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CONTROL OF A SHIP SHAFT TORSIONAL VIBRATION VIA MODIFIED PID CONTROLLER

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Summary

External disturbances and unstable working performance of main engine cause vibration to propulsion plant of a ship. The most commonly encountered vibration is torsional vibration as a result of unsteady power loads and inertia forces on the ship main engine-shaft system. Crankshaft balancing weight as a passive system and torsional damping element as an active system are used in order to prevent these types of vibration. The objective of the present paper is to use the modified Proportional-Integrative-Derivative (PID) control system to enhance the performance of the ship shaft control system and to adjust the torsional vibration for a given reference input. The conventional PID control system did not give desired results. Therefore the performance of the traditional PID controller was improved by moving derivative and proportional blocks on feedback path. All simulation results show that the proposed modified control algorithm gives a better performance than conventional PID control.

Key words: *Modified Proportional-Integrative-Derivative (PID) control, ship main engine propulsion systems, vibration analysis.*

1. Introduction

Vibratory forces generated in ship propulsion systems by some internal sources such as main engine, shaft, propeller and gearbox as well as by some external sources such as wave, current and imbalanced ship loads are often unavoidable. These forces have influence on axial, radial and torsional vibrations. However, their effect on propulsion system can be minimized or reduced by different isolation methods one of which is the use of springs and damping elements for isolation between main engine and its foundations.

Hara et al. [1] established a main engine propeller shaft system by building block approach, and analysed the torsional, axial and lateral vibrations. The three dimensional solid model of the crankshaft was analysed by using the Finite Element Method (FEM), besides the equivalent beam model was constructed. Propeller was modelled as a rotor. The properties like weight, inertia and polar moment of inertia were entered to the equations and the additional mass of water was also taken into account. Bearings and rigidity of the body were considered as a spring in the model established. Vibration dampers placed in different positions were analysed. Then, results of the model were compared with measured results.

Because main engine shaft system has many elements having damping effect, it can be concluded that predictions of these approaches for the model need to be accurate.

Rao [2] modelled the power transmission system with spring and damping element as four-degrees of freedom system. Damping elements of the system were positioned between main engine and gearbox to reduce torsional vibration, and between thrust bearing and propeller to reduce lateral vibration. The springs were placed in between main engine, gearbox, bearings, propeller and foundations. Radial forces and propeller thrust forces were thought as moving forces on main engine and propeller, respectively. The equations of model were solved theoretically and then calculated values of torsional vibration were compared to practically measured values. Some of the calculated and measured values of frequencies are very close, while there are differences in some cases, which are recognized as suitable for ISO vibration standards.

Shu et al. [3] expressed torsional and axial vibrations of the engine crankshaft with Rayleigh differential method. The engine crankshaft was modelled as a mass spring system to obtain natural frequency of torsional and axial vibrations. Calculated results were compared with measured results and it is recognized that torsional vibration has an enormous effect on axial vibration that causes noise and vibration in engine.

MacPherson et al. [4] investigated methods to calculate the additional water mass of a propeller for vibration analysis. In this paper, the effect of added mass on the torsional and axial vibrations of the propeller was examined. In order to calculate the added water mass of Weldsma BS-VII series of propellers, a new formula and a new prediction was applied. The authors indicated that this prediction method would be beneficial for vibration analyses of propulsion systems because the reliability of the method has increased by the contribution of damping effect.

Grzadziela [5] carried out a study by using Matlab-Simulink, in which a propeller shaft system had four-degrees of freedom. In this study, the rotation torque of the main engine, fixed blade propeller torque, axial force and the shaft line bearings hydrodynamics effects have been taken into consideration. Model was established by using the FEM, the propeller shaft speed ranging in different frequency values were determined according to different forms of support. Simulation results showed a maximum 10% error with the measured values converges.

Zhang et al. [6] investigated the influence of angular and axial forces acting on propeller by modelling propeller and crankshaft. Frequency changes in torsional vibrations have been observed between the propeller and 1st order crankshaft journal, besides between 5th and 6th order piston crank journals. For axial vibrations, on the other hand, changes in frequency values are expressed almost in the same way. Considering coupled vibrations, a nonlinear behaviour effect was observed. They determined that large amount of errors would be experienced, when the propeller crankshaft analysis was assumed to be linear.

