Design software will then be used to model the shaft using values obtained from the mathematical design and physical properties of the material. The modelled shaft will then undergo vibration study using ANSYS computer aided design software from which the response of the shaft to different frequencies will be examined.

A. Mathematical design of the shaft

The specifications of the bulk carrier are shown in Table III.

TABLE III: Specifications of Bulk Carrier

Specifications		Units
Bulk Carrier Size (At scantling draught)	35000	Deadweight Tonnage (DWT)
Average design speed	30.0	Knots
SMCR Power	85	Hp
Indicative RPM	3000	RPM

$$\text{Power } (P_B) = 85\text{hp}$$
$$\text{Speed } (V_S) = 30\text{Knots}$$

Revolution per minute (n) =3000 rpm

The speed of advance (V_A) [12] of the propeller can be calculated using (9).

$$V_A = V_S(1 - w) \quad (9)$$

where:

$$w = 0.15 = \text{Taylor's wake friction}$$
$$V_A = 30(1 - 0.15)$$
$$V_A = 25.5\text{Knots}$$

Brake Power (P_B) is the power delivered at the engine coupling or flywheel while shaft power (P_s) is the output power available at gearbox coupling. The correlation between the Brake Power and Shaft Power [13] is shown in (10) while the formula to calculate the developed power is shown in (11).

$$P_B = \frac{P_S}{\eta_S} \quad (10)$$

where η_S = is the shaft efficiency and have values of 0.98 for ships with engine located at aft and 0.97 for engine located amid ship [14] For the purpose of this publication, 0.98 will be used.

$$P_S = P_B\,\eta_S$$
$$P_S = 85 \times 0.98 = 83 \text{ hp}$$

The power delivered to the shaft can be calculated using (11).

$$P_D = P_S\eta_S \quad (11)$$
$$P_D = 83 \times 0.98 = 81\text{hp}$$

Using the chart of type B.3-50 $B_P - \delta$ of Wageningen B series [15] as shown in Fig. 7, the Brake Power coefficient (B_P) can mathematically be presented as shown in (12):

$$\text{The power coefficient } (B_P) = \frac{nP_D{}^{0.5}}{V_A{}^{2.5}} \quad (12)$$
$$B_P = \frac{3000 \times 81^{0.5}}{25.5^{2.5}} = 8$$

Using the chart of type B.3-50 $B_P - \delta$, the values of $\left(\frac{P}{D}\right)$ and δ_{opt} are obtained as 1.17, and 113 respectively. Hence, calculating the Optimum Diameter (D) of the propeller with (13):

$$D = \frac{\delta_{opt}V_A}{n} \quad (13)$$
$$D = \frac{113 \times 25.5}{3000}$$
$$D = 0.96\text{m}$$

Having determined the propeller diameter (D), the Hub Diameter (d) can now be calculated from the Wageningen B series chart for 3 blades with (14):

