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Fin based active control for ship roll motion stabilization

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Abstract. Ship roll motion control is important for vessels engaged in oceanographic research activities and this paper focuses on the design of a controller for fin based roll motion stabilization of a Coastal Research Vessel (CRV). Based on the geometry of a pair of actuator fins installed at the midship of the vessel, the hydrodynamic coefficients are calculated for the vessel including the fin lift capacity. The wave disturbances are simulated as a sine time series. The objective is to design a Linear Quadratic Regulator (LQR), a state feedback controller and obtain the performance of the system. The larger objective is to implement the system eventually in laboratory scale physical simulations in wave environment. This paper primarily presents the design of the control system and evaluation through Simulink in Matlab environment. The global cost function of the system is minimized by precision tuning of the two control parameters (or weighting matrices), Q and R. The system analysis is done using frequency domain and state space approach. The simulation results show that the natural frequency and roll response closely match with the response of the physical model (CRV) in laboratory environment, as observed during the experimental study. The proposed control system is compared with a conventional PID controller. The simulation results demonstrate the effectiveness of the designed roll motion stabilization system with significant roll reduction over the operational range of the vessel.

1 Introduction

Ship roll motion stabilization is of crucial importance and is a challenging problem due to performance limitations in the real environment. So, the roll motion stabilization has become a key research focus. There are passive devices like anti-roll tank, bilge keel and active devices like fin stabilizers which are commonly used stabilization devices. The active fins are highly effective roll motion stabilizers at the design speed.

Fins generate a moment by the principle of hydrodynamic lift, to stabilize the roll motions on the ship induced by the hydrodynamic interaction between the waves and the vessel. Fins are actuated by a control system. This paper focuses on the design of a controller for fin based roll motion stabilization. A mathematical model of the ship dynamics is formulated and hydrodynamic coefficients are calculated from the ship

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hydrostatics and experiments performed on the model scale. The study is extended to include the effect of fin stabilizer and to construct an optimal controller. The LQR controller is investigated through numerical simulations in MATLAB and Simulink. A PID controller is designed and investigated for comparison with the LQR controller.

Bhattacharyya (1978) explained the fundamentals of statics and dynamics of marine vehicles which are referred to model the ship system mathematically with the help of formulae presented. Dallinga (1993) presented hydromechanical aspects of the design of fin stabilizers and explored ways to improve the performance of fin stabilizers. Kawazoe et al. (1994) investigated the influence of the fin area and control methods on the roll reduction and presented a comparison between a PID control and fuzzy logic control. Perez and Blanke (2011) explained the technical feasibility of the roll motion control devices and also explained the challenges associated with the design and development of various ship motion control systems. Lee et al. (2011) presented a design of frequency weighted LQR controller with fin stabilizers and pod propellers to reduce the roll motion.

2 Ship dynamics

The candidate vessel used for the study is a 43m Coastal Research Vessel (CRV) as shown in figure 1. The experiments and simulations are carried out on a 1:17 scaled model of CRV (refer table 1). The model operates in an autonomous self-propelled mode [1]. In an early stage phase of controller design, the basic ship characteristics are estimated.

Table 1. Principal particulars of the vessel.

Particulars	Prototype (43m)		Model (1:17)	
Length overall	43	m	2.529	m
Beam	9.6	m	0.565	m
Draft	2.5	m	0.147	m
Depth	3.7	m	0.217	m
Displacement	615950	kg	121.95	kg
Design speed	12	knots	1.497	m/s

Fig. 1. Ship model in wave basin.

2.1 Hydrodynamic coefficients

The uncoupled equation for roll motion in one degree of freedom motion is adequately described by a simple linear equation as,

$$I\ddot{\phi} + b\dot{\phi} + c\phi = M_w \tag{1}$$

I is the mass moment of inertia, $\ddot{\phi}$ is the roll acceleration, b is damping coefficient, $\dot{\phi}$ is roll velocity, c is restoring force coefficient, M_w is wave exciting moment.

