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Design of an Optimal Controller for the Roll Stabilization of Surface Ships with Active Fins

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Abstract

In this paper, an optimal controller is designed to control the undesired roll motion of a ship under the effect of sea waves by using active fin stabilizers. The roll dynamics is described by a single-degree-of-freedom nonlinear model. An actuator dynamics is also included to the dynamic system. Sinusoidal and random wave models are used to describe the wave elevation that causes disturbance moments in the ship. A worst-case scenario is the application of the periodic wave to bring the ship resonance, whereas the random waves are used to test the system at the smooth and moderate sea states. In designing the controller, the energy optimal control method, which allows both the closed-loop and real-time control of dynamic systems, is employed, and the control law is obtained analytically. The performance of the controller, under the effect of environmental disturbances, is tested by computer simulations and the results are compared with those from LQR controlled ship.

Keywords: *Ship roll motion, Fin stabilizer, Optimal control, Roll damping.*

Gemilerin Aktif Kanatla Yalpa Stabilizasyonu için Bir Optimal Kontrolcü Tasarımı

Öz

Bu çalışmada, dalga etkisi nedeniyle istenmeyen yalpa hareketi yapan bir geminin aktif kanat dengeleme sistemi vasıtasıyla kontrolü için bir optimal kontrolcü tasarımı yapılmıştır. Tek serbestlik derecesine sahip doğrusal olmayan bir model kullanılarak yalpa dinamiği tanımlanmıştır. Ayrıca, kanatlara ait aktüatör modeli de sisteme eklenmiştir. Gemiye bozucu etki yapan deniz dalgalarının modellenmesinde, dalga yüksekliğinin sinüzoidal bir fonksiyon ve rastgele dalga modeli kullanılmasıyla iki yaklaşım benimsenmiştir. Periyodik dalga ile geminin doğal frekansında rezonansa getirilmesiyle olabilecek en kötü durum test edilmeye çalışılırken, küçük ve orta dalgalı deniz durumlarına karşılık gelen iki ayrı rastgele dalga modeli ile gerçekte karşılaşılabilecek durumlar test edilmeye çalışılmıştır. Kontrolcü tasarımında, dinamik sistemlerin gerçek zamanlı ve kapalı çevrim kontrolüne imkan veren enerji optimal kontrol metodu kullanılmıştır. Analitik olarak elde edilen kontrol kuralı vasıtasıyla, bahsedilen bozucu etkiler altında, kontrol performansı bilgisayar simülasyonları ile test edilerek istenmeyen yalpa hareketinin azaltıldığı gösterilmiştir ve bir LQR kontrolcü ile kontrol edilmiş geminin yalpa hareketleriyle karşılaştırılmıştır.

Anahtar Kelimeler: *Gemi Yalpa Hareketi, Kanat Dengeleyici, Optimal Kontrol, Yalpa Sönümleme.*

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1. Introduction

The ship roll motion caused by wave disturbances might affect the passengers, crews, equipment and cargos adversely. In reducing the undesired roll motion of ships, hydraulically actuated fin stabilizers are widely used. Compared to other roll stabilization techniques, which are rudder roll stabilization, bilge keels, gyro-stabilizers, and anti-rolling tanks, active fin stabilizers have higher performance [1]-[3]. Another advantage is that they do not need sophisticated control systems. Therefore, the ship roll stabilization through active fin stabilizers is a widely studied approach.

The challenges in the control of ship roll motion have attracted the attention of researchers. For the roll stabilization of a ship through active fins, the design of a classical controller and an adaptive linear quadratic compensator are reported in [4]. In the gain scheduling adaptive controller, which revealed a superior performance than the classical controller, the gains of the regulator are calculated by a multilayer perceptron neural network. For three different sea conditions, the reduction in the roll motion is exhibited. This is one of the few studies using an optimal control method in the field, since optimal control methods have not been widely applied in the control of ship roll motion. Karakas et al. designed a roll motion control system by using the Lyapunov's direct method [5]. The effectiveness of the controller under the effect of beam seas was shown in a simulation study. In [6], the designed proportional, derivative, second derivative controller was tuned by particle swarm optimization algorithms. In simulations and real-time full-scale sea trials, the control algorithm achieved to damp the roll motion significantly. Another method for the ship roll stabilization is proposed in [7], where the fin control design method is based on an adaptive neural-network. In this approach, the disturbance is estimated and compensated to improve the robustness. The simulation results show that the rolling motion reduced for a ship under the effect of a sinusoidal disturbance. In a recent study [8], the uncertainties in the ship and fin system are identified by a neural network and an adaptive robust fin controller was designed. Another study [9] employing an artificial intelligence technique in the roll stabilization reports the identification of a fishing boat for the roll dynamics and use of a fuzzy logic controller. In a comparative study, it was shown that the fuzzy logic controller handles the nonlinear effects and the time-varying parameters better than the PID controller does. In a recent study, Demirel and Alarçin have designed LMI-based H_2 and H_∞ state-feedback controllers for the roll reduction of a fishing boat. The results show that both controllers are effective in the roll stabilization and H_∞ controller's performance is better [10]. Another recent study discussing the roll reduction for a trawler type fishing boat has proposed the use of a backstepping controller. The results indicate that the roll stabilization by the backstepping controller is highly satisfactory [11].