Sestan et al. [7] described the problem of torsional vibration resonance in the propulsion system of a catamaran ferry. They introduced that in some cases shaft line torsional vibration resonance could be appear due to non-typical reasons.

In any reference change in manipulated signal has some malfunction in many occasions. So as to avoid set points and recover system response, Ogata K. [8] modified PID controller that derivative and proportional blocks moved on feedback path. The negative effects of conventional PID case of internal zeros were recovered by the modified PID controller.

Kou Yamada et al. [9] presented a design method for modified PID controller for stable plants. The method suggests that stable plants and admissible sets of PID gains act independent from each other.

Elksasy et al. [10] modified PID controller as the integral block by adding time delay and derivative block by subtracting time delay. The purpose of the modified PID in this paper is to solve instability problem.

In this study, main engine, shaft and propeller system were modelled by Lagrange Method and the model was solved by using Matlab-Simulink. Analysis of torsional vibration was emphasized since it causes hazardous vibration. The aim of this paper is to control shaft torsional vibration by modified PID controller. As a result, modified PID controller was compared with conventional PID to make better system response.

This paper is organised as follow: In Section 2, the ship model is presented. The PID controllers are considered in Section 3. The torsional vibration analysis and simulation is addressed in Section 4. Finally, conclusions are presented in Section 5.

2. Torsional Vibrations Modelling Of Propulsion Plant

Unequal gas pressure in the cylinder of the main engine causes variable speed of rotation is the main reason of shaft torsional vibrations. There are some methods, one of which is Lagrange equations, to solve this kind of dynamics system. Kinetic, potential and damping energy are considered and described in form of Lagrange Equations (William W. Seto [11]);

$$\frac{d}{dt}\frac{\partial T}{\partial q_i}-\frac{\partial T}{\partial q_i}+\frac{\partial U}{\partial q_i}+\frac{\partial D}{\partial q_i}=Q_i \tag{1}$$

T is total kinetic energy of a system. U is the change of potential energy of a system with respect to its potential energy in the static-equilibrium position. Next, D is total damping energy. In addition Q_i is generalized non potential moments. q_i is the generalized coordinates which satisfies i=1,2,3,…,n.

The general formulation of the matrix equation is shown as follow;

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{F_x\} \tag{2}$$

It is the basic formulation of equations to determine the natural frequencies and mode shapes for two degree of freedom system. M, C and K are inertia, dampings and stiffness materials respectively. $\{x\}$ is referred as generalized coordinates, and $\{F_x\}$ is given as a generalized forces. Damping and springs elements can be reduced by means of dynamics loads.

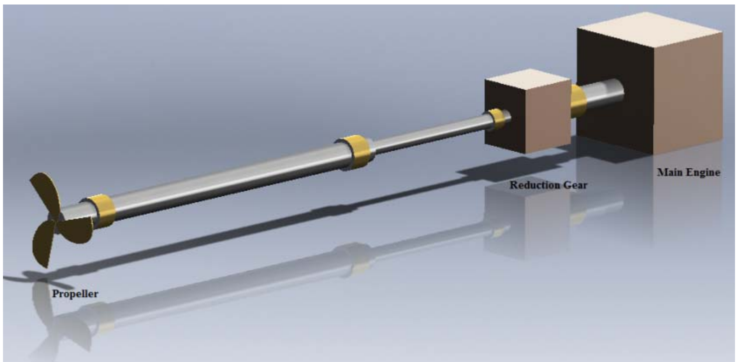


Fig. 1 Main engine and shaft simplified model

The simplified shaft propulsion system is shown in Fig. 1 consisting of the main engine, shaft and propeller. In order to investigate the condition, a physical model of system can be formed as in the Fig. 2. The physical model has been composed of analysis of vibration angular.