The mass moment of inertia is given as a sum of the moment of inertia of the actual mass and the virtual mass. The actual mass moment of inertia is calculated from the mass distribution data for the vessel where m is the mass of the weights distributed in the vessel and r is the radius of gyration,

$$I_{xx} = mr^2 \tag{2}$$

The virtual mass moment of inertia is taken as 10 or 20% of I_{xx} [2].

$$\partial I_{xx} = 0.2 I_{xx} \tag{3}$$

$$\therefore I = I_{xx} + \partial I_{xx} \tag{4}$$

The damping coefficient is estimated by means of roll decay test performed on the physical model in a wave basin facility (30m x 30m x 3m depth). In roll decay test, the vessel is displaced from its equilibrium position and allowed to oscillate freely. Due to the damping characteristics, the amplitude of the oscillations decreases and the vessel tries to gain its equilibrium position. The natural decay of the roll motion is recorded with fins set to zero deflection

The damping coefficient is given by,

$$b = \frac{v_\Phi \Delta GM_T T_\phi^2}{2\Pi^2} \tag{5}$$

where, v_Φ is decay constant, Δ is displacement, GM_T is the geometric metacenter, T_ϕ is natural roll period.

$$v_\Phi = \frac{1}{T_\Phi} \ln\left(\frac{\Phi_0}{\Phi_n}\right) \tag{6}$$

The restoring force coefficient, c is calculated as,

$$c = \Delta g GM_T \tag{7}$$

The wave exciting moment is calculated by integrating the additional buoyancy due to waves along the length of the ship [2].

$$M_w = 2\rho g\left(volume\right) \times \frac{2}{3} y \tag{8}$$

$$M_w = [\frac{2}{3}\rho g\left(k\eta_a\right) sin\mu \int_{-L/2}^{L/2} \cos(kxcos\mu) y^3] sinw_e t \tag{9}$$

2.2 Fin design

The fin designed for Coastal Research Vessel has NACA 0015 geometrically symmetric section with aspect ratio (ratio of span to chord) one and located at the midship [3] [4]. The optimum fin area is calculated based on the wave slope capacity of the fin. Wave slope capacity is the hull heel angle caused by the maximum lift force of the fin in calm sea. [5] [6]

- Wave slope capacity, θ_{wsc} -

$$\theta_{wsc} = \sin^{-1}\frac{(F \times L)}{\Delta GM_T} \tag{10}$$

where F is fin lift force and L is the moment arm lever.

- Fin area based on wave slope capacity A_f –

$$A_f = \frac{\Delta(GM_T)\sin\theta_{wsc}}{\rho C_L V^2 L} \tag{11}$$

where C_L is the coefficient of lift and V is ship speed.

Table 2 gives the fin particulars and figure 2 shows the cross section and planform of the fin. Figure 3 depicts the computer aided drawing model of the fin fitted to the hull. The fin is fabricated and fitted to the 1:17 model of CRV as shown in figure 4.

Table 2. Fin particulars.

Particulars	For prototype (43m)
Number of fins	2
Planform area	4m^2
Profile	NACA 0015
Aspect Ratio	1

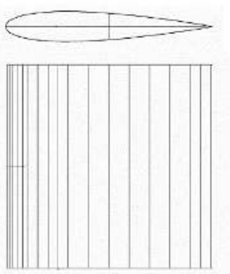

Fig. 2. Fin section and planform.

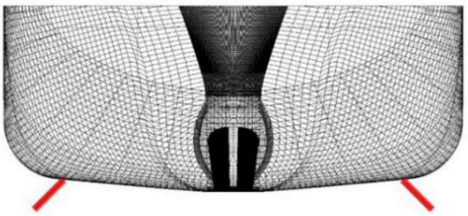

Fig. 3. Fin fitted to the hull.

Fig. 4. Fin fitted to the fabricated model of CRV.

3 Control system design

The feedback controller is designed to generate a fin moment to oppose the exciting moment [7][8]. The controller is designed with fin rate limited to ±25deg/s.

The general layout of the control system is shown in the figure 5. The ship is subjected to the wave disturbances which induces roll motion in the ship. This motion is sensed by the Motion Reference Unit (MRU) fitted on the ship. The feedback from the MRU is taken by the controller and an error signal is generated with reference to the desired roll angle. The controller gives the control signals to the stepper motor, which acts as an actuator. The stepper motor actuates the fins to stabilize the roll.

