International Journal of Physics, 2016, Vol. 4, No. 6, 181-186
Available online at http://pubs.sciepub.com/ijp/4/6/5

DOI:10.12691/ijp-4-6-5

Control of Roll Motion of Fishing Vessel by Fin-Stabilizer Using PID Controller

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Abstract The aim of this study is to diminish the roll motion of the fishing vessel using fin-roll stabilizer. In this regard, lift coefficient of the fin and the hydrodynamic coefficients of the roll equation are calculated by empirical formulas. In effect, constrained LQR (Linear Quadratic Regulator) controller is designed and used to control the roll motion in the presence of operational constraints of fin's actuator. In order to boost the validity of our results, the performance of this controller is compared with a conventional PID (Proportional-Integral-Derivative) controller. Finally, simulation results indicate the significant amount of reduction in roll amplitude.

Keywords: *PID controller, fishing vessel, fin roll stabilizer*

Cite This Article: Hassan Ghassemi, Hamid Malekizade, and Arash Ashrafi, "Control of Roll Motion of Fishing Vessel by Fin-Stabilizer Using PID Controller." *International Journal of Physics*, vol. 4, no. 6 (2016): 181-186. doi: 10.12691/ijp-4-6-5.

1. Introduction

Among the motions of a ship at sea, roll motion is the most important one. The accelerations due to wave-induced roll motions negatively influence on a fishing vessel performances by making limitation in comfort, workability and safety. The roll stabilization systems have been widely studied for more than three decades and various types of anti-rolling devices have been introduced to reduce the undesirable roll motion [1,2]. The active fin stabilizer has been considered as the most effective anti-rolling technique for ships which normally operates above certain speeds. It reduces roll motion by controlling the mechanical angle of the fin according to the ship roll angle and roll rate [3].

The studies of the nonlinear roll motion model have been done by Taylan [4]. In those models, nonlinear restoring terms were considered as a third order polynomial; likewise, nonlinear damping was regarded as a second-order polynomial. In research of Surendran et. al. [5], the roll dynamics of a Ro–Ro ship taking into account the many types of combinations of loads in linear and nonlinear forms. A lift feedback fuzzy-PID control method was developed to better deal with these problems, and this lift feedback fin stabilizer system was simulated under different sea condition [6]. The roll amplitude of the fishing vessel under the wave effect was analyzed using the nonlinear mathematical model in Alarçin's paper [7].

The PID controller has mainly been used in the ship fin stabilizers. The results of using a combined neural network and PID for roll control of ship with small draught were presented [8]. Modified PID control design was presented for roll fin actuator of nonlinear modeling of the fishing boat [9], roll damping characteristics of a trimaran displacement ship [10]. However, because of difficulty, non-linearity, and constriction of fin stabilizers in this method, attaining the optimal performance for control system is very intricate [11]. A multi-input multi-output (MIMO) optimal control system that has two control inputs such as fin stabilizers and pod propellers is designed. The linear quadratic regulator (LQR) control algorithm is applied to reduce the roll motion of cruise ships in regular waves [12]. A robust fin controller based on L2 gain design is proposed, in order to reduce the roll motion of surface ships. The plant consists of the ship roll dynamics and that of the fin actuator [13].

The purpose of this paper is the nonlinear modeling and its coefficients extraction as well as designing constrained controller for fin-roll stabilizer in a fishing vessel. To achieve this goal, the active fin stabilizer with NACA0015 section is used as the fin-roll stabilizer. Also, CFD method is used for the flow analysis so that the lift coefficient would be extracted which, in turn, is validated by use of empirical formulas. The roll motion model is derived in the presence of irregular waves. In this model, the nonlinear restoring moment is considered as a 3rd order polynomial and its coefficients are calculated by the GZ curve and the empirical formulas. A nonlinear term is also considered for the roll damping moment. Furthermore, the linear damping coefficient, mass moment of inertia and added mass moment of inertia are calculated by using the Free Roll decay test and empirical formulas.

Finally, the constrained LQR is designed as optimal stabilizer to control the roll motion. In order to fortify the results of this study, the performance of this controller is compared with a conventional PID controller. Simulation results, demonstrated in time domain, are presented to show the effectiveness of the constrained LQR.