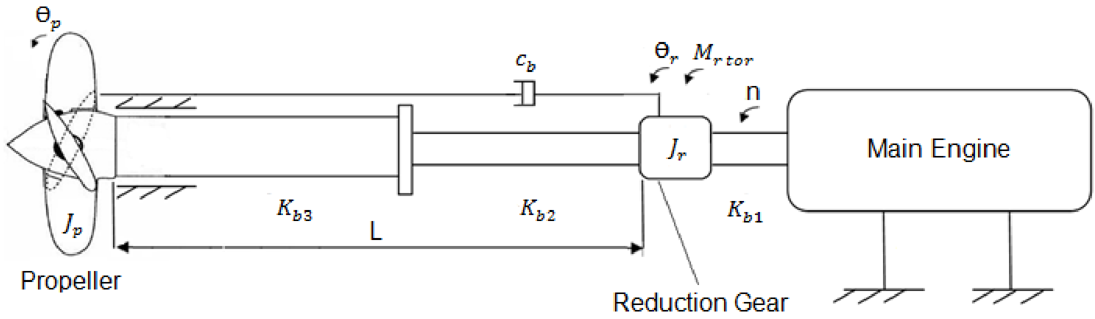


Fig. 2 Physical model of system

Considering physical model in Fig. 2, and when Lagrange method was applied,

$$T = \frac{1}{2} J_r \dot{\theta}_r^{\;2} + \frac{1}{2} J_p \dot{\theta}_p^{\;2} \quad (3)$$

$$U = \frac{1}{2} k_{b1} (\theta_r)^2 + \frac{1}{2} k_b (\theta_r - \theta_p)^2 \quad (4)$$

$$D = \frac{1}{2} c \left(\dot{\theta}_r - \dot{\theta}_p\right)^2 \quad (5)$$

are obtained. k_{b1} is the stiffness factor and c is damping coefficient. With a view to deriving the mathematical equations of this system, $\Theta_r > \Theta_p$ is assumed.

$$J_r . \ddot{\theta}_r + c\left(\dot{\theta}_r - \dot{\theta}_p\right) + k_{b1r} + k_b\left(\theta_r - \theta_p\right) = M_{rtor} \quad (6)$$

$$J_p . \ddot{\theta}_p - c\left(\dot{\theta}_r - \dot{\theta}_p\right) - k_b\left(\theta_r - \theta_p\right) = 0 \quad (7)$$

Θ_r and Θ_p are the angles of propeller and shaft motion. M_{rtor}, J_r and J_p are the constants that indicate reduction gear torque, reduction gear inertia and propeller gear inertia, respectively. Differential equations of motion for two degree of freedom system are indicated in matrix form;

$$\begin{bmatrix} J_r & 0 \\ 0 & J_p \end{bmatrix} \begin{bmatrix} \ddot{\theta}_r \\ \ddot{\theta}_p \end{bmatrix} + \begin{bmatrix} c & -c \\ -c & c \end{bmatrix} \begin{bmatrix} \dot{\theta}_r \\ \dot{\theta}_p \end{bmatrix} + \begin{bmatrix} k_b + k_{b1} & -k_b \\ -k_b & k_b \end{bmatrix} \begin{bmatrix} \theta_r \\ \theta_p \end{bmatrix} = \begin{bmatrix} M_{rtor} \\ 0 \end{bmatrix} \quad (8)$$

The complete solution can be written by setting $\theta = \theta \sin(\omega_n t)$ and $\omega_n^{\;2} = \lambda$ as;

$$J_r . J_p . \lambda^2 + \lambda\left(-J_r k_b - J_p k_b - J_p k_{b1}\right) + k_b . k_{b1} = 0 \quad (9)$$

Table 1 Some properties of transmission system

MAN 8 L 32/40	*Values*
Reduction gear inertia	*3118 Kgm^2*
Propeller inertia	*50725 Kgm^2*
Torsional stiffness coefficient 1	*237673 N m/rad*
Torsional stiffness coefficient 2	*85582 N m/rad*
Damping coefficient	*5000 Ns/m*
Reduction gear torque	*50955 Nm*

From the expression (9), eigenvalues and natural frequencies have been found by considering data from Table 1.

$\lambda_1 = 103.8258$ and $\lambda_2 = 1.2375$.

