Conventional PID and Modified PID Controller Design for Roll Fin Electro-Hydraulic Actuator

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Abstract: The aim of this study is to decrease a fishing boat's roll motion, which is adversely affected by disturbing hydrodynamic forces, by applying fin roll stabilizer. It is ensured that roll motion with nonlinear damping and restoring moment coefficients reaches the desired level by means of classical PID and modified PID algorithms. By Lyapunov's direct method, the key issue- stability which is needed during the course of sailing was examined considering initial conditions, and it was realized that the system was generally stable. Besides, NACA 0015 model was utilized for the fin roll stabilizer, and flow analysis was conducted by CFD method. According to the simulation results, when the same gains were practiced, modified PID controller algorithms were rather more efficient than conventional PID in the roll fin stabilizer system.

Keywords: modified PID control; fin stabilizer; Lyapunov direct method

1   Introduction

For many years, vessels have remained as a research subject for researchers as they lack fundamentals of stability despite meeting the requirements of current laws. It is known that a ship on route is likely to be off the stable zone due to disrupting hydrodynamic forces of a ship roll motion. Grim [1] modelled the roll equation using the gradual change in the restoring moment among the waves. In another study of his, he analyzed unstable roll equation in a more general equation form using Mathieu equation.

Advanced studies on roll motion were carried out by Dalzell [2], and alternative damping models were put forward. Soliman and Thomson [3] solved the nonlinear differential equation of roll motion through 4th level Runga-Kutta method. How damping effect of roll motion stability changed owing to disrupting forces and frequency was studied. Haddara and Wang [4] pointed out that quadratic damping model of speed for small fishing ships came up with good results. Taylan [5] studied a nonlinear roll motion model and the solution of this model through generalized Krylov-Bogoliubov asymptotic method on a time line. In this model, nonlinear restoring terms were considered as a 3rd level polonium; likewise, nonlinear damping was regarded as a 2nd level term. Taylan [6] studied on generalization of the nonlinear equation form indicating roll motion of a ship sailing amidst waves by means of Duffing method on frequency level. Surendran and Reddy [7] discussed the roll dynamics of a Ro-Ro ship taking into account many types of combination of loads in linear and nonlinear forms.

Safety of voyage has to be assured against the disrupting hydrodynamic effects of passengers as well as cargos, and roll amplitude has to be at an acceptable level. Therefore, a number of applications such as fin roll stabilizers and U-tube have been used in literature [8, 9].

Sgoppo and Persons [10] demonstrated that roll motion can be reduced by 35% through fin roll stabilizer system based on WMEC 901 class ship with 3rd degree of freedom model. Surendran et al. [11] pointed out that nonlinear roll motion could be minimized with the fin roll stabilizer based on PID (Proportional-Integral-Derivative) control algorithm considering a frigate warship. In this study, he acquired lift characteristics of the fin roll stabilizer stemming from hydrodynamic flow using CFD (Computational Fluid Dynamics). In simulations for different occasions, an 80% reduction in roll value was observed.

Guan and Zhang [12], who expressed nonlinear damping terms with the Backstepping Integrator theory, controlled the nonlinear fin roll stabilizer by means of PID and close loop algorithms by ignoring high level nonlinear terms. Ghassemi et al. [13] expressed that fin roll stabilizers are effective components in reducing roll amplitude out of wave movements. It was pointed out that lift forces occurring around the angle of attack and the fin roll stabilizer create a reversed moment. That's why he showed roll amplitude could be reduced using PID and Neural Network combination to control attack angle. Karakaş et al. [14] designed a controller based on Lyapunov method for fin roll damping systems in beam seas. The roll motion of the ship was considered as a single degree of freedom and 3rd degree nonlinear terms were added to the model. It was mentioned that the likelihood of capsize was reduced to a considerable extent by means of the controller based on Lyapunov method. PID controllers' versions have been investigated by many researchers Alfaro et al. [24], Precup et al. [25], Hadlovska and Jajcisin [26].