One of the difficulties in the ship roll motion control is the transport delay due to the hydraulic actuator system. In this direction, a ship roll stabilization system based on a variable structure robust control of fins proposed in [12]. By considering the active anti-rolling fin stabilizer as a mismatching uncertain system, a variable structure robust controller is designed. It is shown that the stability of the closed loop is not affected by the time constant of the actuator. Another difficulty arises from the unsteady hydrodynamic characteristics of the fins. Perez and Goodwin [13] proposed the use of a model predictive controller to prevent the effects of dynamic stall in fins. By imposing constraints on both the mechanical angle of the fins and the estimated effective angle of attack in the proposed control approach, the performance of the roll stabilization was improved.

In the literature of ship roll stabilization, the studies mostly do not consider the following factors all together: A realistic time constant of fin actuator, a variable lift coefficient, and a random wave disturbance with an effective amplitude. In this paper, all these factors are taken into consideration. In the roll motion stabilization of a ship with active fin stabilizers, only the roll dynamics related model of the ship is considered, since the other motions are irrelevant. This nonlinear model includes the dynamics of roll, actuator system, fins, and the environmental disturbances. As the roll dynamics, a widely used model is employed in this study. The actuator model represents the first-order dynamics of a hydraulic system, but the details related with hydraulic components are not included. The control force is the roll moment generated by a fin, which is described by the lift equation. In this equation, the lift coefficient is a time-varying parameter and taken as a linear function of the angle of attack. The environmental disturbance force is the sea surface elevation and its effect as a roll moment contributes to the roll equations of motion. Two different surface elevation models are used: i) A sinusoidal wave model, which is used to disturb the ship in its natural frequency, and ii) random wave models, which represent stochastic waves.

The control method used in this research was developed by Fukushima [14]. This method proposes the optimal control of mechanical systems by employing the energy balance of the system. The flexibility of determining the performance criterion enables the criteria function to be of any form. The criteria function to be minimized includes the energy equation and the control-performance, which are not necessarily to be in quadratic form as in the classical optimal control theory. After applying the necessary minimization condition, the control law is obtained. Then, the control can be performed by interconnecting the plant with the optimal control law in a closed-loop system. This method allows that the control law is obtained analytically and the system can be controlled in real-time [15].

Using this control method for the roll stabilization, an optimal control law has been obtained without solving the nonlinear differential equations. The terms coming from the chosen performance indices appear in the control law and play a major role. On the other hand, the terms coming from the equation of motion are only the damping terms and they contribute to the stability. By the application of this control law, behavior of the controlled ship is investigated for three case studies, where the ship exposes to the disturbances of a periodic wave and random sea waves for two different sea states.

To compare the results obtained by the application of the proposed controller with a classical controller, an LQR controller is designed and the same scenarios are tested by the application of this optimal controller. Even though both controllers are optimal, which is favorable for a fair comparison, the design of the latter controller requires a linearized model. Since it is more realistic to use the lift coefficient as a function of time, the system is modelled as a time-varying linear system.

The remainder of this paper is organized as follows: In Section 2, the mathematical models of the ship, fins and actuator are described. In Section 3, the applied control method is briefly introduced and the designs of this proposed controller and LQR controller are presented. The results of simulations are given in Section 4.

2. Mathematical Model of the System

2.1. Ship Roll Motion

In this section, the mathematical model of nonlinear roll motion is described. In practice, instead of full 6-DOF ship model, a 4-DOF or a 1-DOF model can be used. In this work, the 1-DOF nonlinear model is employed, since the aim is to stabilize only the roll motion. General roll motion equation for a ship, which is under the excitation of wave disturbance, is given by [16] and [17]:

$$(I_{xx} + \delta I_{xx})\ddot{\phi} + B(\dot{\phi}, \phi) + \nabla\, GZ(\phi) = T_e \tag{1}$$

where ϕ is the roll angle, I_{xx} is the mass moment of inertia, δI_{xx} is the added mass moment of inertia, ∇ is the displacement volume, GZ is the righting arm, and T_e is the environmental disturbance forces, which is described in Section 2.3.

If the fin roll moment is expressed as follows [18]:

$$T_F = 2\, K_\alpha\, \alpha_e \tag{2}$$

$$K_\alpha = \frac{1}{2}\rho V^2 A_F C_L L_F\, \alpha_e\, sgn(C_L) \tag{3}$$

$$\alpha_e = \alpha_m + \frac{\dot{\phi}}{V} L_F \tag{4}$$

and the damping and restoring forces are selected as discussed in [18], Equation (1) can be rewritten as follows:

$$I_d\, \ddot{\phi} + K_p\, \dot{\phi} + K_{p|p|}\, \dot{\phi}|\dot{\phi}| + K_\phi\, \phi = T_e + T_f \tag{5}$$

where;

$$I_d = I_{xx} + \delta I_{xx}$$

$$K_\phi = \rho\, g\, \nabla\, GM$$

$$K_\phi = I_d\, \omega_\phi^2.$$

In Equations (2)-(5), ρ is the density of fluid, V is the relative speed between fins and the flow, A_F is the surface area of the fins, C_L is the lift coefficient of the fins, L_F is the moment arm of fins, α_m is the mechanical angle of the fins (control input), GM is the metacentric height, K_p and $K_{p|p|}$ are the hydrodynamic coefficients, and ω_ϕ is the natural frequency of the motion. The values of these parameters used in the simulations are given in Table 1. For further details about the model, the reader can be referred to [18] and [19].

2.2. Control Force

The roll moment generated by any fin is given in Equation (2). The relative speed, V, can be assumed to be equal to the forward speed of the ship, U. In Equation (4), the addition of the terms on the right-hand side of the equality represents the effective angle of attack, α_e , between the fin and the fluid velocity. Note that the effect of random disturbance due to the waves on the angle of attack is neglected in this study.