$$\frac{d}{D} = 0.18 \quad (14)$$
$$d = 0.18 \times 0.96$$
$$d = 0.17m$$

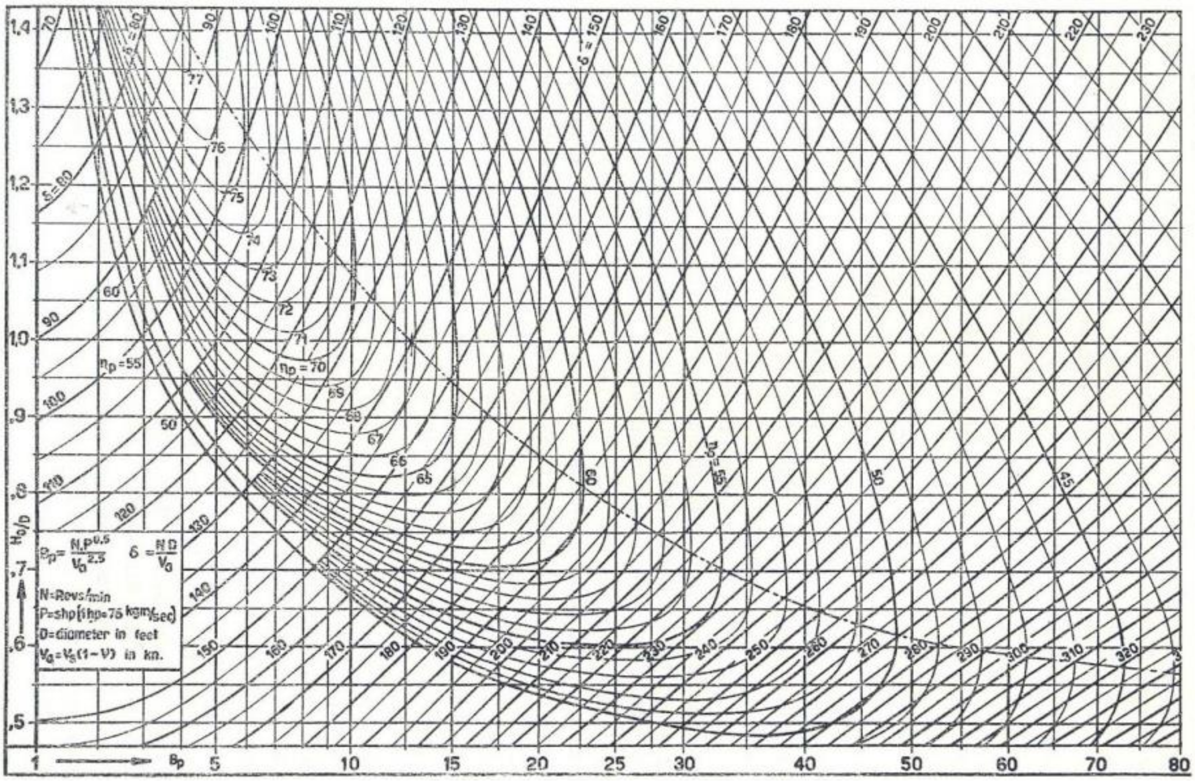

Fig. 7. B.3-50 $B_P - \delta$ Wageningen Blade series

B. Polar moment of inertia of designed propeller

In order to compute the polar moment of inertia of the propeller (I_P) and added moment of inertia of the entrapped water (I_E), (15) and (16) can be used [16].

$$I_P = 0.275Wr^2 \quad (15)$$

$$I_E = KI_P \quad (16)$$

where

W = Weight of propeller
r = Radius of the propeller
K is a correction factor and is equal to 0.25

The weight (W) of the blades can be calculated using (17):

$$W = 1.98 B_{tf} \varsigma Y r^3 \quad (17)$$

where:

$B_{tf} = 0.05$
$\varsigma = 0.50$
$\text{Radius}(r) = \frac{0.96}{2} = 0.48\text{m}$
Y = Specific weight of material
Material [17] used is Naval Brass (Uninhibited - C46400)

Y = specific weight of water × specific gravity of material
specific weight of water = 9.807 kN/m^3
specific gravity of material = 8.410
Modulus of Rigidity (G) = 3.861 × $10^{10} N/m^2$

$$Y = 9.807 \times 8.410 = 82.48 \text{KNm}^3$$
$$\therefore W = 1.98 \times 0.05 \times 0.50 \times 0.48^3 \times 82480$$
$$W = 451.52\text{N}$$

Hence:

$$I_P = 0.275 \times 451.52 \times 0.48^2$$
$$I_P = J_2 = 28.61 \text{Nm}^2$$
$$I_E = K I_P$$
$$I_E = 0.25 \times 28.61$$
$$I_E = 7.15 \text{Nm}^2$$

The polar moment of entrapped water (I_E) acts as a damper during operation to reduce the effect of vibration. Hence, the effective polar mass moment at the propeller during operation is the contribution of both Torsional moment of inertia of the propeller (J_2) and that of the entrapped water (I_E).