In this paper, the problem of a nonlinear roll motion of fishing vessel is considered. Section 2 deals with mathematical model of nonlinear roll motion. Section 3 analyses stability of roll motion by Lyapunov's Direct Method. Section 4 describes the fin roll stabilizer system and classical PID and the modified PID controller. Nonlinear roll motion of a fishing boat, which will be capable of operating in the Black Sea and the Mediterranean Sea, was controlled by fin stabilizer system. Section 5 discusses simulation results.

2   Mathematical Model of Nonlinear Roll Motion

In order to ease to design mathematical model of ship motions, some important assumptions are made in modelling a fishing boat rolling motion, neglecting all other degrees of freedom of ships, xz-plane symmetry, rigid body and homogenous mass distribution. Considering some simplifications, the following nonlinear expression for the roll equation is obtained,

$$(I+J)\,\ddot{\varphi}+B_1\dot{\varphi}+B_2\,\dot{\varphi}\,|\dot{\varphi}|+\Delta\,(\,c_1\varphi+c_3\,\varphi^3+c_5\,\varphi^5\,)=\omega_e^2\,\alpha_m\,I\cos(\,\omega_e t\,)-M_f \qquad (1)$$

where $\varphi,\dot{\varphi},\ddot{\varphi}$ are angle, angular velocity and angular acceleration of roll motion, respectively. I and J are the mass moment of inertia and the added mass moment of inertia, respectively. B_1,B_2 are roll damping coefficients, c_1,c_2,c_3 are determined by restoring force coefficients and Δ is the weight displacement of the ship, ω_e is encounter frequency of the wave, α_m is the maximum wave slope, M_f is the control moment of active fins. Dividing the equation (1) throughout by $(I+J)$,

$$\ddot{\varphi}+b_1\,\dot{\varphi}+b_2\dot{\varphi}\,|\dot{\varphi}|+\Delta(c_1\,\varphi+c_3\,\varphi^3+c_5\,\varphi^5\,)=\frac{\omega_e^2\,\alpha_m\,I\cos(\,\omega_e t\,)}{I+J}-\frac{M_F}{I+J} \qquad (2)$$

Inertia moment arises due to the ship's reaction to a movement and it is proportional to the acceleration of motion. Added inertia moment is a reaction of sea water to the ship motion. These inertia values can be calculated depending on weight displacement of ship (Δ), breadth (B) and the vertical distance of the center of gravity (KG),

$$(I+J)=\frac{\Delta}{12\,g}(\,B^2+4\,KG^2\,) \qquad (3)$$

The roll damping coefficient for a ship hull form has several contributions. These components are considered as skin friction of the hull, eddy shedding from the hull, free surface waves, lift effect damping and bilge keel damping. Theoretical and semi-empirical methods have been used to evaluate the roll damping by Ikeda

et al. [15] and Ikeda [16]. A non-dimensional damping coefficient for different ship types is expressed as follows:

$$B_1 = \frac{2a\sqrt{(I+J)\Delta GM}}{\pi} \tag{4}$$

$$B_2 = \frac{3}{4}b(I+J) \tag{5}$$

These coefficients are directly related to a linear damping coefficient B_1 and a non-linear damping coefficient, B_2 represents quadratic drag [2]. The non-dimensional damping coefficients for fishing boat were obtained as follow (*a=0.1, b=0.0140*). The curve for righting arm has been represented by the polynomial,

$$M(\varphi) = c_1\varphi + c_3\varphi^3 + c_5\varphi^5 + ...$$

where $c_1 > 0$, $c_3 < 0$, $c_5 > 0$. The roll restoring moment coefficients are defined by ref. [6]

$$c_1 = \frac{d(GZ)}{d\phi} = GM \quad c_3 = \frac{4}{\phi_v^4}(3A_{\phi v} - GM\phi_v^2) \quad c_5 = -\frac{3}{\phi_v^6}(4A_{\phi v} - GM\phi_v^2) \tag{6}$$