2. Nonlinear Modeling

It is assumed that the fin stabilizer is approximately located in amidships close to the center of gravity and there is a minimum coupling with the other motions. Consequently, the coupling with other motions can be regarded and just one degree of freedom can be considered. The model of roll motion can be represented as

$$\begin{aligned} I_{xx}\ddot{\phi} &= \tau_d - \tau_f - \tau_h \\ p &= \dot{\phi} \end{aligned} \tag{1}$$

where ϕ, p, I_{xx}, τ_f, τ_h and τ_d are roll angle, roll rate, roll moment of inertia, moment created by fins, hydrodynamic moment and wave excitation moment, respectively. The moment due to the fins and the hydrodynamic moment are calculated for a fishing vessel with characteristics which presented in the Table 1 and the body plane illustrated in the Figure 1.

Table 1. The specifications of the fishing vessel

Item	Value
Length (m)	20
Breadth (m)	5.714
Draft (m)	2.854
Vertical center of gravity (m)	2.4
Metacentric height (m)	0.57
Service speed (m/s)	12.5
Block coefficient	0.457

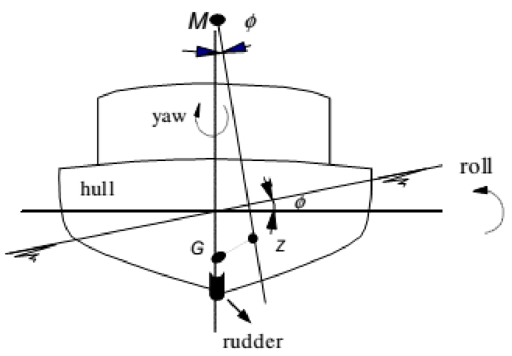

Figure 1. Fishing vessel at roll angle

2.1. Fin Data

The coefficients of fin stabilizer which conisdered in this study are presented in Table 2.

Table 2. Characteristics of the fin

Item	Value
Fin area (m^2)	3.7
Fin span (m)	1.4
Mean chord (m)	2.2
Fin lever (m)	5.8
Aspect ratio	0.72
Fin section	NACA0015

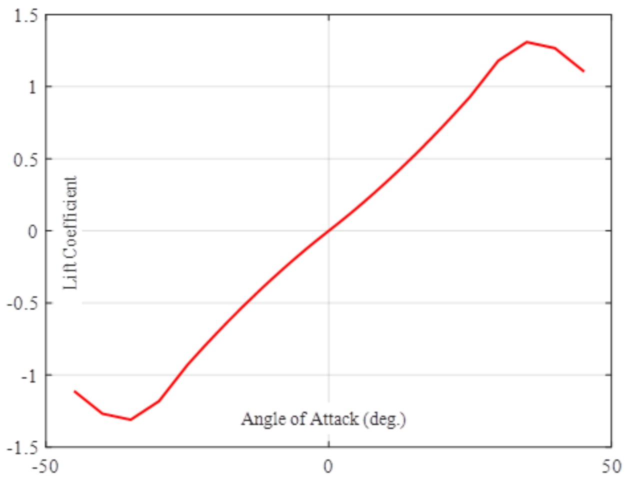

Figure 2. Lift coefficient of the fin

The fins moment can be determined by the Eq. (2).

$$\tau_f = \rho V^2 A_f R_f C_L(\alpha_e) \tag{2}$$

where ρ, V, A_f, R_f and $C_L(\alpha_e)$ are the density of the flow, ship speed, fin area, fin lever, and lift coefficient respectively. It should be noted that lift coefficient is a function of the effective angle of attack between the flow and the fin (Figure 2).

