Hindawi
Shock and Vibration
Volume 2017, Article ID 9095287, 13 pages
https://doi.org/10.1155/2017/9095287

Research Article

Design Fuzzy Input-Based Adaptive Sliding Mode Control for Vessel Lift-Feedback Fin Stabilizers with Shock and Vibration of Waves

Lihua Liang, Mingxiao Sun, and Tiantian Luan

College of Automation, Harbin Engineering University, Harbin 150001, China

Correspondence should be addressed to Mingxiao Sun; sunmingxiao1986@163.com

Received 25 November 2016; Revised 27 April 2017; Accepted 3 May 2017; Published 25 May 2017

Academic Editor: Mahmoud Bayat

An adaptive sliding mode controller based on fuzzy input design is presented, in order to reduce the roll motion of surface vessel fin stabilizers with shock and vibration of waves. The nonlinearities and uncertainties of the system including feedback errors and disturbance induced by waves are analyzed. And the lift-feedback system is proposed, which improves the shortage of conventional fin angle-feedback. Then the fuzzy input-based adaptive sliding mode control is designed for the system. In the controller design, the Lyapunov function is adopted to guarantee the system stability. Finally, experimental results demonstrate the superior performance of the controller designed using fuzzy input, when compared to the PID controller used in practical engineering.

1. Introduction

In rough ocean environment, excessive roll motion induced by shock and vibration of waves can severely affect the capacity of vessels to perform their missions and prevent the fighting effectiveness of naval vessels [1]. Severe roll motion degrades the performance of crews, can produce cargo damage, and may affect the operation of certain devices [2]. Therefore, many types of equipment with their associated control systems have been developed to reduce roll motion. Generally, the techniques are described as roll stabilization [3].

During the last decades, several widely taken devices include bilge keels, antirolling tanks, fin stabilizers, and rudder-roll reductions. Among the antirolling devices, fin stabilizer is one of the most attractive devices due to high feasibility and effectiveness. Especially for high speed vessels, it can provide considerable damping [4].

The key to antirolling performance of the fin stabilizer depends on its control and hydrodynamic characteristics of fin [4, 5]. On the basis of determining shape structure of fin, excellent control system and control strategy can develop hydrodynamic characteristics and improve antirolling effect. The theoretical effect of fin stabilizers can be more than 90% [5]. However, since many interference factors make system error too large, actual effect can not reach the theoretical index [6]. There are two main reasons for the errors: firstly, whether the system feedback is accurate; secondly, whether the structure of control system is reasonable and control strategy is effective. Therefore, in engineering applications, aiming at system error of fin stabilizers, the control system with nonlinearity and uncertainty and the disturbance should be considered especially [2, 7].

For the bigger feedback error in conventional fin stabilizer, it is mainly that the feedback is indirectly calculated through fin angle, which can not reflect real demand [8], while the actual lift force from dynamic hydrodynamic force is real feedback. However, the marine environment is rough, and installation problem of sensors is considered in practical engineering, so it is extremely difficult to detect directly actual lift force [9, 10]. As a result, fin angle-feedback control is generally adopted in conventional fin stabilizers. Since the simple and reliable measuring device located in the interior of hull is convenient for maintenance and replacement. Nevertheless, in fin angle-feedback, theoretical value of lift force is approximately calculated based on ideal constant hydrodynamic force. Actually, the dynamic hydrodynamic force is much more complicated than the corresponding static [3, 5, 8]. In addition, the relationship is seriously nonlinear and uncertain between fin angle and actual lift force due to interaction of fins, bilge keels, and hull induces [2, 11]. Therefore, there is a big error between theoretical value and actual value for lift force as the feedback.

In addition to the above, due to the design of conventional fin stabilizers based on linear roll model, the serious nonlinear and uncertain factors are neglected [1, 3]. To sum up, there are two problems: firstly, real vessels model causes serious nonlinearity; secondly, some design parameters are uncertain [5, 8]. These effects can be attributed to the nonlinear and uncertain problems of damping moment and restoring moment due to shock and vibration of random waves [11]. Thus, the control strategy should also be improved.

For control system of fin stabilizers, the design of controller develops an important role in achieving roll stabilization. Several control strategies have been presented for fin stabilizers, including the conventional control and advanced control strategies [3].