Angle of vanishing stability ϕ_v, area under the *GZ* curve $A_{\phi v}$, and dynamic characteristics of the *GZ* curve such as metacentric height *GM*. Based on the above-mentioned coefficients, numerical calculations were carried out for a fishing boat, whose body plan is given in Fig. 1.
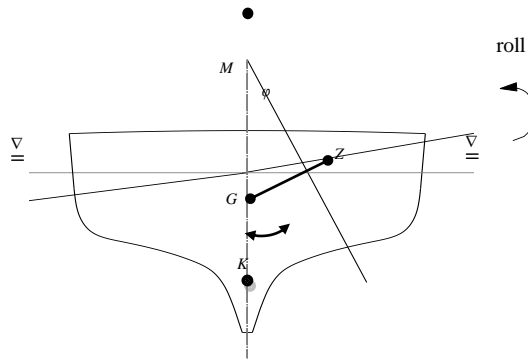

Figure 1
Body plan of the fishing boat

The righting arm curve is a graphical representation of the fishing boat's stability in Fig. 2 [17]. From this figure, it can be seen that Hydrodynamic lift effect develops in an approximately linear manner with an increasing angle of attack.

The area under the curve is an indication of the fishing boat's ability to counter the capsizing moments acting on the boat.
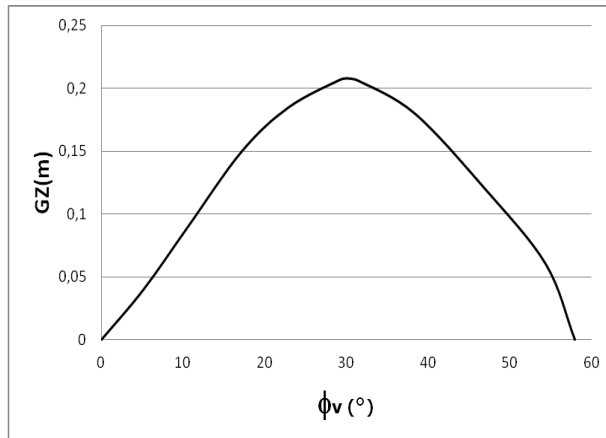

Figure 2
Righting arm curve of fishing boat

Due to the influences of high wave frequency, ships sailing on the sea produce undesirable roll motion. It is expressed the wave moment to consist of the encounter frequency as,

$$M_w = \omega_e^2 \, \alpha_m \, I \, cos\,(\,\omega_e\,t\,) \tag{7}$$

$$\omega_e = \omega_w - \frac{\omega_w^2}{g} V \, cos\,(\,\mu_w\,)$$

M_w is the wave moment, ω_e is the encounter frequency of the wave, ω_w is the wave frequency, α_m is the maximum wave slope, μ_w is the wave encounter angle of the ship. It can be envisaged that the wave excitation will depend not only on amplitude and frequency of the waves but also on encounter angle and speed.

3   Stability Analysis of Roll Motion by Lyapunov's Direct Method

A fishing boat is called stable when it has enough positive stability to counter the external forces generated by current weather, fishing conditions and it will return to its upright position [18]. Lyapunov's Direct Method was used for stability analyses by Ozkan [19]. This is a very powerful and applicable technique since it

does not require any knowledge about the explicit solutions of the equations. By using state variables of equation (1),

$$\dot{\varphi} = \dot{\varphi}_1 = \varphi_2$$

$$\dot{\varphi}_2 = -(\omega_0^2 \varphi_1 + m_3 \varphi_1^3 + m_5 \varphi_1^5) - (b_1 \varphi_2 - b_2 \varphi_2 |\varphi_2|) \qquad (8)$$

Lyapunov function V_x satisfying

- $V_x > 0$ positive definite and $V(0)=0$

- $\dfrac{dV(\vec{x})}{dt} \leq 0$

- $V(x) \to \infty$ as $\|x\| \to \infty$

Lyapunov second method will be used to test the system stability.

$$\dot{V}(t) = \frac{dV(\vec{x})}{dt} = \nabla V^T \dot{\vec{x}}$$

If symmetric coefficients are assumed equal to zero, derivative of the Lyapunov function is negative. Lyapunov function is obtained depending on the non-linear roll damping coefficient. If this value is smaller than zero, non-linear roll motion can be said to be stable.

$$\dot{V}(x) = -\varphi_2^2 (b_1 + b_2 |\varphi_2|) < 0 \qquad (9)$$

$$V(x) = \omega_0^2 \frac{\varphi_1^2}{2} + m_3 \frac{\varphi_1^4}{4} + m_5 \frac{\varphi_1^6}{6} \qquad (10)$$