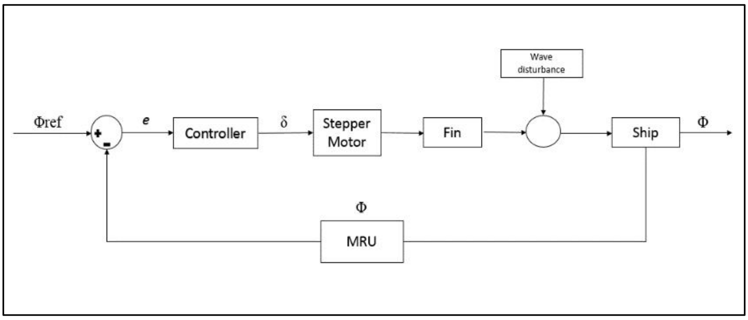

Fig. 5. General layout of the control system.

The equation of roll motion for the vessel with fins is given by,

$$I\ddot{\phi} + b\dot{\phi} + c\phi = M_w + M_f \tag{12}$$

where M_f is the fin moment and is expressed as,

$$M_f = M_{fo}\alpha \tag{13}$$

$$M_{fo} = \frac{1}{2}\rho V^2 SL\frac{dC_L}{d\alpha} \tag{14}$$

α is the fin reaction and S is the planform area of fin.

3.1 LQR Controller

LQR operates by estimating future outputs based on the past outputs to minimize the global cost function of the system and thus giving a better regulation. LQR is well preferred when we linearize a non-linear system as it matches closely to the real-time system.

In the concept of state-space, the state-space representation of a system allows us to represent any nth order differential equation in the form of single first order matrix differential equation. [9][10][11]

The equation is represented in the form of –

$$\dot{x} = Ax + Bu \tag{15}$$

$$y = Cx \tag{16}$$

where A is state matrix, B is input matrix, C is output matrix, x is state vector and u is input/control vector.

The 1-DOF roll motion model is written in the form of state-space,

$$I\ddot{\phi} + b\dot{\phi} + c\phi = M_w + M_{fo}\alpha \tag{17}$$

$$\ddot{\phi} = \frac{M_w}{I} + \frac{M_{fo}\alpha}{I} - \frac{b}{I}\dot{\phi} - \frac{c}{I}\phi \tag{18}$$

Let,

$$x_1 = \phi \tag{19}$$

$$x_2 = \dot{\phi} \tag{20}$$

The state-space representation of the equation is,

$$\begin{pmatrix} \dot{x}_1 \\ \dot{x}_2 \end{pmatrix} = \begin{pmatrix} 0 & 1 \\ -\frac{c}{I} & -\frac{b}{I} \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \end{pmatrix} + \begin{pmatrix} 0 \\ \frac{1}{M_w} \end{pmatrix} + \begin{pmatrix} 0 \\ \frac{1}{M_{fo}\alpha} \end{pmatrix} \tag{21}$$

$$y = \begin{pmatrix} 1 & 0 \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \end{pmatrix} \tag{22}$$

In order to implement the state space control design, it is necessary to verify the important properties of the system – Stability, Controllability and Observability. The system is controllable if the controllability matrix Co has full rank.

$$Co = \left[B\ AB\right] \tag{23}$$

The observability matrix is given by,

$$Ob = \begin{pmatrix} C \\ CA \end{pmatrix} \tag{24}$$

The objective of the Linear Quadratic method is to obtain a gain matrix K to minimize the performance index (or cost function) J [10]. The cost function is given by the equation,

$$J = \int_0^\infty \left(x^T Q x + u^T R u \right) dt \tag{25}$$

The LQR controller calculates the feedback gain matrix K, to minimize the above cost function, in MATLAB.

3.2 PID Controller

The transfer function of the system is given by,

$$\frac{\phi(s)}{\alpha(s)} = \frac{k w_n{}^2}{s^2 + 2\xi w_n + w_n{}^2} \tag{26}$$

where, *k* is the gain value

Natural frequency,

$$w_n = 3.112 rad/s \tag{27}$$

Damping ratio,

$$\xi = 0.05 \tag{28}$$

The fin reaction is given by,

$$\alpha = K_p \phi + K_D \dot{\phi} + K_I \ddot{\phi} \tag{29}$$

where K_P, K_I and K_D are proportional, integral and derivative gains respectively.