In practical engineering applications, most surface vessels can be stabilized equipping with fin stabilizers controlled by PID method [12, 13]. PID controller is simple and it is of high reliability. However, the PID control strategy can only play well for a limited number of vessels types and environmental conditions. Since a nonadaptive controller may produce roll exacerbation rather than roll stabilization, one of the key issues in antirolling controllers is whether it can adapt to changes in the rough random waves. In addition, PID controller design depends on the exact model [7, 8, 12, 14, 15], while the rolling frequency, damping coefficient and loading conditions, and other variable parameters can directly affect the control effect.

In dealing with nonlinearities, uncertainties, and wave disturbances in the control system of fin stabilizers, some advanced control strategies have implemented better antirolling performances than the classical strategies. For example, Perez and Goodwin [1] discussed the constrained predictive control of vessel fin stabilizers to prevent dynamic stall. Hinostroza et al. [3] studied a robust fin control for vessel roll stabilization based on $L2$-gain design. Song et al. [5] designed an inverse controller of zero-speed fin stabilizer based on RBF neural network. Liu et al. [6] considered an extended radiated energy method and its application to a vessel roll stabilization control system. Fang and Luo [10] researched on the track keeping and roll reduction of the vessel in random waves using different sliding mode controllers. Fang et al. [12] studied the application of the self-tuning neural network PID controller on the vessel roll reduction in random waves. Surendran et al. [13] researched on an algorithm to control the roll motion using active fins. Crossland [14] analyzed the effect of roll stabilization controllers on war-vessel operational performance. Ling et al. [7] designed a fuzzy-PID controlled lift-feedback fin stabilizer. Xia et al. [16] applied a fuzzy neural network-based robust adaptive control for dynamic positioning of underwater vehicles with input dead-zone. Wang and Meng [17] proposed a fuzzy modeling and control of vessel lift-feedback fin stabilizer system. Xiu and Ren [18] implemented a fuzzy controller design and stability analysis for vessel's lift-feedback-fin stabilizer. Wang [19] developed the fuzzy systems which are universal approximators. Yoo and Ham [20] addressed an adaptive control of robot manipulators using fuzzy compensator. Yang et al. [21] presented a robust adaptive fuzzy control and its application to vessel roll stabilization. Li et al. [22] designed an adaptive neural network approach for vessel roll stabilization via fin control. Alarçïn [23] discussed the internal model control using neural network for vessel roll stabilization. Lee et al. [24] implemented design of the roll stabilization controller. Luo et al. [25] proposed robust fin control for vessel roll stabilization by using functional-link neural networks. Moradi and Malekizade [26] designed a robust adaptive first–second-order sliding mode control to stabilize the uncertain fin-roll dynamic. Perez and Blanke [27] investigated a vessel roll damping control. Therefore, there have been some applications and, universally, developing fuzzy adaptive control is of interest in the studies on roll reduction.

Among the control strategies, the fuzzy adaptive control is one of the main choices and develops an effective way to deal with nonlinearities, uncertainties, and wave disturbances. Due to the variable structure control, the perturbation has no influence on the sliding mode, which is completely adaptive to the disturbance and parameter changes of the system. In the design of sliding mode controller, the upper bound value of wave interference and system uncertainty need to be estimated, but this is very difficult for fin stabilizers [28]. In the sliding mode surface, the switching gain is difficult to be determined, so the actual control is often determined by experience. If it is chosen too large, then system will have a large chattering. Conversely, if it is chosen too small, then control system is unstable. To solve this problem, fuzzy control is adopted to adjust the control switching gain [29, 30]. Therefore, in this paper, a new fuzzy input-based adaptive sliding mode control is designed for vessel lift-feedback fin stabilizers.

The structure of this paper is organized as follows. Section 2 describes the control problem of vessel lift-feedback fin stabilizers subject to nonlinear and uncertain terms and disturbances. Section 3 presents the design of adaptive sliding mode controller. In Section 4, simulation studies of experimental system are described to verify the effectiveness of the proposed control scheme. Finally, conclusions are made in Section 5.