2.2. The Hydrodynamic Moment

The hydrodynamic moment due to the interaction between fluid and ship is defined by Eq. (3).

$$\tau_h = K_{\dot{p}}\dot{p} + f_1(\phi,\dot{\phi}) + f_2(\phi) \tag{3}$$

where $K_{\dot{p}}\dot{p}$ regards a hydrodynamic moment in roll due to pressure variation that is proportional to the roll accelerations and the coefficient $K_{\dot{p}}$, called roll added mass. The $f_1(\phi,\dot{\phi})$ is damping term and it can be represented as Eq. (4).

$$f_1(\phi,\dot{\phi}) = k_p\dot{\phi} + k_{p|p|}\dot{\phi}|\dot{\phi}| \tag{4}$$

where $k_p\phi$ is a linear damping term which includes forces due to wave-making and linear skin-friction. Besides, the coefficient k_p is denoted a linear damping coefficient. The $k_{p|p|}\dot{\phi}|\dot{\phi}|$ is a nonlinear damping term, which contains moments due to viscous effects, alike nonlinear skin friction and eddy making due to flow separation. Also, the coefficient $k_{p|p|}$ is denoted a nonlinear damping coefficient. the $k_{p|p|}$ is about 60% of k_p at an advance speed of zero knots, $k_{p|p|}$ is reduced to 5% of k_p at an advance speed of 15 knots, and $k_{p|p|}$ is practically zero at an advance speed of 30 knots. The $f_2(\phi)$ is the restoring moment term due to gravity and buoyancy and it can be specified as Eq. (5).

$$f_2(\phi) = \Delta GZ(\phi) \tag{5}$$

where Δ is the ship's displacement and $GZ(\phi)$ is the restoring moment arm that it is function of the roll angle.

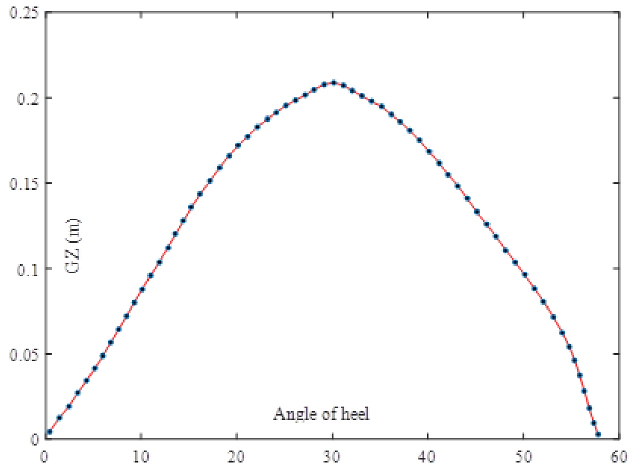

Figure 3. GZ curve of fishing vessel

The GZ curve is an odd function and therefore represent with odd order polynomial. Meanwhile, GZ curve of the fishing vessel is shown in Figure 3. The GZ-curve for the restoring moment arm has been defined by the following polynomial formula

$$GZ(\phi) = c_1\phi + c_3\phi^3 + c_5\phi^5 \tag{6}$$

where the coefficients c_1, c_3, and c_5 defined by Taylan [4]:

$$\begin{aligned} c_1 &= \frac{d(GZ)}{d\phi} = GM \\ c_3 &= \frac{4}{\phi_v^4}(3A_{\phi v} - GM\,\phi_v^2) \\ c_5 &= -\frac{3}{\phi_v^6}(4A_{\phi v} - GM\,\phi_v^2) \end{aligned} \tag{7}$$

where GM, ϕ, and $A_{\phi v}$ are, respectively, the metacentric height, angle of vanishing stability, and undersurface the GZ curve.

2.3. Calculation of Hydrodynamic Coefficients

The non-dimensional damping coefficient can be calculated as follows:

$$\zeta_\phi \cong \frac{\ln(\phi_1/\phi_2)}{2\pi}. \tag{8}$$

Based on the results above and the following standard relationships, the hydrodynamic coefficients may be calculated by the following formulas

$$\omega_\phi = \frac{2\pi}{T_\phi} \tag{9}$$

$$I_{xx} + K_{\dot p} = \frac{\Delta GM}{\omega_\phi^2} \tag{10}$$

$$k_p = 2\zeta_\phi\sqrt{\Delta GM(I_{xx} + K_{\dot p})} \tag{11}$$

The wave exciting moment is defined by

$$\tau_d = I_{xx}\omega_e^2\alpha_{\max}\cos(\omega_e t) \tag{12}$$

where α_{max} the maximum wave slope and ω_e is the wave encounter frequency that is described as

$$\omega_e = \omega\,(1 - \frac{\omega}{g}V\cos\mu) \tag{13}$$

where ω and μ are the wave frequency and the wave encounter angle.