The PID controller has three control parameters which need to be well tuned based on the system requirements. The advantage of the PID controller is the feasibility and ease of application. However, it is difficult to balance all the three gains (K_P, K_I, K_D) and obtain precise values. In such cases, the controller compromises with the transient response of the system, that is, the settling time, the overshoot and the steady state error.

As the control gain values are not precisely estimated, the controller may not resist the disturbances and uncertainties. This leads to a control system with low robustness. Even though the gains are precisely estimated, the PID controller has low robustness as compared to the modern robust controllers like LQR controller.

4 Simulation results

The simulations are carried out in MATLAB and Simulink for PID and LQR controller. Based on the sea-keeping tests performed on model in regular waves, the peak response of roll occurs at a frequency of 3.2057rad/s, which is the resonance frequency.

The figure 6 and figure 7 show the comparison of the roll response obtained though laboratory sca

le physical simulations in wave environment with the roll response simulated in Simulink. It is observed that the experimental and simulation results are in good agreement.

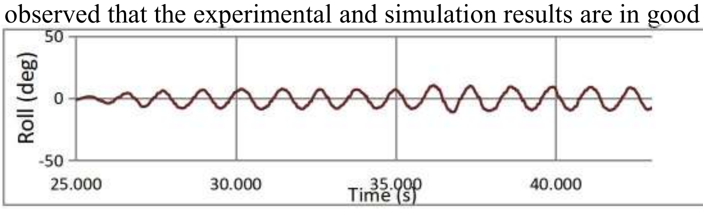

Fig. 6. Roll response simulated in wave basin.

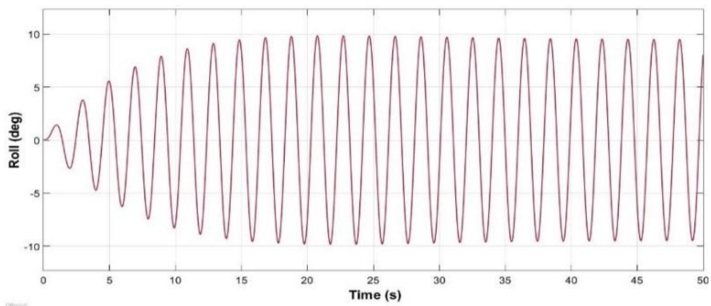

Fig. 7. Roll response simulated in Simulink.

The designed controllers are tested for various speeds – 6 knots, 8 knots, 10 knots and 12 knots. The controller performance is verified by plotting the simulation results shown in the figure 8-15.

4.1 LQR Controller simulations

Figure 8-11 show the simulations results with LQR control algorithm applied.

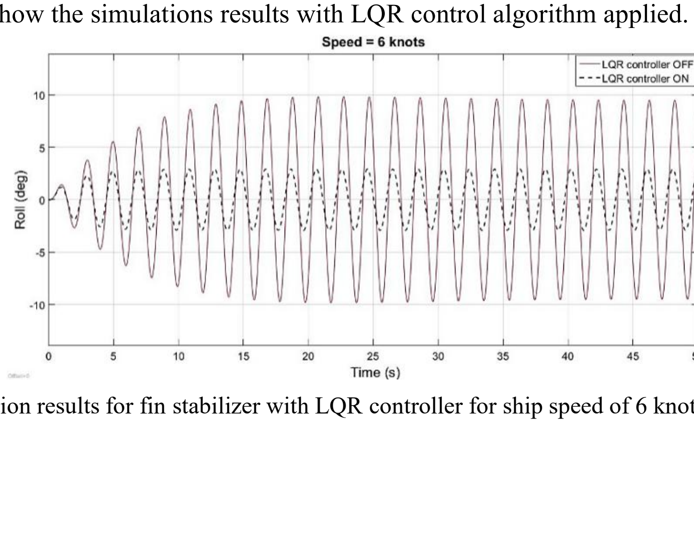

Fig. 8. Simulation results for fin stabilizer with LQR controller for ship speed of 6 knots.