C. Input Specifications of Propeller Shaft

$$\text{Torque(T) on driven shaft} = \frac{\text{Delivered Power } (P_D)}{\text{Speed}}$$

Since 1Hp = 745.7W

$$P_D = \frac{81\text{hp} \times 745.7\text{W}}{\text{hp}} = 60402\text{W}$$

$$\text{But gear ratio} = \frac{n_1}{n_2} = 2$$

Hence,

$$\frac{3000}{2} = 1500\text{rpm} = n_2$$
$$n_2 = \frac{1500\text{rev} \times 2\pi \text{ rad} \times 1 \text{ min}}{\text{min(rev)} \times 60\text{s}}$$
$$n_2 = 157 \text{ rad/s}$$
$$\text{Torque (T)} = \frac{60402}{157}$$
$$T_2 = 384.72\text{Nm}$$
$$\text{Torque from driver shaft } (T_1) = \frac{85 \times 745.7}{314} = 202\text{Nm}$$

Since 1Hp = 745.7W

For a value of 0.17m outside diameter, a corresponding value of 0.10m internal diameter is used.

1-inch diameter should have maximum unsupported length of 30 inch [18]. Hence, the length of the shaft is obtained by the multiplication of its diameter by 30.

$$\text{Diameter of shaft} = 0.10\text{m}$$
$$\therefore \text{Length of shaft} = 0.10 \times 30 = 3\text{m}$$

To calculate the polar moment (J) of the driven shaft, (18) is used

$$J = \pi \left(\frac{{d_0}^4 - {d_i}^4}{32}\right) \quad (18)$$

$$\text{Polar Moment of Shaft (J)} = \frac{\pi(0.10^4 - 0.09^4)}{32} \text{m}^4$$

Hence $J = 3.38 \times 10^{-6} \text{m}^4$

In Fig. 8, K_1 and K_2 are the torsional stiffnesses of shaft 1 and 2 while J_1 and J_2 are the polar mass moments of inertia of the flywheel and propeller respectively.

When the input shaft rotates with an angle (ψ), the output shaft also rotates accordingly. Then the potential energy (U) of the output shaft is as shown in (19).

$$U = \frac{1}{2} K_2 {\psi^2}_2 = \frac{1}{2} K_2 \left(\frac{\psi_1}{R}\right)^2 \quad (19)$$

Also, the potential energy of the input shaft is as shown in (20)

$$U = \frac{1}{2} K_1 {\psi_1}^2 \quad (20)$$

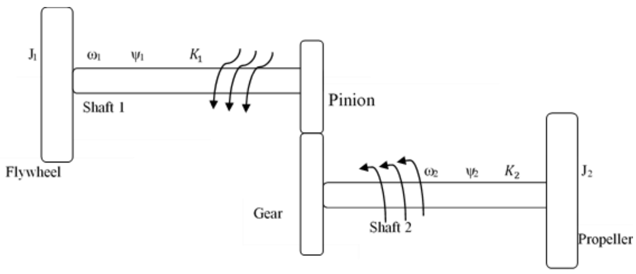

Fig. 8. Schematic of propeller shafting system

Equating (19) and (20) yields (21):

$$\frac{1}{2} K_1 {\psi_1}^2 = \frac{1}{2} K_2 \left(\frac{\psi_1}{R}\right)^2$$
$$\therefore K_1 = \frac{K_2}{R^2} \quad (21)$$

Calculating the Torsional stiffness (k_2) of the propeller shaft, (22) is used:

$$k_2 = \frac{GJ}{l} \quad (22)$$
$$\therefore k_2 = \frac{3.861 \times 10^{10} \times 3.38 \times 10^{-6}}{3}$$
$$k_2 = 4.35 \times 10^4 \text{Nm/rad}$$

Hence:

$$k_1 = \frac{4.35 \times 10^4}{4}$$
$$k_1 = 1.09 \times 10^4 \text{Nm/rad}$$

Since the masses of the gears and shafts are negligible, their respective polar mass moments of inertia are also

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neglected.

Calculating the polar mass moment of the shaft system, the kinetic energy (KE) of both the driven and driver shafts are compared as shown in (23):

$$\mathrm{KE} = \frac{1}{2}J_1\dot{\psi}_1^{\,2} = \frac{1}{2}J_2\dot{\psi}_2^{\,2} \qquad (23)$$

But

$$\frac{\psi_1}{\psi_2} = R$$

$$\frac{1}{2}J_1\dot{\psi}_1^{\,2} = \frac{1}{2}J_2\left(\frac{\dot{\psi}_1}{\dot{R}}\right)^2$$

$$\frac{J_2}{J_1} = R^2$$

$$J_1 = \frac{J_2}{R^2} \qquad (24)$$

$$J_1 = \frac{28{\cdot}61}{2^2}$$

$$J_1 = 7.15\mathrm{Nm}^2$$

Fig. 9 shows the series connection of the shafts.