There are four poles and a zero of shaft torsional motion in uncontrolled system. It is obviously seen in Fig. 3 that the system has the conjugate complex roots. Therefore, it has given an unstable result at shaft torsional motion in uncontrolled system.

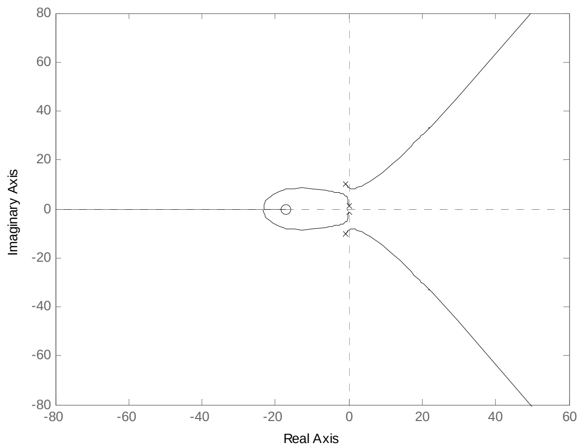
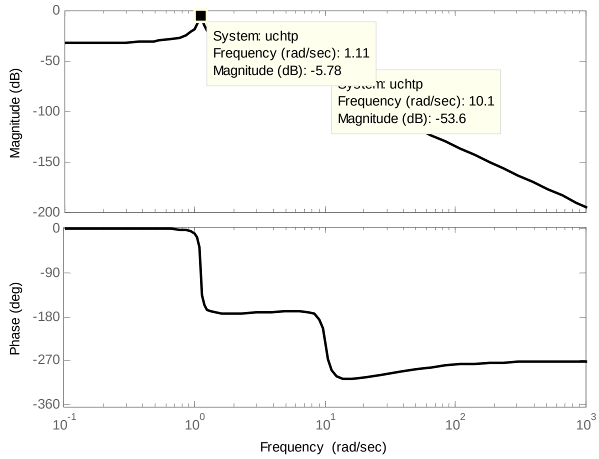


Fig. 3 Eigenvalues of shaft torsional motion



Fi**g.4** Frequency diagram of shaft torsional motion

The natural frequencies were determined as ω_{n1}=1.112 rad/s for shaft, and ω_{n2}=10.189 rad/s for propeller, and were demonstrated in Fig. 4.

3. PID Controller

3.1 Conventional PID Controller

PID controllers have been generally used successfully in feedback control applications (Khare Y. B. et al.v [12], Wei T. et al.[13], Xuan K. et al. [14]). The proportional, integrand and derivative component of a conventional PID controller produces an output signal that is directly proportional to the error, the difference between the measured variable and its reference value. The integrand component of a conventional PID controller ameliorates the final error of system response. The proportional action is generally the main drive in a control loop and reduces entire error. The derivative block has no effect on final error. However, this helps reduce overshoot and oscillation.

Typically, the closed loop diagram of feedback system is shown in Fig. 5.

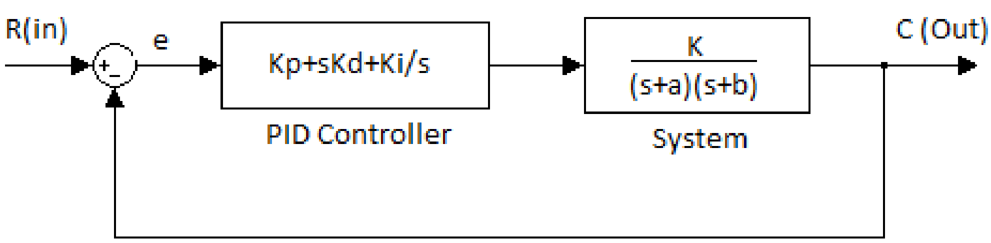


Fig.5 The model of closed loop system with conventional PID controller

In the PID controller block K_p, K_d and K_i are the constant that detonate proportional, derivative and integral part.

$$e = R(in) - C(out) \quad (10)$$

When the desired value called reference signal *R(in)* is equal to output signal *C(out*), the error *(e)* is zero. It means controller has adjusted system perfectly.