2.4. Fin Dynamic Model

Block diagram of the fin-roll closed loop control system is summarized in Figure 4. The roll angle and roll rate are measured by gyroscope. The active fin is actuated by Electro-Hydraulic Servomechanism which is called fin's actuator. Fin's actuator has a first order dynamic model as Eq. (14).

$$T_e\dot\alpha_m + \alpha_m = K_{dc}\alpha_c \tag{14}$$

where, K_{dc} is dc gain of the actuator and T_e is time constant due to the delay between α_c and α_m.

The relationship between mechanical and effective angle of attack is as Eq. (15).

$$\begin{aligned} \alpha_e &= -\alpha_f - \alpha_m \\ \alpha_f &= \arctan\left(\frac{R_f\,p}{V}\right) \approx \frac{R_f\,p}{V} \end{aligned} \tag{15}$$

where α_f is flow angle which derived by the combination of the local roll-induced velocity together with the forward velocity of the fishing vessel.

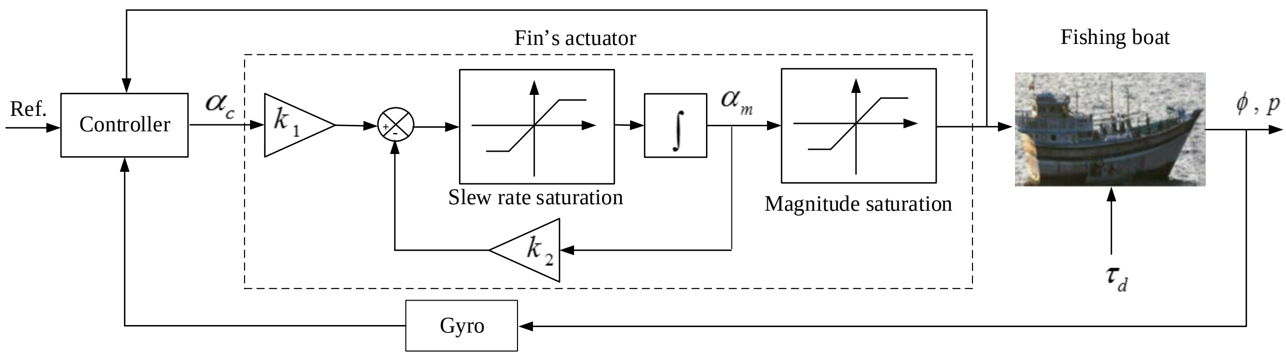

Figure 4. Block diagram of the fin-roll closed loop control system

2.5. State Space Model

According to Eqs. (1) and (15), the nonlinear model of roll motion can be summarized as following equation:

$$\begin{aligned}
&\left(I_{xx}+K_p\right)\ddot{\phi}+k_p\dot{\phi}+K_{p|p|}\dot{\phi}\left|\dot{\phi}\right| \\
&+\Delta\left(c_1\phi+c_3\phi^3+c_5\phi^5\right)=\tau_d-K_\alpha\alpha_e \\
&\alpha_e=-\frac{R_f\,p}{V}-\alpha_m \\
&T_e\,\dot{\alpha}_m+\alpha_m=K_{dc}\alpha_c.
\end{aligned} \tag{16}$$

Also, based on the assumptions listed in section 2.3, it is considered linear model roll motion. A linear state space model to describe the dynamic of the roll motion can be presented by

$$\begin{aligned}
&\dot{x}_1(t)=x_2(t) \\
&\dot{x}_2(t)=\frac{-\Delta GM}{I_{xx}+K_p}x_1(t)+\frac{\left(k_\alpha R_f/V\right)-k_p}{I_{xx}+K_p}x_2(t) \\
&\quad+\frac{k_\alpha}{I_{xx}+K_p}u(t)+\frac{1}{I_{xx}+K_p}\tau_d(t) \\
&\dot{x}_3(t)=k_2x_3(t)+k_1u(t)
\end{aligned} \tag{17}$$

where $x(t)=[x_1(t),x_2(t),x_3(t)]^T$ is $[\phi(t),p(t),\alpha_m(t)]^T$ and $u(t)$ is $\alpha_c(t)$. As well as $k_1=k_{dk}/T_e$ and $k_2=1/T_e$. Also, the operational constraints should be considered as:

• Input constraints which reflect the saturation of the mechanical fin angle:

$$\left|\alpha_c\right|\le\alpha_{sat} \tag{18}$$

• State constraint that is used in order to preventing the dynamic stall:

$$\left|\alpha_e\right|=\left|\frac{R_f}{V}P+\alpha_m\right|\le\alpha_{stall}. \tag{19}$$

3. Controller Design

In this section, a constrained LQR as an optimal controller for stabilizing the ship in roll motion was designed. The aim of controller is regulating the state variables with minimum control action. Therefore, it is considered the continuous-time linear time-invariant system as:

$$\dot{x}(t)=A_cx(t)+B_cu(t). \tag{20}$$

And it is sampled discrete-time version as Eq. (21).

$$x(t+1)=Ax(t)+Bu(t) \tag{21}$$

where $x\,\varepsilon\,R^3$ and $u\,\varepsilon\,R$.

The optimal control input, $u(t)$, should minimize the cost function, J.

$$\begin{aligned}
&J\left(u(t),u(t+1),\ldots,x(t)\right) \\
&=\sum_{\tau=t}^{\infty}x(\tau)^{\tau}Qx(\tau)+u(\tau)^{\tau}Ru(\tau)
\end{aligned} \tag{22}$$

Also, it is subjected to the linear constraints as Eq. (23).

$$\begin{aligned}
&G\,x(\tau+1)\le g \\
&H\,u(\tau)\le h.
\end{aligned} \tag{23}$$

For all $\tau\ge t$, where $R>0$, $Q\ge0$ and $G\,\varepsilon\,R^{1\times3}$ and $H\,\varepsilon\,R$. The pair (A, B) is controllable, and it is assumed that $g,h>0$ to ensure that the origin is an interior point in the acceptable region. The optimal cost function is defined as

$$F(x(t))=\min_{u(t),u(t+1),\ldots}J(u(t),u(t+1),\ldots,x(t)) \tag{24}$$

where the minimization is subject to the dynamics of the system (Eq. 21), and the constraints (Eq. 23) are imposed at every time $\tau\ \varepsilon\ \{t,\ t+1,\ t+2,\ldots\}$ on the trajectory. There are different methods for solving these kinds of problems. It has been done the constrained LQR design using Sequential Quadratic Programming (SQP) methodology with quadratic objective functions that is an effective algorithm.

4. Simulation Results and Discussion

The sailing condition is assumed for a random sea and beam sea condition at the forward speed 20 knots. Such a profile corresponding to a regular wave with period of 10s with a steepness of 1/60 is assumed for the calculation of exciting moment. The exciting moment was derived from a MATLAB code, and is depicted in Figure 5.

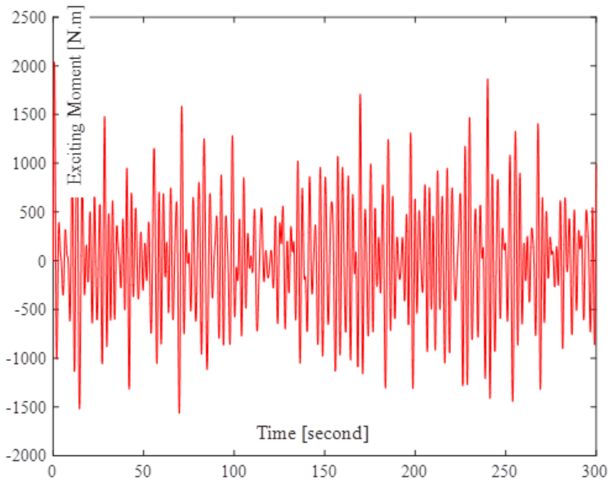

Figure 5. Plot of exciting moment

Table 3. The calculated coefficients of the vessel

Item	Value
$c_1\,(kg\,m^2\,s^{-2})$	0.73
$c_3\,(kg\,m^2\,s^{-2})$	-0.0504
$c_5\,(kg\,m^2\,s^{-2})$	0.1912
$k_p\,(kg\,m^2\,s^{-1})$	8.75
$I_{xx}+k_{\dot{p}}\,(kg\,m^2)$	53.65
$k_{p\|p\|}\,(kg\,m^2)$	0.4375