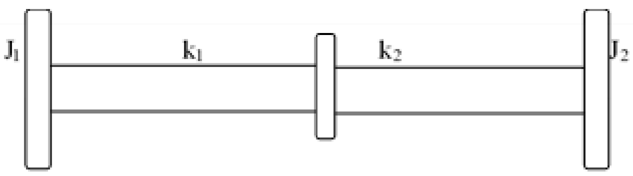

Fig. 9. Series connection of propulsion shaft

The effective stiffness (k_e) of the series connection is calculated as shown in (25):

$$\frac{1}{K_{eq}} = \frac{1}{K_1} + \frac{1}{K_2} \qquad (25)$$

$$K_{eq} = \frac{K_1K_2}{K_1+K_2}$$

$$K_{eq} = \frac{(1.094\times10^4)(4.38\times10^4)}{(1.094\times10^4)+(4.38\times10^4)}$$

$$K_{eq} = 8752\mathrm{Nm/rad}$$

The natural frequency (w_{nf}) of the system can thus be calculated as shown in (26) [19].

$$w_{nf} = \sqrt{\frac{K_{eq}(J_1+J_2)}{J_1J_2}} \qquad (26)$$

$$w_{nf} = \sqrt{\frac{8754(28.61+7.15)}{(28.61)(7.15)}}$$

$$w_{nf} = 39.12\mathrm{rad/sec}$$

$$f = 39.12\times2\pi$$

$$f = 246\mathrm{Hz}$$

D. Harmonic Analysis

Harmonic Analysis is used to determine the response of the propeller shaft under steady-state sinusoidal loading at a given frequency. Equation (27) is used to analyse harmonic analysis.

$$(-\Omega^2[M] + j\Omega[C] + [K])\{x_1 + jx_2\} = \{F_1 + jF_2\} \qquad (27)$$

Where:

M is the mass of the shaft
C is the Damping constant
K is the stiffness of the shaft
F is the Load applied at a particular frequency

Loads of 202Nm and 384.72Nm are applied on the driver and driven shafts respectively. Damping constant of 7.15 which is derived from the polar moment of the entrapped water and stiffness of 8752Nm/rad which is derived from the equivalent stiffness coefficient calculation are also employed in carrying out the harmonic analysis on ANSYS environment.

IV. Result and Analysis

The idea of harmonic analysis is to calculate the response of the structure at several frequencies. This is done to obtain a graph of some response quantity. Peak responses are then identified on the graph and investigated at the corresponding frequencies.

A. Displacement-Frequency Analysis

Fig. 10 shows the harmonic response of the shaft when exposed to frequency range of $0 \le f_n \le 500$ and when sinusoidal loads of 384.72Nm and 202Nm act on the shaft. The displacement is rapidly increased to $2.73\times10^{-7}m$ at about 60Hz then drastically reduced $to\ 8.935\times10^{-8}m$ before increasing steadily to the peak of $7.87\times10^{-6}m$ at about 280Hz. The peak displacement connotes possible failure of the shaft if driven at this frequency.

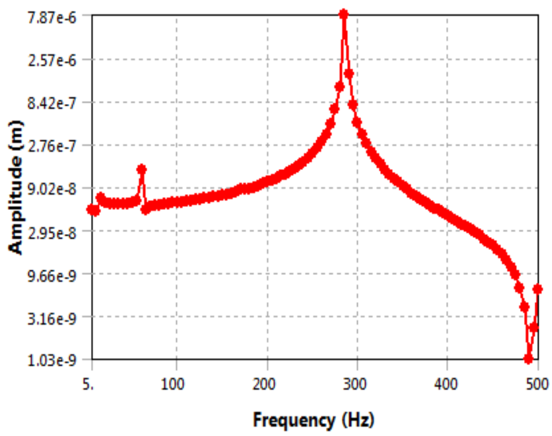

Fig. 10. Displacement-Frequency response of shaft

Although, the manual calculation of the natural frequency of 246Hz does not equal the value obtained from the numerical value of 280Hz, it is advisable to avoid the speed range of 246Hz to 280Hz for smooth operation and longevity of the shaft as well as comfort of passengers.