In order to make a fair comparison of closed loop system with conventional PID controller and modified PID controller the following system is assumed;

$$Gp(s) = \frac{K}{(s+a)(s+b)} \tag{11}$$

Where *a* and *b* are the roots that make system stable, *K* indicates system gain. The PID controller is obtained as follows;

$$Gc(t) = K[e(t) + \frac{1}{\tau_i}\int e(t)\,dt + \tau_d \frac{de(t)}{dt}] \tag{12}$$

Gp(s) and *Gc(t)* are assuming system transfer function and controller block, respectively. The Laplace transform of PID controller given in (12) becomes;

$$Gc(s) = Kp + \frac{Ki}{s} + sKd \tag{13}$$

After one more step PID controller yields below;

$$Gc(s) = \frac{s^2Kd + sKp + Ki}{s} \tag{14}$$

The general formulation of closed loop transfer function can be given as (15);

$$\frac{C(s)}{R(s)} = \frac{Gc(s)Gp(s)}{1 + Gc(s)Gp(s)} \tag{15}$$

According to (15) formulation by using G_p and G_s following equations can be expressed;

$$\frac{C(s)}{R(s)} = \frac{(s^2Kd + sKp + Ki)K}{s^3 + s^2(a + b + KdK) + s(ab + KpK) + KiK} \tag{16}$$

As shown in expression (16), it is obviously seen that the model of closed loop system with conventional PID includes two zeros.

3.2 Modified PID Controller

The conventional PID controller is the simplest form of controllers that utilize the derivative and integration operations in the compensation of control systems. The flexibility of this controller makes it easier to be used in many applications and many control problems (Hagiwara T. et al. [15], Visioli A., [16],). However, it has also some drawbacks such as instability which does not response desired values in nonlinear systems. So as to overcome this problem, PID controller was modified. The model of closed loop system with modified PID is shown in Fig. 6. Position of integral action which affects the difference of reference signal and feedback signal has remained unchanged on forward path. However, derivative and proportional actions have moved on feedback path to affect only the output signal (Ogata K. [8]). The aim of the control system is to activate the shaft vibration stabilizer to minimize torsional motion. In a practical, data collection hardware device (sensor) and software program are used. In such a system the torsional angle sensing device and the drive signal generated the control system. The signal is compared with the potentiometer signal, and the difference in signal forms the basis of the feedback control system.

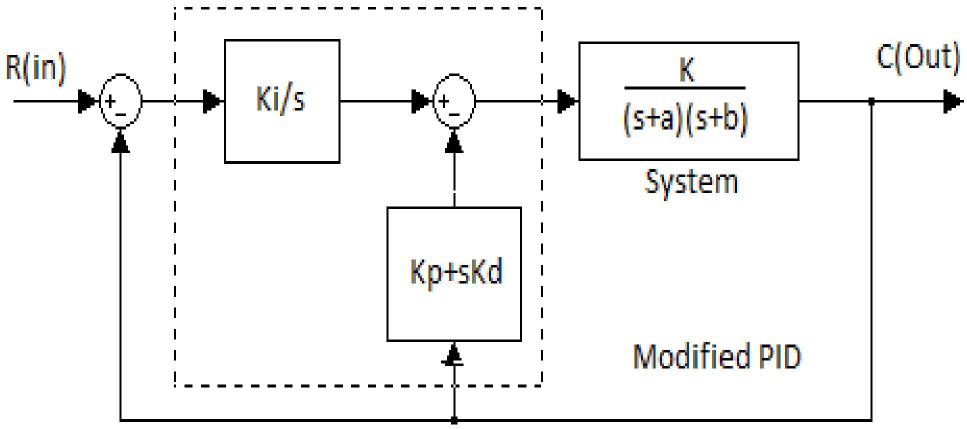


Fig.6 The model of closed loop system with modified PID controller (Ogata K. [8])

The modified PID controller is presented as follow;

$$Gc(t) = K[\frac{1}{\tau_i}\int e(t)dt - e(t) - \tau_d \frac{de(t)}{dt}] \tag{17}$$