2. Problem Formulation

2.1. Mathematical Model Analysis. According to Conolly theory, the nonlinear roll model of a vessel equipped with fin stabilizer is approximated to

$$\left(I_x + \Delta I_x\right) \frac{d^2\theta}{dt^2} + N_1 \frac{d\theta}{dt} + N_2 \left|\frac{d\theta}{dt}\right| \frac{d\theta}{dt} + G_1\theta + G_3\theta^3 + G_5\theta^5 = -K_\omega - K_c, \tag{1}$$

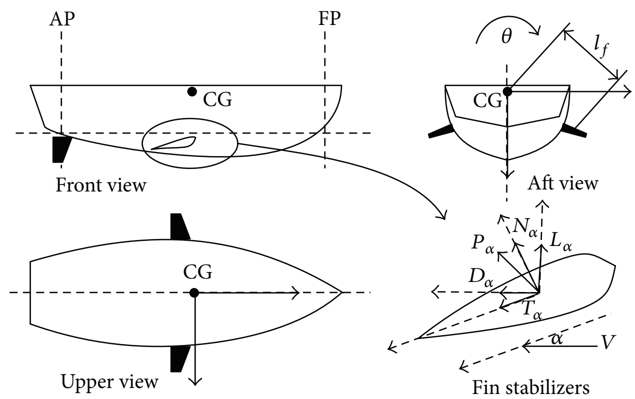


Figure 1: Vessel and fin stabilizers.

where $(I_x + \Delta I_x)d^2\theta/dt^2$ is inertia moment of the vessel. $N_1 d\theta/dt + N_2|d\theta/dt|d\theta/dt$ is damping moment of the vessel. $G_1\theta + G_3\theta^3 + G_5\theta^5$ is restoring moment. K_ω is disturbance moment of waves. K_c is control moment of the fin stabilizer.

Ideally, if the disturbance moment of waves is completely offset by control moment of fin stabilizers, the vessel is roll stabilization. Thus, the accurate control moment is the key of roll stabilization.

In the dynamic hydrodynamic force of fins, lift force is the real factor of roll reduction. Since the fin arrangement is symmetrical, when the lift force of one side fin is upward, the lift force of the other side fin is down.

So the control moment can be expressed as

$$K_c = 2L_\alpha l_\alpha, \tag{2}$$

where L_α is the lift force on a fin. l_α is the arm of control moment.

The vessel and fin stabilizers are shown in Figure 1.

Therefore, the key of control problem is focused on accurate feedback methods and control strategies.

2.2. Error Analysis

2.2.1. Feedback Error. According to the force coefficient calculation theory for fin in viscous fluid, when the angle between fin and flow is fixed, lift force is approximately kept constant, which is shown as follows:

$$L_\alpha = \frac{1}{2}\rho V^2 A C_0 \left(1 - \frac{A_2}{A_0}\right)(1+k)\,\alpha, \tag{3}$$

where ρ is the fluid density. V is the fluid velocity. A is the projection area of fins. α is the fin angle. $k = \tan(\beta/\alpha)$. $C_0 = 4\mu A_0/\rho V^2$, $A_0 = (\sqrt{2}a_0/\delta_1)\theta_0$, $\theta_0 = \arctan(V\cos\alpha/\sqrt{2}a_0) - \arctan(\mu/\sqrt{2}a_0)$, $A_2 = (\sqrt{2}a_0/\delta_1)n\pi$, $(n = 0, 1, 2, \ldots)$. μ is viscosity coefficient. β is the angle between the traction force of fins and the velocity of infinity. a_0 is the velocity of stagnation point. δ_1 is the thickness of boundary condition.

Defining lift coefficient $C_L(\alpha)$, then

$$C_L(\alpha) = C_0\left(1 - \frac{A_2}{A_0}\right)(1+k)\,\alpha. \tag{4}$$

Figure 2: Experimental fins used in water tank experiment.

Table 1: Main model parameters of fin.

Parameter	Value	Unit
Chord length of root	571.2	mm
Chord length of tip	316.8	mm
Fin height	236	mm
Shaft distances root	254	mm
Shaft distances tip	182.17	mm
Sweepback	31.33	mm
Shaft coordinate	−0.18	deg
$l_0/571.2$	0.445	mm
$l_0/444$	0.410	mm

From (2) and (4), before the fin stall, lift coefficient and fin angle are linear. In practical engineering, the system of fin stabilizers is designed according to this kind of linear relation. But in fact, the dynamic hydrodynamic characteristics of fins are very complex.

In order to obtain the relationship between lift coefficient and fin angle, the dynamic hydrodynamic characteristics of fins in water tank are done. And the main experimental fins are shown in Figure 2.