In modelling the roll moment by fins, one of the parameters that raises difficulty in control is the lift coefficient. This parameter actually occurs as a time-varying parameter in the plant model. The variation of C_L with respect to α_e can be seen in the plot of steady-flow characteristic of the lift in Figure 1. This study takes into account only the steady-characteristic of the fins. At the stall angle, α_{stall} , a flow separation develops and the lift force starts to decrease. From this angle, the behavior of the C_L is nonlinear. In the design of controller, up to αs_{tall} the relation between C_L and α_e can be accepted as linear and be approximated as:

$$C_L = \left.\frac{\partial C_L}{\partial \alpha_e}\right|_{\alpha_e = 0}. \tag{6}$$

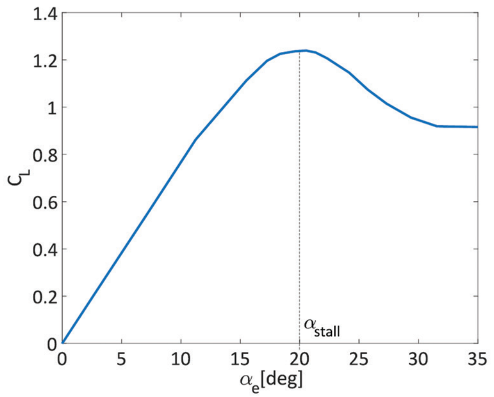

Figure 1. *Steady Free-stream Lift Characteristic of the Fin*

Hence, the following model is employed in the simulations:

$$C_L = \frac{C_{L_{max}}}{\alpha_{max}} \alpha_e \qquad (7)$$

where the values of parameters are given in Table 1.

In high sea states, large mechanical angles, in where nonlinear effects due to unsteady hydrodynamics of the fins arise, are demanded by the controller. To prevent the deteriorating effects of the fins, that causes dynamic stall, the fins can be operated in a range up to the stall angle. We assume in this study that the stall angle is fixed for all the forward speeds of the ship.

Another important element in the controlled system is the model of the actuator. The actuation of the fin stabilizers are generally provided by electro-hydraulic system. Use of such a system poses challenge in the roll stabilization due to the lagged response of the fins. Demanding less time delays requires a more powerful machinery, which affects the cost and volume of the actuation system [12]. It is obvious that the time delay cannot be ignored or be taken an arbitrarily small value. Thus, neglecting the effect of the actuator in the design of controller degrades the performance of the closed-loop system, so the stabilization may not be possible, and it may even cause instability.

Control of the fins are constrained by the characteristics of this hydraulic system. These constraints appear as the magnitude saturation, which is the maximum α_m, the slew rate saturation, which is the maximum rate of α_m, and the time delay, which is the delay between the commanded control input, α_c, and the actual mechanical angle of the fin [20]. The model of the actuation system can be simplified by the following first-order linear system [21]:

$$T_a \, \dot{\alpha}_m + \alpha_m = K_a \, \alpha_c \qquad (8)$$

where T_a is the time constant of the actuator and K_a is the gain of the control input.

2.3. Sea Wave Disturbance Model

In modeling the sea wave disturbance, two different models are employed: Periodic and random (irregular) wave models. The proposed controller can be tested by applying the periodic disturbance excitation in the natural frequency of the ship. Since ocean waves are random, the stochastic model serves for testing the roll stabilization realistically.

In the first model, we consider that the ship is under the excitation of regular sinusoidal waves with no phase lag [16]:

$$T_e = \omega_e^2 a_m I_{xx} \cos(\omega_e t) \qquad (9)$$

where ω_e is the frequency of encounter, and a_m is the maximum wave steepness.

To describe the waves in random seas mathematically, a stochastic modeling approach is used. In that approach, the random wave elevation can be written as [22]:

$$\eta(x,y,t) = \sum_{j=-s}^{s} \sum_{i=0}^{r} a_i \, \{ \, cos(\frac{\omega_i^2}{g} (\, x \, cos(\theta_i) + y \, sin(\theta_i)) \, - \omega_i \, t \, + \epsilon_{i,j})\} \qquad (10)$$

where x and y are the position coordinates, g is the acceleration of gravity, θ is the directional angle, and ϵ is the phase, which takes random values between π and $-\pi$. In Equation (10), a_i can be given as follows:

$$a_i = \sqrt{2\, S(\omega_i)\Delta\omega_i\, \Delta\theta_i f(\theta)} \tag{11}$$

where $\Delta\omega_i=\omega_i-\omega_{i-1}$ and $\Delta\theta_i=\theta_i-\theta_{i-1}$. The last term in Equation (11) describes the directional spreading of waves [23]:

$$f(\theta) = \begin{cases} \frac{2}{\pi}\cos^2(\theta) & \theta < \frac{\pi}{2} \\ 0 & otherwise \end{cases} \tag{12}$$

To describe the wave spectral density function in Equation (11),

$$S(\omega) = A\, \omega^{-5}\, e^{-B\, \omega^{-4}} \tag{13}$$

is used. Here, ω is the frequency of the waves. According to the recommendation of ITTC for the Modified Pierson-Moskowitz family, the parameters A and B can be taken as

$$A = \frac{173\, H_{1/3}^2}{T_1^4}\ ; \qquad B = \frac{691}{T_1^4} \tag{14}$$

where $H_{1/3}$ is the significant wave height and T_1 is the average wave period [18].

3. Controller Design

3.1. Energy Optimal Controller

The purpose of the control in this study is to generate the corrective roll moment through the controlled fins in order to stabilize the roll motion. Two symmetrically placed hydrofoils are used as the fin roll stabilizer. Hence, the objective is to find an optimal control law, α_c, so that the controlled fin would stabilize the ship.