The Laplace transform of PID controller given as follow;

$$Gc(s) = \frac{Ki}{s} - (sKd + Kp) \tag{18}$$

Thus, the modified PID controller can be indicated as follow;

$$Gc(s) = \frac{Ki - sKp - s^2Kd}{s} \tag{19}$$

Transfer function of closed loop system formulation is calculated according to (15) and given below;

$$\frac{C(s)}{R(s)} = \frac{KiK}{s^3 + s^2(a+b) + s(ab + KiKdK) + KiKpK} \tag{20}$$

There are two zeros at the system with conventional PID controller. It is hard to adjust system response due to these zeros. Their effect is occurred as earlier peak or higher overshoot. The proposed modified PID controller recovers from these effects and ameliorates system response by adding proportional and derivative blocks of PID on feedback path instead of on forward path. Therefore better solution of system response is achieved in modified PID in comparison with conventional PID.

4. Torsional Vibration Analysis and Simulation

In this study, a fishing boat which is 20 m in length, 5.7 m in width, 2285 m in depth and in MAN 8L 32/40 is considered as the main engine which consists of 8 cylinder diesel engine. It is assumed that the engine is operated at the constant speed of 750 rpm. The specific parameters about this engine are all given in Table 2 to calculate the torsional vibration. The considered ship propulsion system is shown in Fig.7.

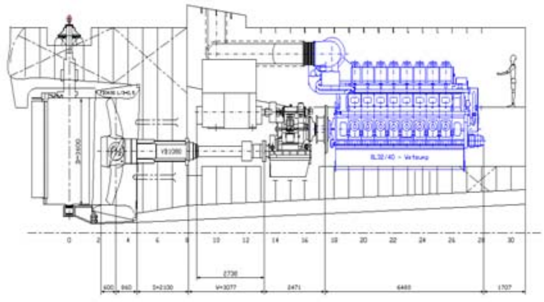


Fig. 7 Based on ships and main engine, propulsion systems (MAN [17])

Table 2 Main engine-propeller and shaft- particulars

MAN 8 L 32/40	*Values*
Piston Stroke	*40 cm*
Cylinder bore	*32 cm*
Number of cylinders	*8*
Speed	*750 rpm*
Propeller mass	*7000 kg*
Reduction Gear mass	*5700 kg*
Power	*4000KW*

Ship main engine, shaft and propeller system was analysed with uncontrolled, conventional PID and modified PID controlled structures.

Gains of the PID controller (K_p, K_i and K_d) are obtained using Ziegler-Nicholes method (Ogata K. [8]). Modified and conventional PID controller's gains are indicated in Table 3.

Table 3 The PID controller parameters

Controller Gains	*Modified PID*	*Conventional PID*
Kp	*51*	*1*
Ki	*14*	*0.092*
Kd	*38*	*2.27*

Fig. 8 shows the torsional angular displacement of the shaft system in case of using conventional PID controller, modified PID controller and uncontrolled model. Shaft torsional displacement has been decreasing up to 0.2 rad point at the uncontrolled condition, so the displacement is insufficient in the uncontrolled situation. System has responded quicker at the implementation of the conventional PID controller; however shaft vibrations oscillate randomly as in the uncontrolled situations at the first 8 seconds. Then displacement curve proceeds to 0.05 rad with little amplitudes. After performing modified PID controller the reference input has been rapidly achieved. It can be stated that modified PID are best solution for shaft angular displacement.