The NACA0015 fin type is chosen to carry out hydrodynamic experiments. The main model parameters of fin are shown in Table 1.

The water tank experimental results are shown in Figure 3.

In Figure 3, model 1~model 5 are the experimental results of dynamic hydrodynamic force for fin in water tank. The experimental status table is shown in Table 2. And model 6 is static relationship between lift coefficient and fin angle.

This experimental results show that the dynamic relationship between lift coefficient and fin angle is not an ideal linear relationship, while the relationship presents a closed

Table 2: Experimental status of fin.

Mode	Speed	Swing	Swing period	Dimensionless frequency
Mode 1	3.0	25	2.467	0.06
Mode 2	3.0	25	3.700	0.04
Mode 3	3.0	25	4.933	0.03
Mode 4	3.0	25	6.167	0.024
Mode 5	3.0	25	7.400	0.02

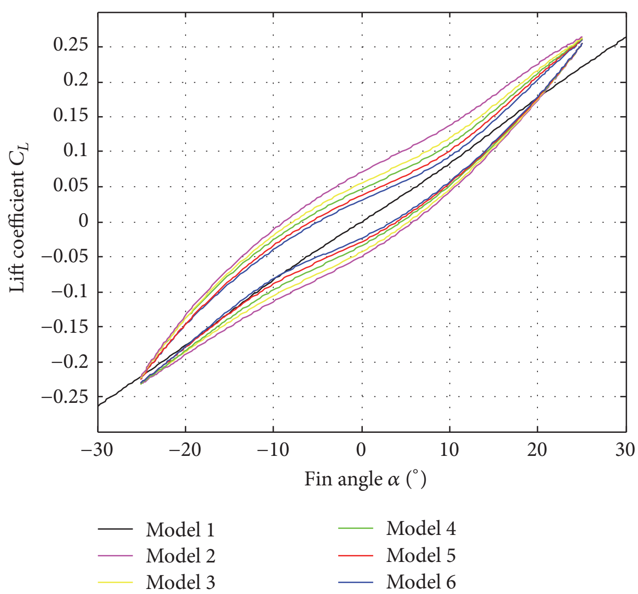


Figure 3: Relationship between lift coefficient and fin angle.

curve. There is large error between theoretical value and actual value; that is, the large error exists in the feedback of fin stabilizers. Therefore, the fin stabilizer has difficulty in accurately resisting wave disturbance, which will definitely reduce the antirolling effect.

2.2.2. Control Error. In practical applications, the conventional PID control is adopted in linear roll model of a vessel equipped with fin stabilizer. But the model is simplified as follows.

(1) Assume that the roll motion of a vessel is independent, and there are not coupling factors and other degrees of freedom motion.

(2) Assume that vessel width is much smaller than wave length.

(3) Internal pressure field of the waves is not affected by a vessel.

But in fact, the motion of a vessel in the ocean waves is a complex six-degree-of-freedom movement, in which there is a certain degree of coupling among the roll, sway, and yaw. And internal pressure field is not ideal owing to the existence of a vessel, so it will have a certain impact on the internal pressure field.

The rolling period of roll model

$$T_\phi = \frac{2\pi}{\sqrt{Dh/\left(I_{xx} + \Delta I_{xx}\right)}}. \quad (5)$$

The dimensionless damping coefficient

$$n_\mu = \frac{N_\mu}{\sqrt{Dh\left(I_{xx} + \Delta I_{xx}\right)}}. \quad (6)$$

For different types of vessels, T_ϕ and n_μ are different because of the different design parameters. The calculations are determined by empirical formulas or experiments, which are uncertain and variable.

Fin angle-feedback control is usually used in conventional fin stabilizers, which is based on the static linear relationship. Its actual antirolling effect has difficulty in reaching theoretical design requirements. In Figure 3, the experimental results of water tank prove the problem. So the control system and controller are to be designed according to actual lift.

2.3. Lift-Feedback Control System Design

2.3.1. System Composition. For the fin stabilizers, the control principle is different between lift-feedback and conventional fin angle-feedback, so there is a difference in the system composition. The composition of lift-feedback fin stabilizer is three parts: integrated controller, electrohydraulic servo system, and vessel state feedback part, as shown in Figure 4.

