Study of Effective Parameters in Stability and Vibration of Marine Propulsion Shafting Systems

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Abstract. The marine propulsion system is a system that a ship depends on in travelling and maneuvering in the existence of different types of waves, which reduce the efficiency of the system. In this work, different types of vibration such as torsional, lateral and longitudinal and their effect on ship and propulsion system are studied. Vibration seriously threatens the reliability and safety of the ship. The finite element method is established to study vibration at different rotational speeds. In this work vibration, analysis and modeling with simulation for cases of lateral and torsional vibration are performed by using MatLab software. In addition, ANSYS 2019 R3 is used to study some cases of marine propulsion system. In this work, effective parameters that play important role in vibration reduction such as the diameter, stiffness, couple unbalance and mass unbalance are discussed. The results of that analysis provided can help to predict the problems. In general, to achieve better stability and safety, all the conditions of the propulsion system must be taken into account to diagnose the problem. Hence, the data which are provided in diagrams and tables can be used as a guide in the design stage or maintenance stage to treat system problem and then take the proper action.

1. Introduction
To move from one place to another the ships usually need a drive system to do so. The marine propulsion system is responsible for travelling and maneuvering. The drive system consists of three main parts: an energy source or prime mover, a transmission and a propulsor. For most propulsion system reduction of the vibration is important to prevent the noise and strain that could impact reliability and safety [1]. During the operation of a system related to marine propulsion and navigation of the ship, the propulsion suffers from all kinds of exciting forces within hull forces. The vibrations are excited from the propulsion system includes longitudinal, torsional vibration, and transverse vibration with coupled vibration forms. Hence, the internal and external active coupled leads to dangerous problems that should be avoided [2]. There are many types of prime movers such as diesel engine, gas turbine, steam turbine, electric motor and others. The most common prime mover is the diesel engine [3]. The efficiency of a propeller depends on multiple factors such as the diameter of the propeller, rake, and pitch. Otherwise, the distance between the tip of the propeller and the bottom of the vessel hull is important to avoid vibration and high noise [4]. The efficiency of marine propulsion in the ship is discussed while the location of the propeller is changed under the hull vessel. Some researchers employed ANSYS-CFX with CFD code and RANSE solver [5] to investigate the efficiency of the propulsion system. ANSYS analysis software's was used to calculate the natural frequencies for the shaft frequency where a percent error of 12 % was obtained for the natural
frequency calculation. ANSYS-Analysis system shows also the phase difference at 180°, which occurred at 280 Hz that give a misalignment for shaft as a signal. The main reason for misalignment at the shaft is to wear due to fraction [6]. The authors present a guideline, which is useful to reduce the vibration obtained from the marine propulsion of a ship. By the procedure of finite element, model to ensure that, the natural frequencies of the system are far away from operating points at different loading condition to prevent the resonance [7]. The resonance frequencies and amplitudes are usually described by using of torsional vibration calculations. Also, the system vibration at its critical frequency explains the self-exited vibration in the system [8]. A propulsion system can be represented as the diagram figure 1.

Figure 1. Marine propulsion system [1].

2. Equation of Torsional Vibration

Torsional vibration in the reciprocating propellers “power, the combustion engine and transmission system are the result for pulsing torque”. This torque is dangerous in the marine transmission system for the shaft line because it may lead to either damage to the propulsion line or cause fatigue in material of the shafting system. By the FEM computer program, the analysis of torsional vibration for marine transmission system was done to describe its dynamic behaviour. While general equation, which is used to calculate torsional vibration [9], is given as, follow:

\[
I \ddot{\phi} + C \dot{\phi} + K \phi = M_E(t)
\]  \hspace{1cm} (1)

\(I\) : moment of inertia masses matrix.
\(C\) : torsional damping matrix.
\(K\) : torsional stiffness matrix.
\(M_E\) : The excitation moment.
\(\phi\) : Rotational angle.

Figure 2. Torsional vibration model of propulsion system [9].

Damping (C) is insignificant on mode and natural frequency. Then equation above becomes;

\[
I \ddot{\phi} + K \phi = 0
\]  \hspace{1cm} (2)
Where, $J$ : the equivalent mass moment of inertia, $C$ : equivalent of torsional damping.