4   Controller Design for Roll Fin Actuator

The objective of the control is to generate the input current such that the angular position of the control fin is regulated to the desired position. The motion of a ship can be affected by fin actuators that impart forces and moments. Actuators play a very important role in the control system structure. When the roll fin stabilizers attack to the fluid, it can bee seen that the lifting force caused by the rotation and the angle of attack occurs on the surface of fins. The lift force and the lift in non-dimensional form are as in the following form in Fig. 3 [8],
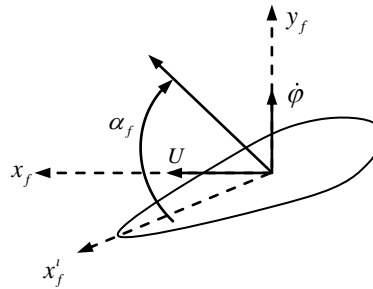
Figure 3

Ship roll fin stabilizer

$$L = \frac{1}{2}\rho V A_F C_L \tag{11}$$

$$C_L = \frac{L}{0.5\,\rho V^2 A_F} \tag{12}$$

where L lifting force (N); ρ density of fluid (t/m^3); A_F fins area (m^2); C_L fins lift coefficient (lift coefficient/rad); V the ship speed (m/s). General formulas of fin roll stabilizer are expressed as the following equations:

$$M_F = \rho V^2 A_F C_L I_F (\,\alpha_F + \frac{\dot{\varphi}}{v}\,I_F\,) \tag{13}$$

where M_F, fin roll stabilizer moment; I_F the fins force arm; α_F attack angle. The electro-hydraulic system dynamics of fin stabilizer system are assumed to be governed by

$$\dot{\alpha}_F + t_1\,\alpha_F = t_2\,u_F \tag{14}$$

where α_F is the actuator output (actual fin angle), u is the input to the electro-hydraulic systems. The hydraulic control model presented in Fig. 4. Conrol surfaces are commanded by hydraulic machinery that implement the action demanded by controller.
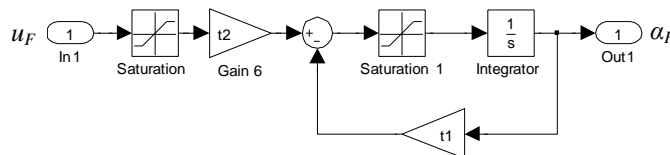

Figure 4

Block diagram of fin stabilizer system

As can be seen in Fig. 4, a saturator has been intermitted between the controller and the fin actuator.

This element models the saturating behaviour of the controller. The foil motion is constrained to move within certain maximum angle.

$$-\alpha_{F\,max} \leq \alpha_F \leq \alpha_{F\,max}$$

The results of hydrodynamic lift coefficient in function of attack angle of fin and roll angle of ship are presented in Fig. 5.
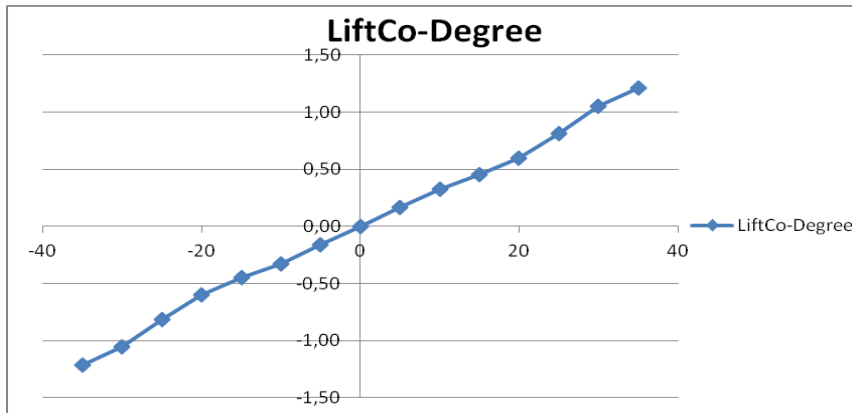