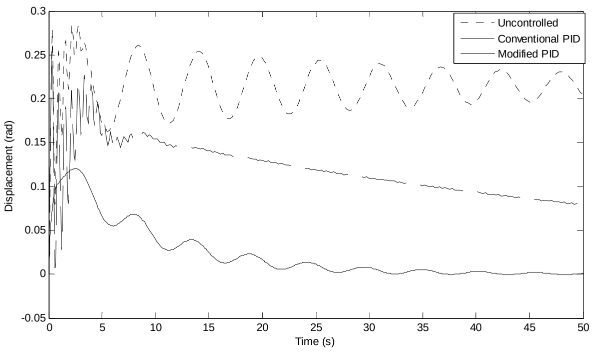


Fig. 8 Shaft torsional angular displacement at shaft

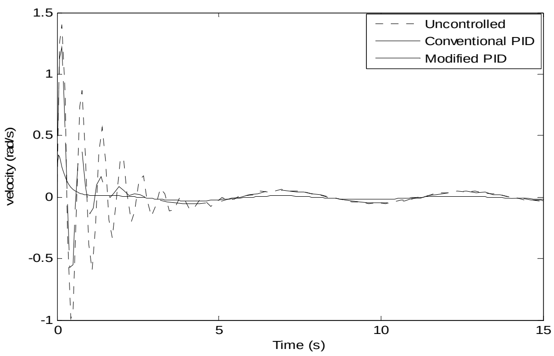


Fig. 9 Shaft torsional angular velocity change at shaft

Shaft torsional velocity is shown in Fig. 9. Values of angular velocity have been changed from 1.5 rad/s to -1 rad/s in the first second at the uncontrolled situation. Then velocity come close to 0 rad/s 50 seconds later and these oscillations follow almost the same line as conventional PID. Conventional PID controller has not affected the system adequately.

The response of system has significantly fast and amplitudes directions of the line turned to zero after 2 seconds in modified PID. It is obvious that modified PID controller affects the rising time of system effectively.

Shaft torsional acceleration is shown in Fig.10. Acceleration has oscillated significantly high value and frequently acted as a curve in the first seconds of simulation in uncontrolled model. Conventional PID controller cannot decrease oscillations as satisfactorily as modified PID controller. System has responded magnificently with modified PID controller.

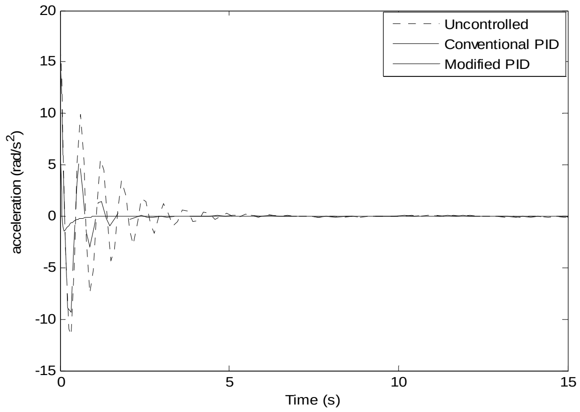


Fig. 10 Shaft Torsional angular acceleration change at shaft

Propeller displacement is shown in Fig. 11. Displacement amplitude has behaved continuously as sinusoidal without any controller. By applying conventional PID controller system has responded quicker and approach to the input reference value. System has not responded as promptly as desired conventional PID, but amplitude has settled to reference line more quickly after some small fluctuation in modified PID controller.

Propeller velocity is shown in Fig. 12. Angular travel of propeller velocity has acted as sinusoidal and rate of velocity are high level at the beginning stage. The conventional controller has affected efficiently the system. Amplitude has reached to reference signal in the first 10 second, but the velocity has too many oscillations at beginning stage. It appears that these amplitudes have more vibration than uncontrolled model. The quality performance of system has been observed through applying modified PID controller. The velocity has oscillated between 0.1rad/s and -0.1rad/ s in the first two second then settled to zero line after 10 seconds. It has demonstrated that modified PID controller has capability to damp the vibrations of the system.

Propeller torsional acceleration is shown in Fig. 13. Oscillations of propeller accelerations have acted as sinusoidal with decreased curve in uncontrolled applications. The acceleration value has indicated 0.5 rad/s^2 for first seconds. Then it has reached up to reference line after 10 seconds with conventional PID controller. But results showed that acceleration signal has settled to the reference signal more quickly. So it has demonstrated better rising time. Furthermore, there have been fewer oscillations with modified PID controller.

It is shown in Fig. 14 that uncontrolled system and with conventional PID system have crisp change in the first natural frequency 1.112 rad/s. However, the soft crossing has been seen with modified PID system. On the other hand, each system has significant change in the second natural frequency.