According to the empirical formulas and the specifications of the fishing vessel, with fin and in the linear region of the lift curve, obtained values are $C_L = 0.61$, $dC_L/d\alpha_e = 1.762\ (rad^{-1})$ and $\alpha_{stall} = 0.62\ (rad)$. Also, the fins moment is $\tau_c = 2047.83\ (N.m)$. By using of the roll decay test, the first two peaks are $\phi_1 = 8^{\circ}$ and $\phi_2 = 5.7^{\circ}$. The non- dimensional damping coefficient and roll natural period obtain $\zeta_{\phi} = 0.054$ and $T_{\phi} = 5.5$ sec. by using Eq. (8). The calculated restoring and damping moment coefficients for the fishing vessel are expressed in Table 3. Some other data are given as follows: Sea water density: $\rho = 1025\ (kg/m^3)$, Gravity constant: $g = 9.81 (m/s^2)$, $k_{p|p|}$: 5% of k_p, Added mass moment of inertia: 20% of displacement.

The open loop roll system is simulated using MSS toolbox. The roll responses for two state linear and nonlinear models for initial roll angle 5° are shown in Figure 6 and Figure 7. The results show the nonlinear terms cause deviation of linear model but the amplitude of these deviations, in order to design controller, is negligible.

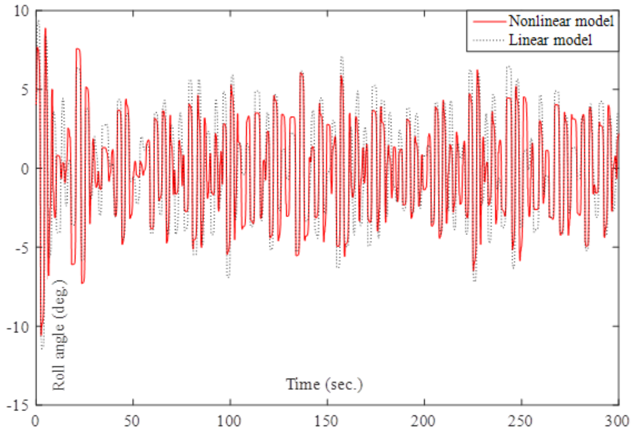

Figure 6. Comparison of open loop response for linear and nonlinear models with initial roll angle 5°

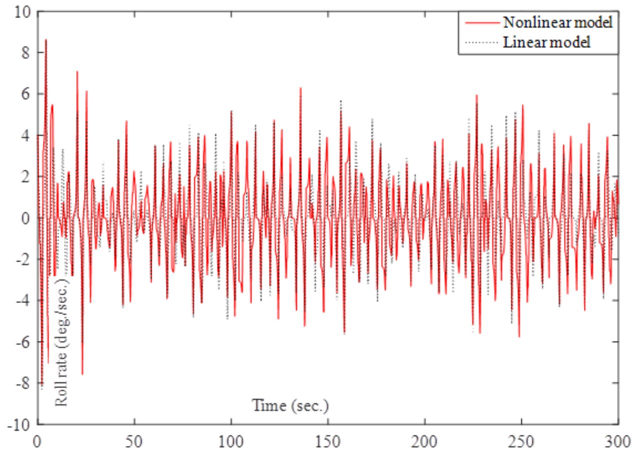

Figure 7. Comparison of open loop response for linear and nonlinear models

The closed loop fin-roll system is simulated by employing the constrained LQR controller with fin for initial roll angle 5°. Simulation results were compared with the results of the PID controller presented by [14] and the uncontrolled system. Figure 8 and Figure 9 show that the amplitude of the roll angle and roll rate in the constrained LQR controller is smaller and smoother than PID controller.