$$\%Error = \frac{280-246}{280}\times100 = 12\%$$

B. Phase Angle-Frequency Plot

Phase angle in this harmonic analysis refers to the phase of the response relative to the frequency. The amplitudes of $2.73\times10^{-7}m$ and $7.87\times10^{-6}m$ which correspond with the frequencies of 60Hz and 280Hz respectively, are in phase as shown in Fig. 11. While at 280Hz, the response of displacement occurs at 180^0, at 60Hz, the displacement occurs at -180^0.

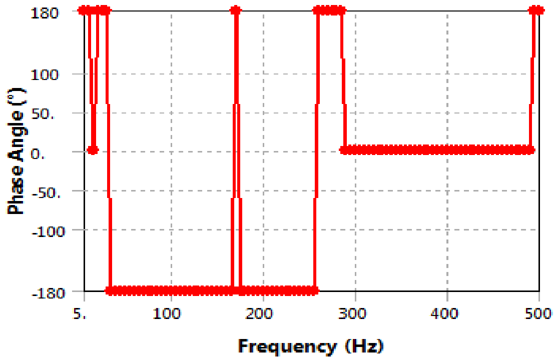

Fig. 11. Phase Angle-Frequency response of Shaft

Phase is used to determine misalignment which occurs when the shaft length of two directly mating components meet at an angle and are offset from one another [20]. Misalignment produces high amplitudes with a phase difference of 180^0.

Hence, the peak amplitude obtained might be an evidence of misalignment which can contribute to failure of the shaft due to excessive vibration at the 280Hz.

Vibrations that occur at phase shift of approximately zero degree are caused by reaction forces [21]. Hence reaction forces which originate from connections like coupling flanges, thrust blocks, engine connection flanges, etc. contribute to vibration along the length of the shaft. Fig. 12 shows the misalignment of the propeller shaft due to deformation of stern tube and intermediate bearings. Shaft misalignment is caused by variation in coupling stiffness on rotation and the forcing frequency generated are harmonics of the shaft's speed [22].

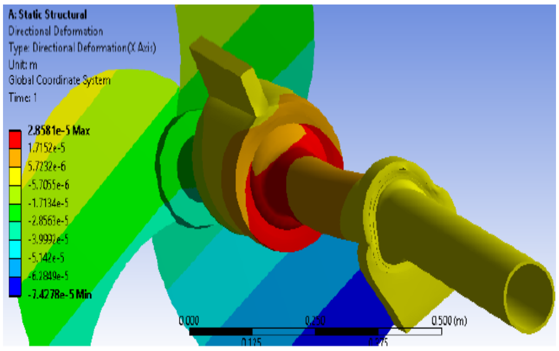

Fig. 12. Stern tube and intermediate bearing deformation

V. Conclusion and Recommendations

A. Conclusion

A 3-bladed marine propeller shaft have been mathematically designed. The natural frequency of the shaft was mathematically obtained. Solidworks software was used to model the propeller shaft according to the specifications obtained from the mathematical analysis. ANSYS software was used to numerically analyse the shaft to find out the critical frequency. Graph obtained from the ANSYS shows that the amplitude of vibration is the highest at the critical frequency. The maximum amplitude obtained at 280Hz is to be avoided if the shaft is to operate smoothly and last longer.

Phase angle is another important aspect in the vibration of shaft. The analysis reveals that out-of-phase shafts cause vibration. The sharp or peak amplitude obtained from the graph at the resonance frequency is shown to have occurred when the response is out of phase. This is caused by misalignment of shaft as obtained in the phase-frequency plot.

B. Recommendations

In light of this thesis, the recommendations are as follows:

1. Bulk carrier should never be operated at the natural frequency of the engine.
2. Angles at each end of shaft should be equal with each other to cancel out torsional vibration.
3. The critical speed arrangements should always be carried out from preliminary design stage.

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