2.3.2. Working Process. When the vessel is subjected to the action of disturbance moment, the sensors of state feedback part can measure roll angle, roll velocity, and roll acceleration. The signals are input to data processor, and they are adjusted, calculated, and amplified, and then they are transmitted to integrated controller as input quantity.

In the integrated controller, the signal is calculated by control strategy, and the lift force needed for fin stabilizers to counter disturbance moment in real time is obtained, which is used as the control signal input to servo system.

The lift control signal is converted to the instruction of turning fin by angle amplifier, which is transmitted to electrohydraulic servo valve. The hydraulic cylinder drives fin to rotate for the corresponding angle. Due to the effect of fluid dynamics, the control moment is generated to resist disturbance moment by fin. At the same time, the actual lift force is measured directly by lift sensor, which is transmitted to servo system controller as a feedback signal.

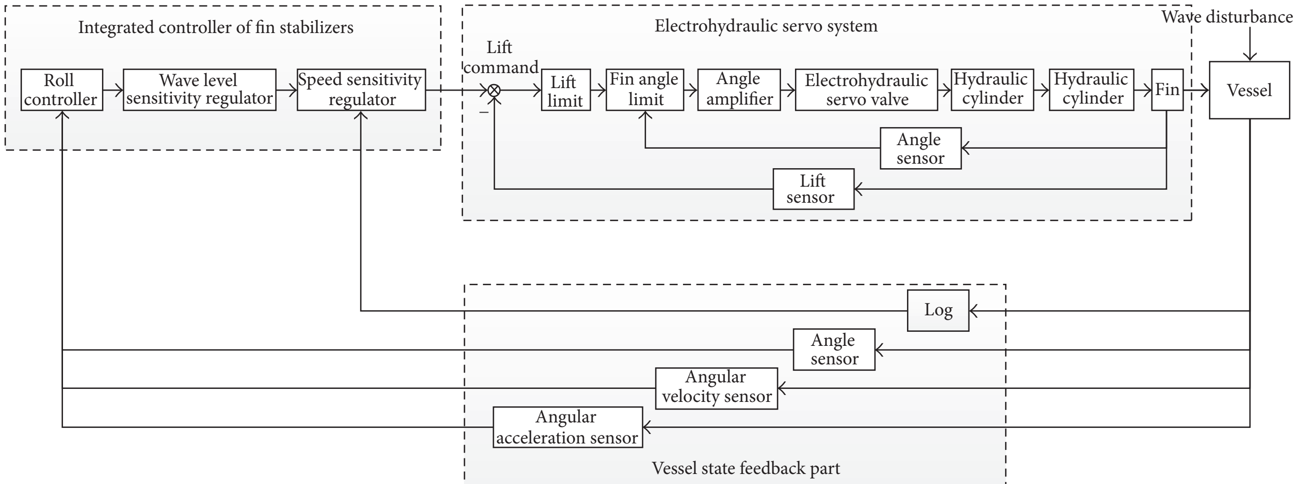


FIGURE 4: Composition of lift-feedback system.

2.3.3. Improved Advantage. In control form, the lift-feedback fin stabilizer is similar to fin angle-feedback fin stabilizer. But there are essential differences, which are mainly reflected in the following aspects.

(1) The output of controller is lift force, which is the direct control of the vessel's control command instead of indirect fin angle command.

(2) The effect of speed sensitivity regulator is different. In the fin angle-feedback system, the speed sensitivity regulator is generally placed in the output circuit of controller, and its function is to ensure the stability in the same roll state. And the speed of lift-feedback system is as an input signal of controller. The output of controller is detected by speed sensitivity regulator, which ensures that the output command is always realized. So the system can be guaranteed to work properly.

(3) The wave sensitivity regulator can limit the saturation rate of the system to the maximum stable moment, which can avoid the wear of mechanical devices.

3. Controller Design

3.1. System Description. According to (2) in Section 2, for the fin stabilizer system, the control moment is as shown:

$$K_C = W_C u, \tag{7}$$

where u is a lift control instruction. W_C is the transfer function of the actuator in the hydraulic servo system.