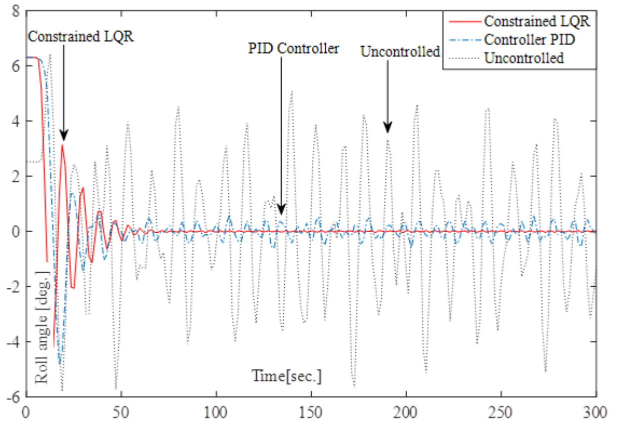

Figure 8. Comparison of roll angle response for the constrained LQR, PID controller and uncontrolled with initial roll angle 5°

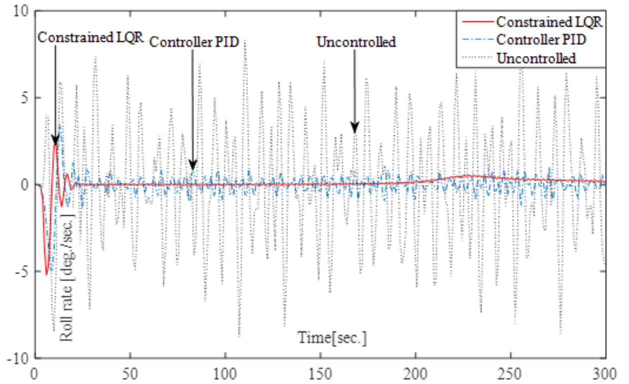

Figure 9. Comparison of roll rate response for the constrained LQR, PID controller, and uncontrolled

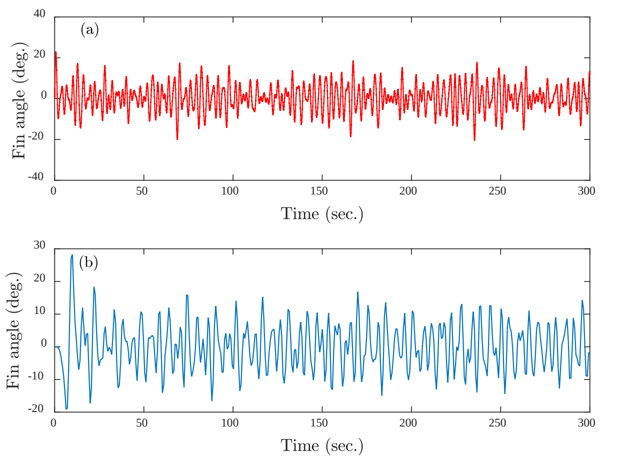

Figure 10. Variations of the fin regarding to the two controllers, a) Constrained LQR, b) PID controller

The mechanical angles of the fin in the simulations for two LQR and PID controller have been shown in Figure 10. The characteristics of simulation results for two controllers are summarized in table (4) which shows roll angle and roll rate, by using constrained LQR control are improved about 50% in comparison with the results of PID controller.

Table 4. Comparison of the controller's performance

Controller	Max. roll angle (deg.)	Max. roll rate (deg./s)
Uncontrolled	6.5	8
PID Controller	2	2.5
Constrained LQR	1	0.7

5. Conclusion

In this paper, the nonlinear modeling was derived for the fin-roll motion. The coefficients of nonlinear model, the nonlinear damping, and restoring terms were computed by means of the empirical formulas and numerical calculations by using the CFD method and MATLAB software. By considering the NACA0015 section, flow analysis and lift coefficient computation are presented by CFD method. Also, a constrained LQR was designed to satisfy the operational constraints of the roll motion including the mechanical fin angle and the dynamic stall saturations in order to achieve the desired performance. The simulation results for two constrained LQR and PID controllers were presented and compared in the presence of irregular wave. All in all, the simulation results showed that the constrained LQR reduced the RMS value of the roll motion.

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