3. Equation of axial vibration

The axial vibration is usually induced by:
1. Inertia and radial forces, which lead to deformations in the crankshaft.
2. Longitudinal hydrodynamic forces in the propeller.

This type of vibration in marine propulsion system can be analyzed in a mathematical model as below:

$$M \frac{d^2 q(t)}{dt^2} + C \frac{dq(t)}{dt} + K \cdot q(t) = h(t) \quad (3)$$

Where $M$, $C$, $K$ are mass, damping, stiff matrices and $h(t)$, $q(t)$ are excitation and displacement vectors [10]. Figure 3 explains the longitudinal vibration in marine propulsion shaft while as a maximum level of vibration leads to influence the safety of the marine vessel. Propeller with blades is subjected to a high vibration at its first natural frequency [11].

4. Modeling

Model creation of a system has been modeled by analysis with computer software. The model can be classified into two types: simplified model and computed model.

4.1. Simplified model:

The polar mass moments of inertia as $I_{p1}$ and $I_{p2}$ for rotors of machines as shown in figure 4, the connections of machines by shafts and coupling where the $(K)$ is referred to the stiffness of combined torsional and its unit is (N.m/rad). Otherwise, it is able to make more simplification in case one of the two machines has inertia greater than the other machine inertia.

4.2. Computed model

This model describes the system that has two degrees of freedom (two inertia or more). Figure 5 shows the model of lumped parameter and inertia [12].
5. Simulation approaches
It is usually during the design stage the simulation approach is employed. That will give the designer a good insight into the performance expected of the vessel design. The performance of any dynamic system can be described by two various approaches. The first one is to study the performance of the system from the prediction of some identical systems and observe how the output is changed by changing the input data. This method simulates the real systems. The second approach is describing the performance of the real system. Hence, in this method, the numbers of elements and their names should be known [13].

5.1 Case1 (isotropic-bearing)–variable shaft diameter
By using the finite element method, MatLab software the input characteristic that used as following: length of shaft 1.5 m, number of nodes 7, number of segments 6, number of bearings 2, shaft out diameter 0.250 m, disk out diameter 1m, bearing stiffness 107 MN/m, rotor speed 1500 rpm, disk thick 15 mm, $E = 211$ GN/m$^2$, $G = 8.12$ GN/m$^2$, $\rho = 7810$ kg/m$^3$, input distance length of shaft between (Node 1 and Node 2) is 1 m and remind distance 0.5 m, after applying the parameters above with designated program then model of system and simulation were obtained as follows:

![Figure 6. Model of marine propulsion system.](image)

The results obtained are collected for the first case, which includes a variable shaft diameter as shown in Table 1:

| Shaft diameter (m) | Natural frequency (Hz) Forward whirl mode | Natural frequency (Hz) Backward whirl mode | Speed (rpm) |
|-------------------|------------------------------------------|-------------------------------------------|-------------|
| 0.250             | 11.66                                    | 11.27                                     | 338.1 - 349.8 |
|                   | 38.6                                     | 34.78                                     | 349.8 - 338.1 |
| 0.350             | 10.84                                    | 10.51                                     | 315.2 - 325.2 |
|                   | 27.99                                     | 26.09                                     | 839.7 - 782.6 |
| 0.450             | 9.931                                    | 9.617                                     | 297.9 - 288.5 |
|                   | 22.06                                     | 20.68                                     | 661.7 - 620.3 |
| 0.550             | 9.052                                    | 8.724                                     | 271.6 - 261.7 |
|                   | 17.02                                     | 18.28                                     | 548.4 - 510.7 |
| 0.650             | 8.221                                    | 7.872                                     | 246.6 - 236.2 |
|                   | 15.65                                     | 14.4                                      | 469.4 - 432  |

It is obvious from the results of Table 1 and the natural frequency which is obtained as a pair that is a critical speed is propositional with the increasing of shaft diameter. Where the critical speeds will have a narrower range as the diameter of the shaft increases. This leads to reducing the risks which are induced by variable rotational speeds. Also, it is clear that the natural frequencies decrease while the diameter increases. Therefore the optimum case is the final which the diameter is 0.650 m.
5.2 Case 2 - Couple unbalance – isentropic (variable stiffness)