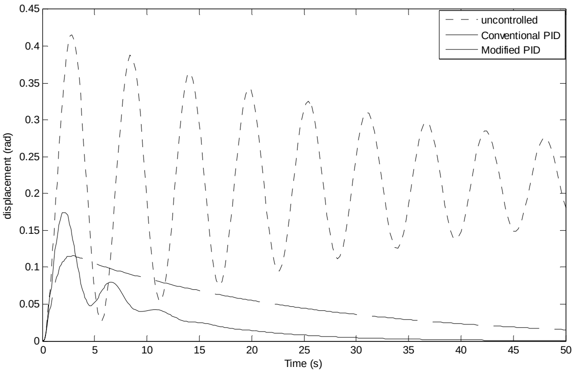


Fig. 11 Propeller torsional angular displacement

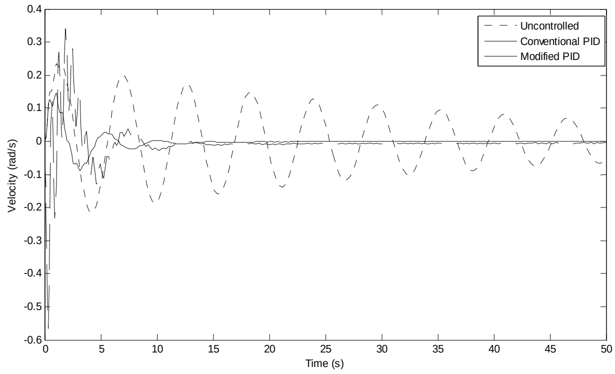


Fig. 12 Propeller torsional angular velocity

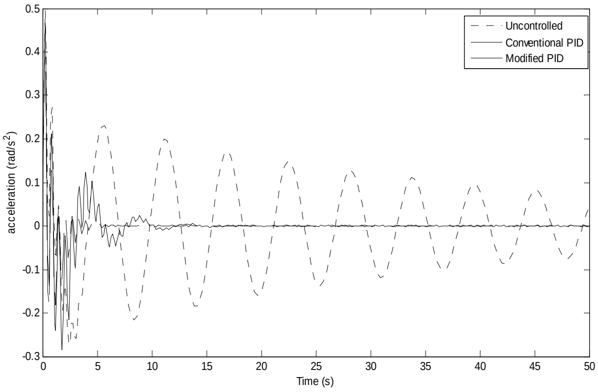


Fig. 13 Propeller torsional angular acceleration

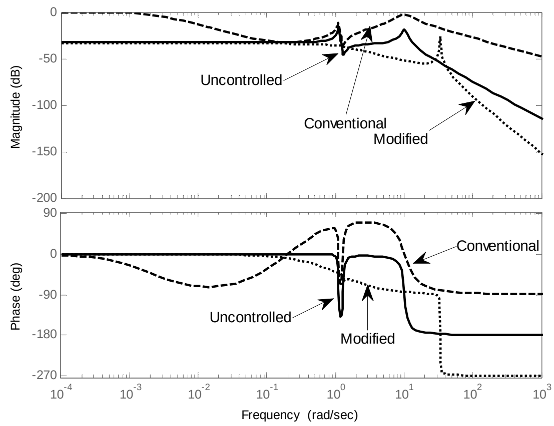


Fig.14 Frequency diagram of displacement of shaft motion

5. Conclusions

In this study, control of a ship shaft torsional vibration by modified PID controller was proposed. Lagrange equation was used to obtain mathematical model of dynamic system. The conventional and modified PID controller was applied to the system. It was deduced that displacement, velocity and acceleration of main engine shaft propeller systems can be substantially reduced by controller. The simulation results of modified PID controller for whole ship propulsion system in case of some interior and exterior effect have been demonstrated and compared with conventional PID along with the uncontrolled system in Table 4. It was observed that conventional PID controller has enhanced displacement of shaft torsional at the rate of %26. Modified PID controller improved displacement of shaft torsional in the ratio of %81. As a result, it is obviously clear that proper performance has been achieved in Modified PID controller.

Table 4 The shaft variable values at 10 second

Shaft	*Modified PID*	*Conventional PID*	*Uncontrolled*
Displacement (rad)	*0.039*	*0.1541*	*0.2110*
Velocity (rad/s)	*0.0183*	*0.0190*	*0.0497*
Acceleration(rad/s^2)	*0.0045*	*0.1247*	*0.0289*

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