In this paper, an energy optimal controller is designed to control the roll motion of the ship. The reader is referred to [15] for the explanation of the method and [24]-[27] for its applications. In this method, optimal control of mechanical systems is sought by employing the energy balance of the system. To compose the criteria function to be minimized, two main indices are required: The power equation of the system and the control performance function. As in the feedback control, the control can be done by interconnecting the plant with the optimal control law in a closed-loop system.

In the design of energy optimal controller, the first step is to determine an energy equality, which is formulated as the power equation, in the system and performance indices. Then, a scalar function is constituted to represent the criteria function. The scalar function to be minimized includes the main characteristics of the roll dynamics through the power equation. The equation describing the power equality is obtained by multiplying the both sides of the roll moment equation, (5), by the roll rate. Since we are interested in only the stabilization of the roll motion, instead of the total power equation of the whole system, the following power equality without the input power, which will appear as a performance index, is calculated:

$$\begin{aligned} P = &-\big(I_d\, \dot\phi\, \ddot\phi + K_p\, \dot\phi^2 \\ &+ K_{p|p|}\, \dot\phi^2\, |\dot\phi| + \ K_\phi\, \phi\, \dot\phi\big) \end{aligned} \tag{15}$$

Note that Equation (15) does not hold the wave disturbance. Hence, the first performance index, which includes P, can be described as

$$J_1 = R_1 \int P\, dt, \tag{16}$$

where R_1 is a weighting factor. The reason for excluding input power from P is to prevent reappearance of the input power delivered to actuators, which can be written as another performance index as follows:

$$J_2 = R_2 \int T_F\, \dot\phi\, dt, \tag{17}$$

where R_2 is a weighting factor. By considering the roll stabilization objective, the control performance to be minimized can be determined. One of the selected measures is the error function of angular position, which represents the deviation of the actual roll angle of the system from the desired one, ϕ_d:

$$J_3 = \int (\phi - \phi_d)^2 \, dt \, . \tag{18}$$

In the design of the proposed controller, the reduction of the roll acceleration is considered as a direct control objective, since it is important for the ship performance. This issue is stated in [24]: "Lateral accelerations caused by roll-reducing devices may be more harmful to human performance than some greater amount of roll". The performance index for the minimum roll acceleration can be defined as:

$$J_4 = R_3 \int \left(\ddot{\phi} - \ddot{\phi}_d\right)^2 dt \tag{19}$$

where $\ddot{\phi}_d$ is the desired roll acceleration.

Simply, the performance measure of the AUV is written as $J=J_1+J_2+J_3+J_4$. As described in [15], a scalar function L can be defined and it is written that $J=\int L \, dt$. Thus, L can be written as the time derivative of J:

$$L = R_1 P + R_2 \, T_F \, \dot{\phi} + (\phi - \phi_d)^2 + R_3\left(\ddot{\phi} - \ddot{\phi}_d\right)^2 . \tag{20}$$

The function L is minimized by applying the integrated Euler equation [15]:

$$\int \frac{\partial L}{\partial \phi} dt - \frac{\partial L}{\partial \dot{\phi}} + \frac{d}{dt}\left(\frac{\partial L}{\partial \ddot{\phi}}\right) = 0 \, . \tag{21}$$

Finally, the resulting control law for the stabilization of roll motion is obtained:

$$\alpha_c = (-U\phi^2 - R_3 \, U\ddot{\phi}^2 - 4R_2 L_F K_\alpha \dot{\phi} - 2R_1 K_p U \dot{\phi} - 3R_1 K_{p|p|} \, \dot{\phi}|\dot{\phi}|) \, / (\, 2R_2 \, K_\alpha \, U) \tag{22}$$

where ϕ_d and $\ddot{\phi}_d$ are set to zero. Note that, in Equation (22), since the roll rate term appears together with the roll angle, deviation of actual roll rate from a desired one, which is typically selected as zero, is not needed as a performance index. It is understood that the last term in the numerator comes from the damping term in Equation (5) and contributes to the stability of the controlled system.

3.2. Linear Quadratic Regulator

To show the effectiveness of the proposed controller in a comparative study, a classical controller is also designed. We select the state-feedback LQR considering that it is as an optimal controller suitable for a fair comparison. Since the LQR method is well known in the literature, derivation of the controller is not repeated here.

The nonlinear system can be linearized as follows:

$$I_d \, \ddot{\phi}(t) + K_p \, \dot{\phi}(t) + K_\phi \, \phi(t) = T_e(t) + T_f(t). \tag{23}$$

By representing the time-varying system in the state-space form

$$\dot{\boldsymbol{x}}(t) = A(t)\boldsymbol{x}(t) + B(t)\boldsymbol{u}(t) \tag{24}$$

where the state vector $\boldsymbol{x}(t)=[\phi(t) \; \dot{\phi}(t)]^T$ and the control vector $\boldsymbol{u}(t)=[\alpha_m]$; the system matrix, A(t), and the input matrix, B(t), can be defined as:

$$A(t) = \begin{bmatrix} 0 & 1 \\ -K_\phi & -K_p + \dfrac{\rho U A_F C_L(t) L_F^2}{I_d} \end{bmatrix} \tag{25}$$

$$B(t) = \begin{bmatrix} 0 \\ \dfrac{\rho U^2 A_F C_L(t) L_F}{I_d} \end{bmatrix}. \tag{26}$$