Figure 5
Lift coefficient

Computational fluid dynamics analysis has been carried out with Star CCM + package software, via 8 Parallel Processor and 24 GB of RAM hardware configuration. Preprocessing, running and finishing parts were integrated in the program. At the time of Preprocessing, Trimmer, Surface Remesher and Prism Layer Masher solution mesh properties were executed simultaneously. Hexahedral meshes analyses were conducted. Variations with 170000, 970000 and 1500000 cells were generated respectively for our three dimensional model and afterwards, the solution continued with the one having 970000 cells 2900000 surface via mesh independence. A sample case of free surface shape around fin is presented in Fig. 6.
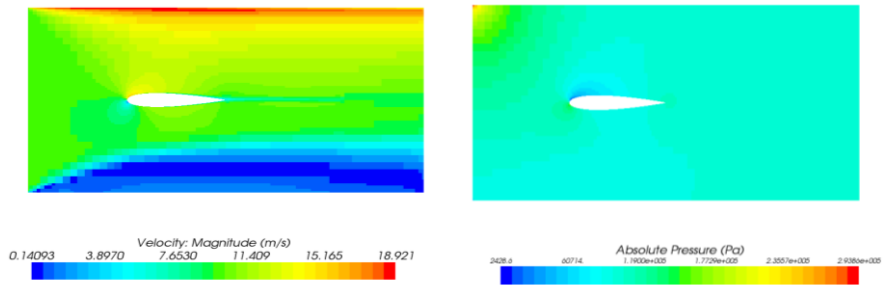

Figure 6
Velocity and pressure gradient cross section (attack angle *+30°*)

We considered only a magnitude constraint for the mechanical angle of the fins as *30°*. Segregated flow, Reynolds averaged Navier Stokes and k-ε turbulence model were used as the solvers for running process. Convergence conditions were approximately *1E-5* (10^{-5}). Running was carried out in parallel with 8 cores. During the final process, along with the velocity and pressure gradients, the lift force coefficient was directly taken from the model. This coefficient was obtained depending on fin roll stabilizer's attack angle (from *-30°* to *+30°*).

4.1 Conventional PID Controller

The parameters of PID controller lead to different effects on system characteristics. The proportional block provides an overall control action, and the integral block reduces steady-state errors. On the other hand, the derivative block improves transient response. The basic elements of a PID controller for a ship roll motion control system are shown in Fig. 7 [20].
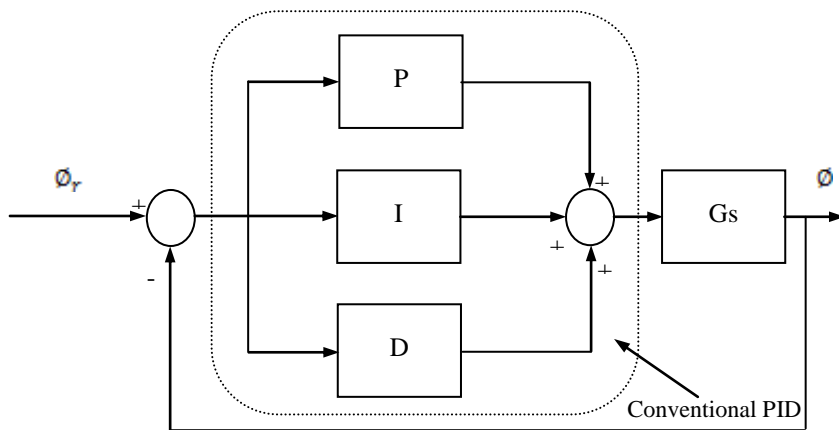

Figure 7
Closed loop system with conventional PID controller

Generally, transfer function of classical PID controller is given below:

$$G_{pid}(t) = K_p \left[e + \frac{1}{\tau_i} \int d(t) + \tau_d \frac{de(t)}{dt} \right] \qquad (15)$$

Where $\tau_i = \dfrac{K_p}{K_i}$, $\tau_d = \dfrac{K_d}{K_p}$ and *e* is the error between the reference and the output of a system, τ_i is the integral time, τ_d is derivative time. K_p, K_i and K_d are

proportional, integral and derivative gains, respectively. The conventional PID controller is the simplest form of controllers that utilize the derivative and integration operations in the compensation of control systems. Thanks to its flexibility, it is easier to use this controller in many applications and many control problems [21, 22, 23, 24, 25, 26, 27 and 28]. The model of closed loop system with modified PID is shown in Fig. 8.