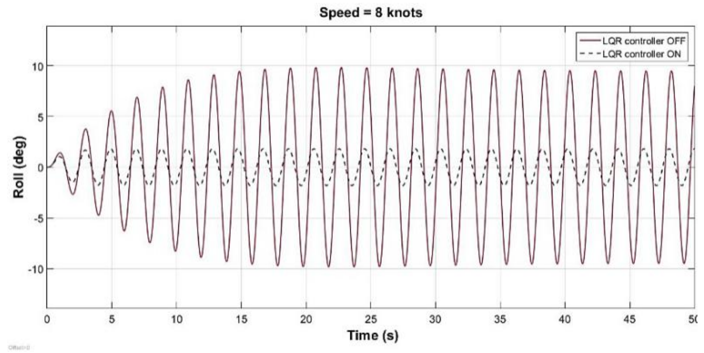

Fig. 9. Simulation results for fin stabilizer with LQR controller for ship speed of 8 knots.

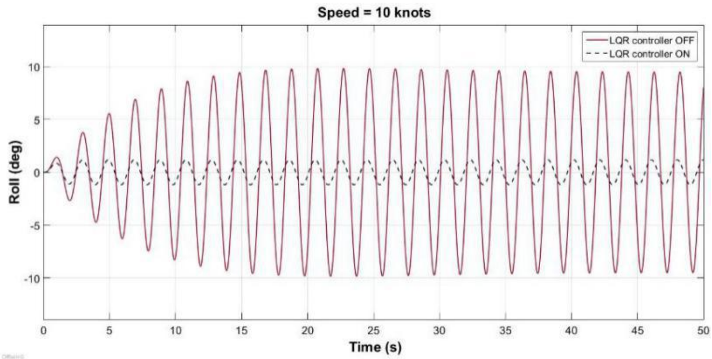

Fig. 10. Simulation results for fin stabilizer with LQR controller for ship speed of 10 knots.

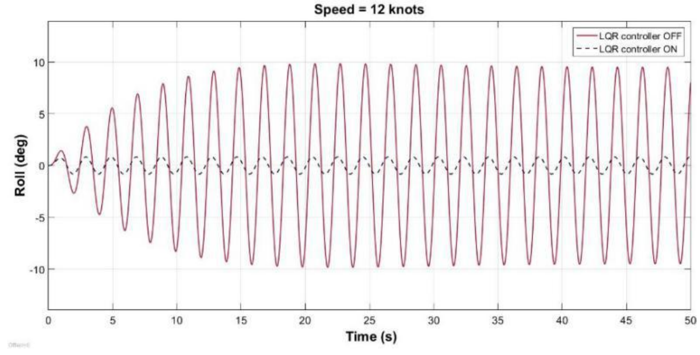

Fig. 11. Simulation results for fin stabilizer with LQR controller for ship speed of 12 knots.

4.2 PID Control Simulations

The simulation results are compared for different conditions as presented in table 3. It is observed that, as the speed increases from 6 to 12 knots the value reduction in roll amplitude increases in case of both the controllers. At low speeds the fin lift force reduces drastically and the effectiveness is reduced. The fins operate at maximum efficiency at higher speeds. The fin rate is limited to ±25deg/s.

Figure 12-15 show the simulation results with PID control algorithm.

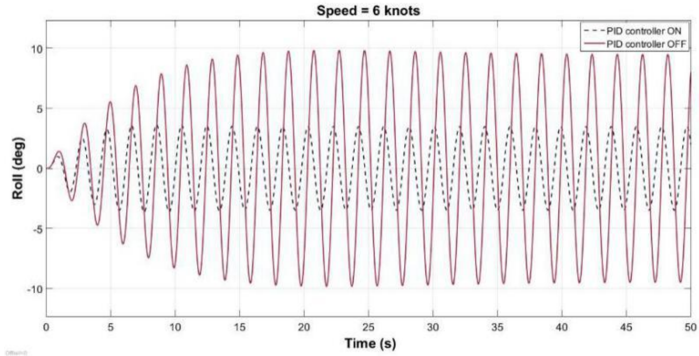

Fig. 12. Simulation results for fin stabilizer with PID controller for ship speed of 6 knots.