There are different stiffness were investigated and their response magnitudes can be written down as in Table 2 in order to make a comparison:

| Item No. | Stiffness (MN/m) | Peak value of response magnitude (m) | Critical speed (rpm) | Node No. |
|----------|------------------|------------------------------------|----------------------|----------|
| 1        | 60               | $1.73 \times 10^{-4}$, $1.94 \times 10^{-4}$ | 3100, 3100           | 4, 7     |
| 2        | 110              | $7.076 \times 10^{-4}$, $4.54 \times 10^{-4}$ | 3200, 3200           | 4, 7     |
| 3        | 210              | $8.70 \times 10^{-4}$, $2.9 \times 10^{-4}$ | 3260, 3260           | 4, 7     |
| 4        | 310              | $3.68 \times 10^{-4}$, $8 \times 10^{-4}$ | 3280, 3280           | 4, 7     |
| 5        | 410              | $4.33 \times 10^{-4}$, $7 \times 10^{-4}$ | 3300, 3300           | 4, 7     |

The vibration response magnitude is related to the stiffness parameter. The variations of bearing stiffness play an effective role in the excitation of the response where it decreases while stiffness is increased as shown in Table 2. While both geometry and material are a function of a stiffness.

5.3 Case 3 axil vibration – free vibration

In this case, a propulsion system is carried out by (ANSYS WORKBENCH) where the shaft length is 3.5 m and the masses of the gearbox, the propeller is 250 kg, 500 kg respectively and stiffness bearing is 4000 N/mm. Where shaft propeller acts as a beam and the engine specifications: the revolution per minute is 600 rpm and 220 hp. Propeller and gearbox are represented by a point of masses at shaft end as in Figure 7. While the rotating shaft is supported on elastic support at one end with specified foundation stiffness elastic support as 10.2 N/mm³ where another end is free, after applying those characteristics above, in addition to the engineering data which are density 7850 kg/m³, Poisson ratio 0.3, Shear modulus 77 GPa and elastic modulus 210 GPa. Then it will be obtained the figures 8, 9, 10, 11, which are acting as axial displacement modes.

![Figure 7. Geomatery of shaft with propeller gear as point masses.](image1)

![Figure 8. The first order axil displacement diagram of a model.](image2)
5.4 Case 4 a hollow shaft propulsion system

In this case, a hollow shaft is studied to simulate the case where control arms go through the shaft for a variable pitch propeller. There are main characteristics that are used as follows: hollow shaft length of 3.5 m, the outer diameter and inner diameters are selected from a number of values. Stiffness foundation is 15 N/mm³, the shaft is divided into seven segments and eight nodes and by using the (ANSYS-WORK BENCH) make a disk which acts as a propeller at the end of the shaft with a diameter of 90 cm, and a thickness of 8 mm, and with the specifications of a copper metal as:

- Density = 8.94 × 10⁻⁶ kg/mm³,
- Elastic Modulus = 1.25 × 10⁵ MPa,
- Poissons ratio = 0.345,
- Bulk Modulus = 1.3441 × 10⁵ MPa,
- Shear Modulus = 46468 MPa,
- Ultimate Tensile Strength = 152 MPa,
- Yield Tensile Strength = 33.5 MPa.

Otherwise the properties of a steel shaft propeller as:

- Density = 7.85 × 10⁻⁶ kg/mm³,
- Elastic Modulus = 1.6667 × 10⁵ MPa,
- Shear Modulus = 76923 MPa,
- Ultimate Compressive Strength = 0 MPa,
- Yield Compressive Strength = 250 MPa.

After entering the mentioned characteristics for the model shown in figure 12, a force is assumed to act in the X and Z axis with the magnitudes of 22000 N and 150 N respectively as shown in figures 13, 14, which show the force in X and Z directions respectively. Otherwise, figure 15 explains the two forces which are subjected at the shaft end.
Figure 14. Geometry of a hollow shaft with (z) direction at the end shaft.

Figure 15. Geometry of a hollow shaft with propeller and two forces are subjected at the end.