Note that, in (25) and (26), the lift coefficient is a time-varying parameter. If the quadratic cost function is written as follows:

$$J = \int_0^\infty (\boldsymbol{x}^T(t)Q\boldsymbol{x}(\boldsymbol{t}) + \boldsymbol{u}^T(t)R\boldsymbol{u}(\boldsymbol{t}))\,dt \tag{27}$$

with the weighting matrices

$$Q = \begin{bmatrix} q_{11} & 0 \\ 0 & q_{22} \end{bmatrix} \qquad R = [\, r_1 \,]. \tag{28}$$

Then, the full state-feedback control law, $\boldsymbol{u}(t)=-\boldsymbol{K}(t)\boldsymbol{x}(t)$, minimizing the cost function, (27), can be calculated by solving the Algebraic Riccati Equation. The resulting control law is calculated as

$$\boldsymbol{K}(t) = [k_1(t) \;\; k_2(t)] \tag{29}$$

where the gains are calculated for each case at each time step online. The coefficients appearing in (28) are given in Table 2.

4. Simulation Results

In order to verify the feasibility of the optimal controller in the controlled system, simulation of disturbed ship condition is implemented. In the three case studies, the vessel used in the simulations is a 360 ton patrol navy vessel, which is based on the benchmark model given in [18]. The simulation parameters of the controlled system and the coefficients of the controllers are given in Table 1 and Table 2, respectively. In Table 3, the parameters used in the models of waves are given. It is important to note that, the parameters of the energy optimal controller is kept fixed, whereas those of LQR controller are recalculated at each time step, in all case studies.

Table 1. *Parameters of the Ship*

Parameter	Value
A_F (m^2)	3,4
K_P ($kg \cdot m^2/s$)	0,5 x 10^6
$K_{p\|p\|}$ ($kg{\cdot}m^2$)	0,416x10^6
GM (m)	1
$I_{xx}+\delta I_{xx}$ ($kg{\cdot}m^2$)	4.100.300
L_F (m)	4,22
U (m/s)	7,717
∇ (m^3)	355,88
ρ (kg/m^3)	1.025
C_{Lmax}	1,33
α_{max} (deg)	28,8
$\dot{\alpha}_{max}$ (deg/s)	25
K_a	1
T_a	0,366

Table 2. *Parameters of the Controller*

Parameter	Value
R_1	-11,644
R_2	1
R_3	1x10^7
q_{11}	1x10^8
q_{22}	1x10^8
r_1	1

Table 3. *Parameters of the Wave Models*

Parameter	Value
a_m (rad)	0,125
g (m/s^2)	9,81
x (m)	0
y (m)	0
T_1 (s)	2 π
$H_{1/3}$ ($sea\ state\ 2$) (m)	0.3
$H_{1/3}$ ($sea\ state\ 4$) (m)	2
s	100
r	100

In the first case study, for the purpose of examining the worst case scenario, the ship is excited by the sinusoidal wave, (9), with a frequency that is equal to the natural frequency of the ship, $\omega_e=\omega_\phi$. The response of the ship, the control input as the mechanical angle of the fins, α_m, and the wave elevation, η , as the disturbance input are shown in Fig. 2. At such severe situation, the roll angle of the controlled ship takes values between ±2 degrees, whereas it is between ±33 degrees in the uncontrolled ship.

To evaluate the performance of the stabilizer, one of the commonly used statistical index is the percentage reduction of statistics of roll and is defined as [18]:

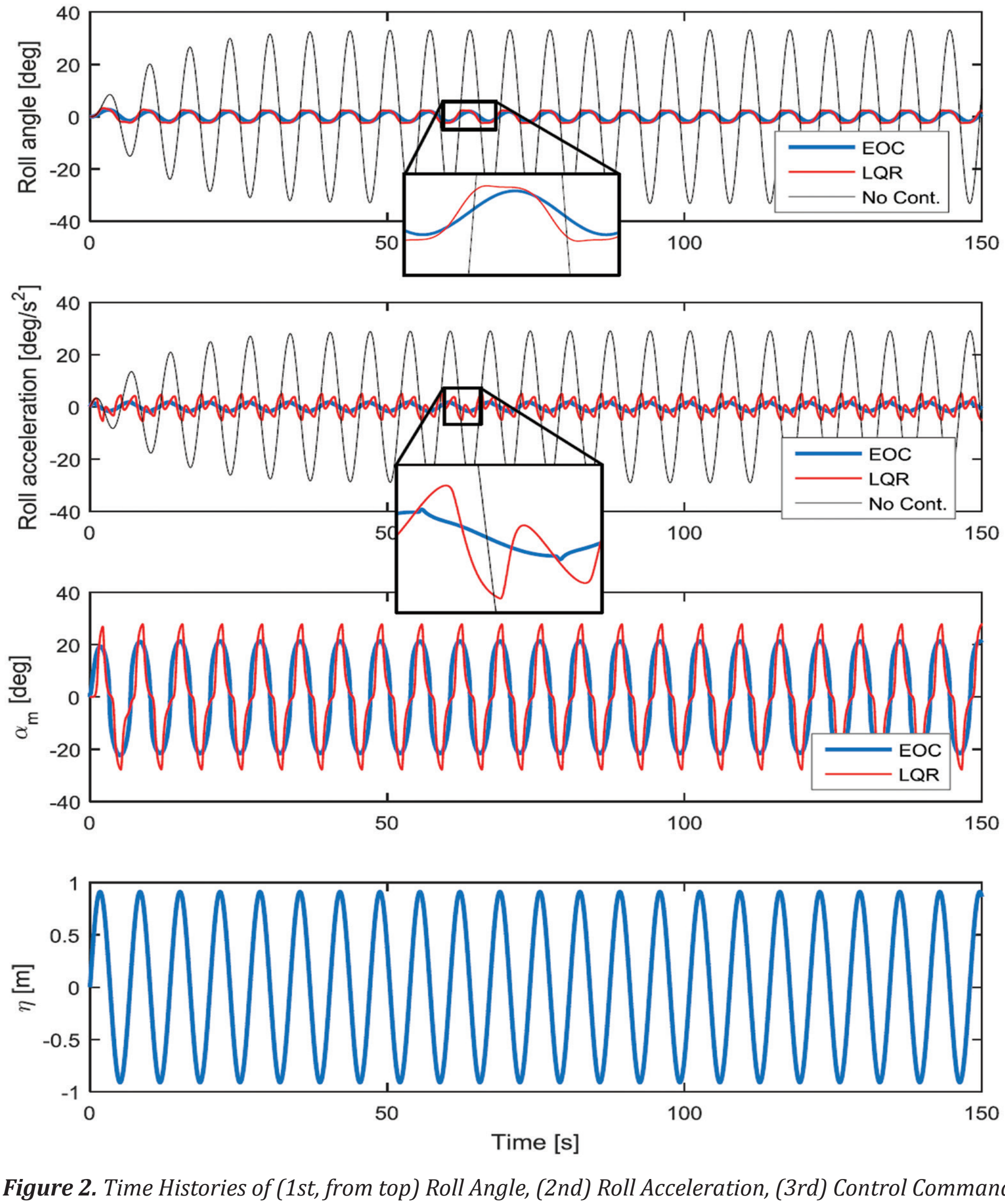

Figure 2. *Time Histories of (1st, from top) Roll Angle, (2nd) Roll Acceleration, (3rd) Control Command, and (4th) Sea Surface Elevation for the Periodic Disturbance Input*

$$RSR = 100\left(1 - \frac{S_s}{S_u}\right) \quad (30)$$

where the subscripts s and u stand for stabilized and unstabilized, respectively, and S mostly selected as variance or root mean square of roll motion evaluated

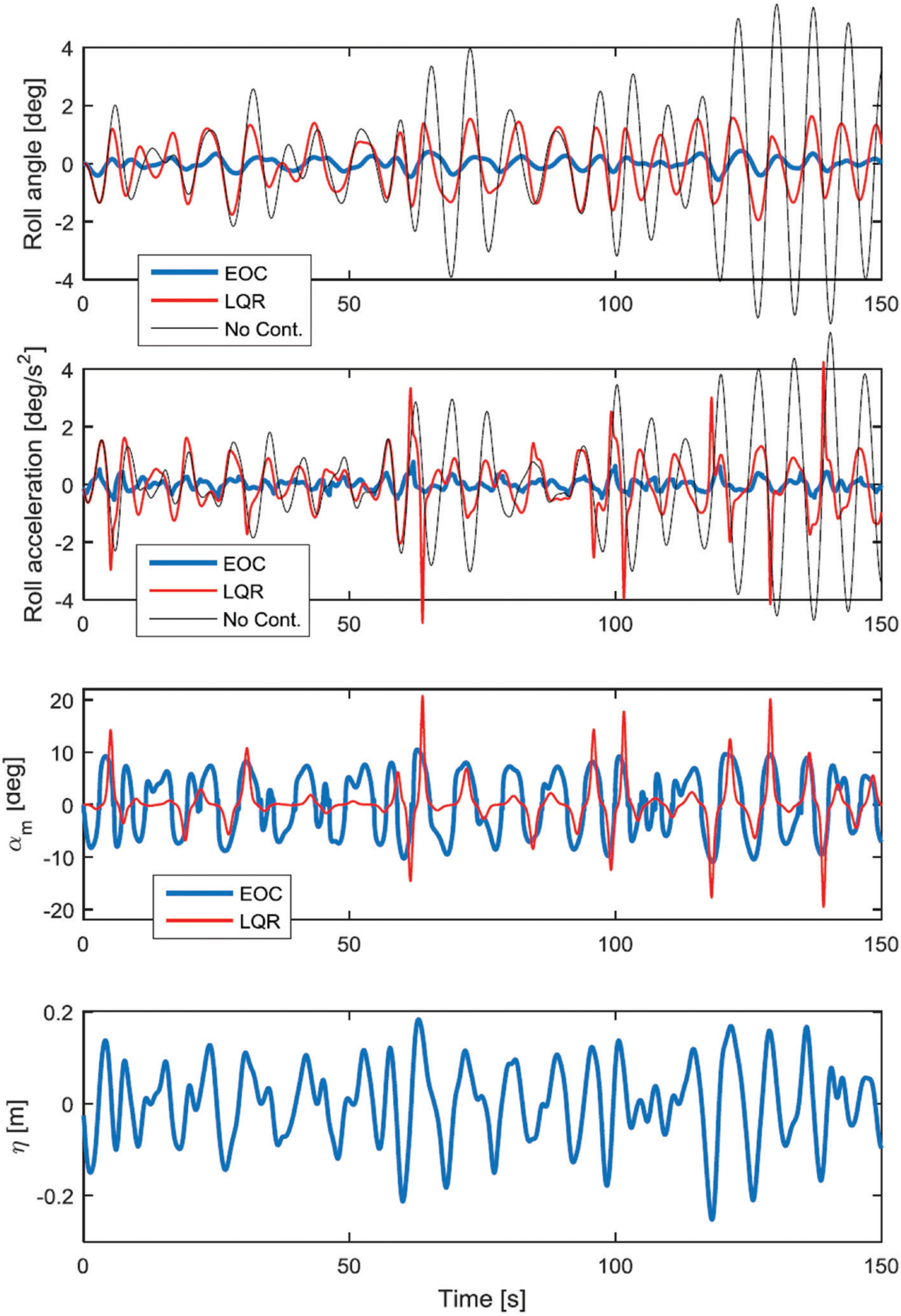

Figure 3. *Time Histories of (1st, from top) Roll Angle, (2nd) Roll Acceleration, (3rd) Control Command, and (4th) Sea Surface Elevation for the Random Disturbance Input, in the Sea State 2*

for a particular sea state. In this study, root mean square of the roll is selected.