4.2   Modified PID Controller

The 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. The objective is to force the system's output to follow a given bounded reference value.
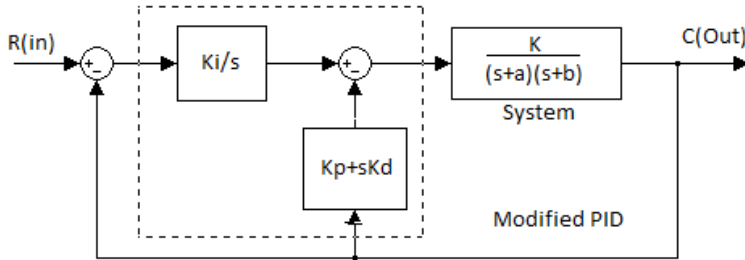

Figure 8

The model of closed loop system with modified PID controller

The modified PID controller can be indicated as follows:

$$G_c(s) = \frac{K_i - sK_p - s^2 K_d}{s} \tag{16}$$

Transfer function of closed loop system is given below:

$$\frac{C(s)}{R(s)} = \frac{K_i K}{s^3 + s^2(a+b) + s(ab + K_i K_d K) + K_i K_p K} \tag{17}$$

There are two zeros in the system with conventional PID controller. It is hard to adjust the system response due to these zeros. Their effect occurs as earlier peak or higher overshoot. The proposed modified PID controller gets over these effects and ameliorates the 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.

5   Simulation

In control application, mathematical model was presented to perform numerical simulations of different scenarios, and to obtain the performance of the fin stabilizer system. These numerical simulations were applied with Matlab simulink program. The fishing boat and the fin (NACA 0015) particulars are shown in Table 1.

Table 1

The fishing boat and the fin (NACA 0015) particulars

Principal Particulars	Symbol	Parameter
Length between perpendiculars	L_{bp}	20 m
Breadth	B	5.714 m
Depth	D	3.2 m
Draught	T	2.285 m
Displacement	∇	119.34 m³
Metacentric height	GM	0.57 m
Vertical center of gravity	KG	2.4 m
Block coefficient	C_B	0.457
Service speed	V	10 knot
Fins area	A_F	2.5 m²
Fins lift coefficient	C_L	0.65
Vanishing angle of stability	ϕ_v	58^0

Non-dimensional damping and restoring moment coefficients for fishing boat are expressed in Table 2.

Table 2

Non-dimensional damping and restoring moment coefficients for fishing boat

b_1	b_2	m_1	m_3	m_5
0.069	0.010	1.204	-1.80	0.61

The flow chart of MPID control system is displayed in Fig. 8. The simulation results for fin roll stabilizer system show roll angle, roll velocity and phase diagram in Figs. 9, 10 and 11. Comparisons of the control performance were made between conventional PID and the MPID controllers. K_p, K_i and K_d control values were obtained by trial method. The values of PID gains $K_p = 0.2145$, $K_d = 1.2288$, $K_i = 2.89$, ensured good roll reduction. The modified PID control response of the fin roll stabilizer is better than PID control.
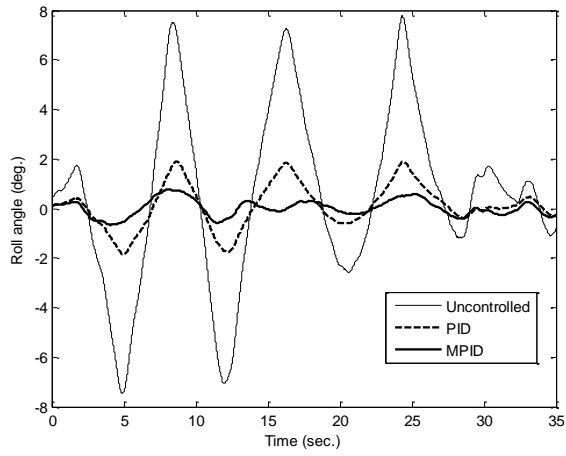
Figure 9