It is possible to prepare a table showing the relation between the variation of the response amplitudes with changing the diameters of the propulsion of a hollow shaft while that is helping to get an optimum design of shaft propeller. It is clear from the results which have been recorded in Table 3 that the response amplitude increases with decreasing the shaft diameters or by the decreased cross-sectional area of the shaft. Also, in the table below, the range of natural frequencies in which the greatest response occurs in each case has been recorded.

| Case No. | Variable diameter (Cm) | Response Amplitude (mm) | Natural Frequency (Hz) |
|----------|------------------------|--------------------------|------------------------|
|          | $D_0$                   | $D_i$                    |                        |
| 1        | 7                      | 15                       | 0.125                  |
|          |                         |                          | 4.424 - 42.96 - 68.87  |
| 2        | 8                      | 15                       | 0.1373                 |
|          |                         |                          | 4.352 - 7.691 - 43.16  |
| 3        | 9                      | 15                       | 0.1528                 |
|          |                         |                          | 4.311 - 44.33 - 64.55  |
| 4        | 10                     | 20                       | $7.351 \times 10^{-2}$ |
|          |                         |                          | 6.814 - 57.57 - 80.028 |
| 5        | 12                     | 20                       | $8.160 \times 10^{-2}$ |
|          |                         |                          | 6.716 - 9.286 - 58.45  |

Table 3. Response amplitude and natural frequency with variable diameter.

5.5 Case 5 comparison between Matlab and ANSYS

5.5.1 At a variable diameter

In this case, the results of the natural frequency that are obtained by the MatLab and ANSYS programs were compared, for shaft diameter 0.250 m with discrepancies ranging between 8.34 percent and 0.78 percent. As a consequence, since the same features are used in both programs, to get an accurate result it is important at the future work that recommend conducting experimental work after considering the numerical method and analytical solutions. The figure 16 is describe the geometry of propulsion system in ANSYS program which is acted by MatLab at previous case.

Table 4. Distinguish frequencies reading by MatLab- ANSYS at variable diameter.

| Order | MatLab results (Hz) | ANSYS results (Hz) | Discrepancy (%) |
|-------|---------------------|--------------------|-----------------|
| 1     | 11.27               | 10.33              | 8.34            |
| 2     | 38.6                | 38.3               | 0.78            |
5.5.2 At a variable stiffness

The geometry of the unbalance system at variable bearing stiffness at 210 MN/m and the difference of response reading is written down in Table 5 that shows the discrepancies between the two programs explained in this case, as well as the peak response results based on the same characteristics used in the Matlab software. Where by ANSYS program the maximum response is 0.084 m (on the shaft's meddle) at node 4, and the minimum response is 0.0027 m at node 7 (on the bearing), this program describes the unbalance system with 5 disks and two bearings at the end is depicted in figure 17.

| The stiffness (MN/m) | Node No. | Peak response by MatLab (m) | Peak response by ANSYS (m) | Discrepancy (%) |
|---------------------|----------|----------------------------|---------------------------|-----------------|
| 210                 | 4        | 0.087                      | 0.084                     | 3.44            |
|                     | 7        | 0.0029                     | 0.0027                    | 6.89            |

Finally, the differences between these programs are normally determined by the number of nodes and the solution mechanism, which is based on the analytic approach.

6. Conclusions
In this thesis, vibration analysis and modelling for a number of cases in marine applications are performed by using MatLab and ANSYS software. The results obtained to provide insight into how to avoid many types of vibration that may arise in marine systems through proper design. The main conclusions can be summarized as follows:

1. The diameter of propulsion shafts plays an important role and has an effect on the extent of the response and resonance speed range, as it was found that the critical speed is inversely proportional to the change of the diameter of the shaft. Also, it is clearly indicated that increasing diameter can reduce the range of critical speeds and hence provide better stability.

2. From the response to a couple of unbalance case under variable bearing stiffness, vibration magnitude is generally inversely proportional to the stiffness except for the stiffness of 60 MN/m. Also, the speed at which maximum response occurs is a function of stiffness too. Where the speed increases when stiffness is increased. However, the relation is not linear.

3. On the other hand, a study of the work for a hollow shaft propeller at different diameters was carried out by (ANSYS WORKBENCH) software to demonstrate the effect of these values on the response amplitudes under harmonic force. The maximum response is achieved at these ranges of frequencies which are important to diagnose in order to avoid them occurs during the operation of the propulsion system.

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