The RSR values in the roll angle and roll acceleration motion are calculated by using the values of the corresponding time histories shown in Fig.2. The reductions in the roll angle and roll acceleration by the application of the energy optimal control (EOC) are calculated as 93,82% and 93,9%, respectively, whereas the values regarding the LQR controlled ship are 91,27% and 85,62%, respectively.

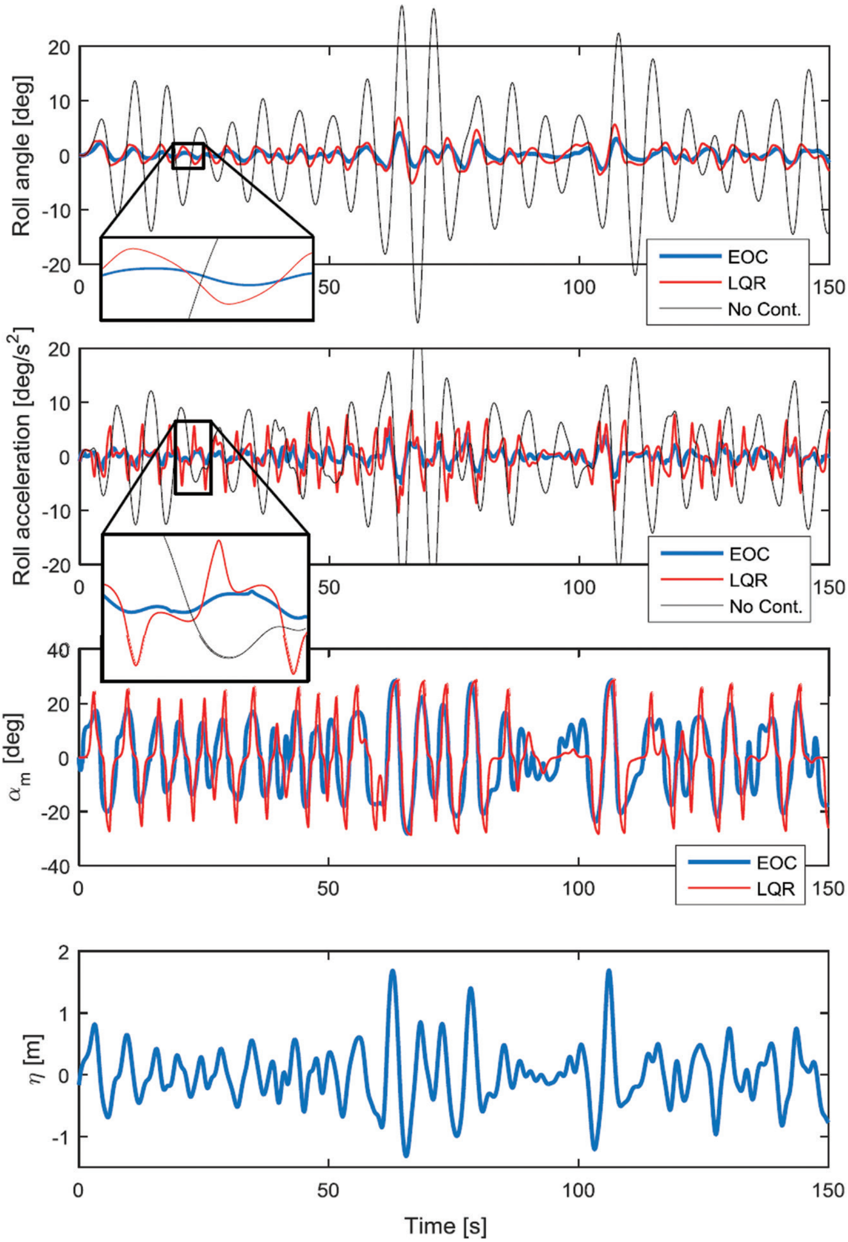

Figure 4. *Time Histories of (1st, from top) Roll Angle, (2nd) Roll Acceleration, (3rd) Control Command, and (4th) Sea Surface Elevation for the Random Disturbance Input, in the Sea State 4*

As seen in Figure 2, the stabilization performance of the EOC is better than that of the LQR control. Especially, the variations in the accelerations indicate that the EOC outperforms the LQR.

As the second and third case studies, behavior of the controlled ship under the effect of random sea waves is investigated for two different scenarios. The random sea waves are described by the ITTC spectrum. In the second case, the wave height is 0,3 m and the sea state is 2 (smooth). In the last case, the wave height is 2 m and the sea state is 4 (moderate). A magnitude constraint for the mechanical angle of the fins is imposed as 28,8 deg and the maximum rate is of 25 deg/s, without any constraint on the effective angle of attack.