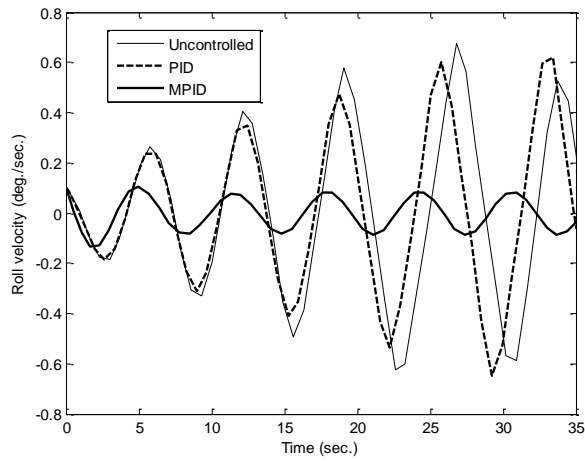Comparison of roll angle response

Figure 10
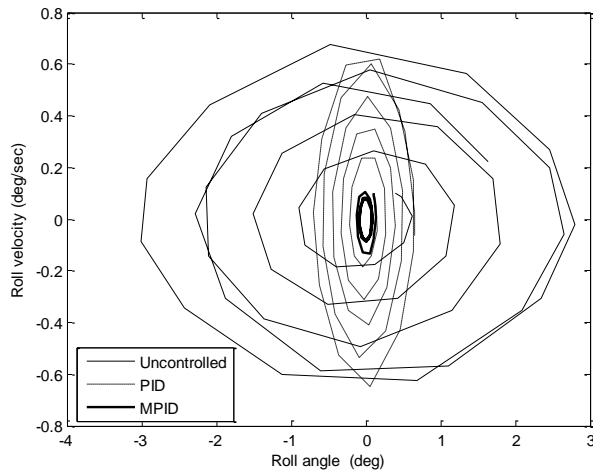
Comparison of roll velocity response
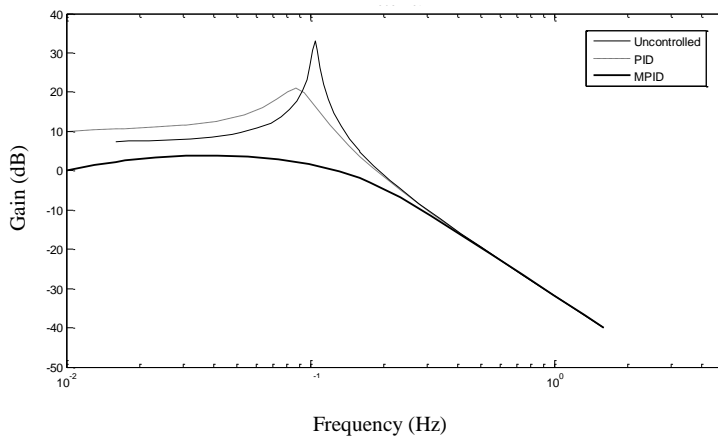
Figure 11

Comparison of phase diagram

Figure 12

Frequency response of the roll motion for uncontrolled, PID and MPID

Since the aim of the controller was to control the roll motion, the improvement has been realized particularly around 1.1 Hz at low frequencies. This is also demonstrated by plotting the frequency response of controlled and uncontrolled body bounce acceleration in Fig. 12.

The comparison of the two controllers is presented in Table 3, which shows roll angle and roll velocity. The settling time for uncontrolled is longer compared to the time for settlement of MPID.

Table 3

Performance Comparisons

Controller	Max. roll angle (deg.)	Max. roll velocity (deg./s)
Uncontrolled	7.8^0	0.7
PID	1.9^0	0.6
MPID	0.7^0	0.1

Conclusions

This paper introduces a mathematical modeling, and examines controlling of nonlinear roll motion by applying fin stabilizer system. Nonlinear restoring and damping moment coefficients are computed by means of empirical equations. Nonlinear roll motion stability of fishing boat is examined through Lyapunov direct method. In the course of the simulation, it is accepted that the control gains receive the same values for PID and MPID algorithms. According to the results of the simulation, it is notable that MPID controller has shown a considerable amelioration in roll magnitude about *91%*. The performance of MPID controller performance has been *15.4%* greater than conventional PID as shown in Table 3.

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