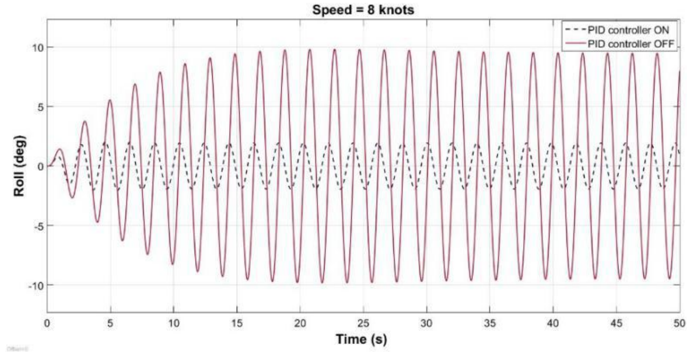

Fig. 13. Simulation results for fin stabilizer with PID controller for ship speed of 8 knots.

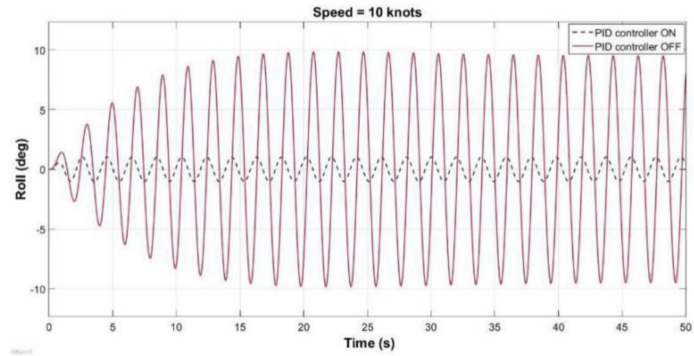

Fig. 14. Simulation results for fin stabilizer with PID controller for ship speed of 10 knots.

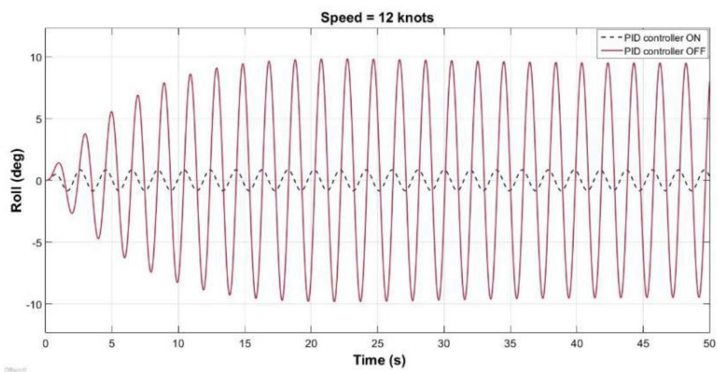

Fig. 15. Simulation results for fin stabilizer with PID controller for ship speed of 12 knots.

Table 3. Simulation results comparison chart.

Speed (knots)	Roll Amplitude (deg)		Percentage reduction in roll amplitude	
	LQR Controller	PID Controller	LQR Controller	PID Controller
0	9.714	9.714		
6	3.030	3.551	68.80%	63.44%
8	1.837	1.938	81.08%	80.05%
10	1.141	1.200	88.25%	87.64%
12	0.825	0.925	91.50%	90.47%

5 Conclusion

The paper represents the design of an optimal controller for fin based ship roll motion stabilization. The candidate vessel is verified for stability and controllability. The damping characteristics and the natural frequency are estimated from the experiments performed on the model scale. The simulation results verify the effectiveness of the proposed design.

In case of LQR controller, when the vessel operates at a speed of 12 knots (high speed), the percentage reduction in roll amplitude is obtained as 91.50%. When the vessel operates at a speed of 6 knots (low speed), the percentage reduction in roll amplitude is 68.80%.

In case of PID controller, when the vessel operates at a speed of 12 knots (high speed), the percentage reduction in roll amplitude is obtained as 90.47%. When the vessel operates at a speed of 6 knots (low speed), the percentage reduction in roll amplitude is 63.44%.

The feasibility of the implementation of the proposed design to the laboratory scale physical simulations is investigated.

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