The first two plots of Fig. 3 show the roll angle and roll acceleration in the controlled and uncontrolled ships. Regarding the EOC applied ship, the reductions in the roll angle and roll accelerations are 90,66% and 89,36%, respectively, whereas they are 55,96% and 46,17%, respectively, in the LQR controlled ship. The third and fourth plots show the control input to the hydraulic actuators and the environmental disturbance as the wave elevation, respectively.

The last case study is chosen to show the response of the ship in the moderate sea condition. The performance in stabilizing the ship appears as very satisfying as seen in Figure 4. In this case, the RSR values are 89,65% and 85,83% for the roll angle and the roll acceleration, respectively, in the EOC applied ship. In the LQR controlled ship, they are 81,50% and 64,53%, respectively.

Instead of exhibiting all simulation results for different sea states, only the RSR values can be used to indicate the effectiveness of the proposed controller. The RSR values of the roll angle and the roll acceleration of the EOC and LQR applied ships under the effect of random waves are shown in Figure 5. As the wave

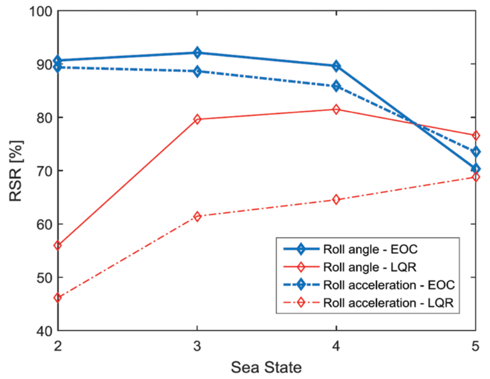

Figure 5. *RSR Values of the Roll Angle and the Roll Acceleration in the EOC and LQR Applied Ships under the Effect of Random Waves for Different Sea States*

elevation increases, an expected decrease in the stabilization performance of the proposed control system can be seen. However, up to sea state 5, the performance of the proposed controller can be accepted as highly satisfying. In real engineering problems, since the conditions over sea state 5 are mostly considered as severe cases [29], higher sea states are not studied in this work.

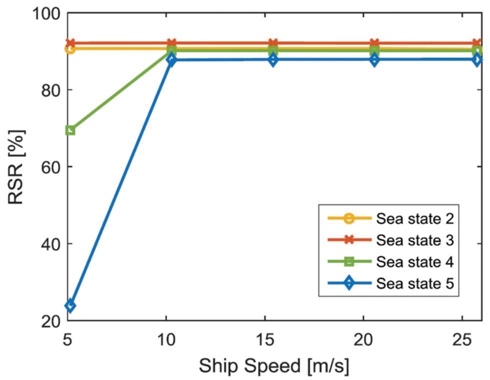

Figure 6. *RSR Values of the Roll Angle in the EOC Applied Ship under the Effect of Random Waves at Different Forward Speeds of the Ship for Different Sea States*

Another issue regarding the performance of the proposed controller at different ship forward speeds can be mentioned. The RSR values of the roll angle in the EOC applied ship with the speeds ranging from 10 knots to 50 knots in the sea states ranging from 2 to 5 are shown in Figure 6. Since the stabilizing torque generated by the fins depends on the velocity of the ship, at relatively low speeds, up to 10 m/s, and in high sea states, such as 4 and higher, the reduction in the roll motion is not significant. In the sea states 2 (smooth) and 3 (slight), the roll stabilization performance is highly satisfying even at low speeds. The RSR values of the roll acceleration are not given since they are very similar to those of roll angles.

5. Conclusion

An optimal controller to reduce the undesired roll motion of a ship with active fin stabilizers has been developed in this study. First, the roll dynamics as a one-degree-of-freedom nonlinear model has been presented, and then, a first-order actuator dynamics to represent the fin actuators has been given. The control force has been described as the moment generated by the lift, whose formulation includes the lift coefficient as a time-varying parameter. Thus, we were able to obtain realistic results. In the sequel, two different wave models, that cause disturbance moments in the ship, have been presented. They are the sinusoidal and random wave models that formulate the sea wave elevation. By disturbing the ship at its natural frequency, its resonance behavior has been tested by the periodic wave. On the other hand, the model of random waves generated by a stochastic model, which is used to test the real-life situations, has been presented.

Two sets of parameters describing smooth and moderate sea states have been used in the random wave model. By employing the energy optimal control method that allows both the closed-loop and real-time control of dynamic systems the controller has been obtained analytically. The performance of the controller, under the effect of disturbance inputs, has been tested through computer simulations.

To show the effectiveness of the proposed controller in a comparative study, the simulation results obtained by the application of the designed LQR controller have been presented. In the linearized model, the dynamics related with the variation of the lift coefficient is included by defining the system as time-varying. By having this dynamics in the control system, the results have become significant from the roll stabilization point of view. Studies with constant lift coefficient, which are not discussed in this work, had shown that time-invariant LQR controller cannot stabilize the ship.

In the case studies, it has shown that the proposed controller outperforms the LQR controller. Besides the better performance in the reduction of the roll angle, reduction of the roll acceleration is also remarkable in the proposed controller. The higher frequencies in the roll acceleration responses of the LQR controlled ship have indicated that such frequencies can be harmful to human performance, although the reduction in the roll angle might be acceptable. On the other hand, due to recalculation of the control gains of the LQR controller at each time step online, the computational cost has been too high compared to that of the proposed controller. The results have showed that the optimal controller achieves roll reduction satisfactorily.

In this study, the transport delay imposed by the hydraulic actuator system has been taken into consideration through the first-order actuator model. However, different values of transport delay are not discussed. The robustness of the proposed controller to the uncertainty due to time delay is considered as a future work.

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