According to (1) and (7), the following nonlinear roll model of fin stabilizer system is considered:

$$\ddot{\theta} = f(\theta, t) + g(\theta, t)\, u(t) + d(t), \tag{8}$$

where $g > 0$. $d(t)$ is unknown disturbance, which is caused by the shock and vibration of the waves:

$$\begin{aligned} f(x,t) &= \frac{-\left(N_1\dot{\theta} + N_2\left|\dot{\theta}\right|\dot{\theta} + G_1\theta + G_3\theta^3 + G_5\theta^5\right)}{\left(I_x + \Delta I_x\right)}, \\ g(\theta,t)\,u(t) &= \frac{-W_C u}{\left(I_x + \Delta I_x\right)}, \\ d(t) &= \frac{-K_\omega}{\left(I_x + \Delta I_x\right)}. \end{aligned} \tag{9}$$

Due to the complex shape of a vessel and the interaction between the vessel and waves, I_x and ΔI_x are empirical or experimental formulas to determine. So they are uncertain and changeable. In addition, the damping coefficients of the vessel N_1 and N_2 are related to many factors, such as hull shape, loading, frequency, amplitude, flow field near the hull, and water viscosity. These variable factors cause them to become uncertain factors. $f(\theta, t)$ and $g(\theta, t)$ are related to these factors of I_x, ΔI_x, N_1, and N_2. So $f(\theta, t)$ and $g(\theta, t)$ are the unknown nonlinear functions.

3.2. Controller Design. For the fin stabilizer system, $\theta_d(t)$ is angle instruction, and the tracking error is $e(t) = \theta(t) - \theta_d(t)$.

The integral sliding mode surface is defined as

$$s(t) = \dot{\theta}(t) - \int_0^t \left(\ddot{\theta}_d - k_1\dot{e}(t) - k_2 e(t)\right) dt, \tag{10}$$

where k_1 and k_2 are nonzero positive constant.

If the fuzzy control is in the ideal state, then $s(t) = \dot{s}(t) = 0$. That is,

$$\ddot{e}(t) + k_1\dot{e}(t) + k_2 e(t) = 0. \tag{11}$$

Tracking error $e(t)$ can converge to zero by determining k_1 and k_2.

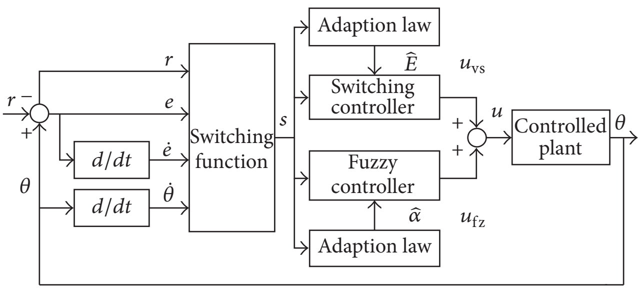


FIGURE 5: Structure of adaptive fuzzy sliding mode controller.

If f, g, and d are known, according to (8), the control law is obtained as follows:

$$u^*(t) = g(\theta,t)^{-1} \cdot \left(-f(\theta,t) - d(t) + \ddot{\theta}_d - k_1\dot{e}(t) - k_2 e(t)\right). \quad (12)$$

If f, g, and d are unknown, $u^*(t)$ is difficult to achieve. The approximation method of fuzzy system can be used to realize the approximation of ideal control law $u^*(t)$.

Supposing that α_i is adjustable parameters and $\boldsymbol{\xi}$ is fuzzy basis vector, then the control law is

$$u_{fz}(s,\boldsymbol{\alpha}) = \boldsymbol{\alpha}^T\boldsymbol{\xi}^T, \quad (13)$$

where $\boldsymbol{\alpha} = [\alpha_1, \alpha_2, \ldots, \alpha_m]^T$ and $\boldsymbol{\xi} = [\xi_1, \xi_2, \ldots, \xi_m]$. ξ_i is defined as $\xi_i = \omega_i / \sum_{i=1}^m \omega_i$.

According to the fuzzy approximation theory, there exists an optimal fuzzy system $u_{fz}^*(s,\boldsymbol{\alpha}^*)$ to approach $u^*(t)$:

$$u^*(t) = u_{fz}(s,\boldsymbol{\alpha}^*) + \varepsilon = \boldsymbol{\alpha}^{*T}\boldsymbol{\xi} + \varepsilon, \quad (14)$$

where ε is the approximation error and satisfies $|\varepsilon| < E$.

The fuzzy system u_{fz} is used to approach $u^*(t)$. Then

$$u_{fz}(s,\hat{\boldsymbol{\alpha}}) = \hat{\boldsymbol{\alpha}}^T\boldsymbol{\xi}, \quad (15)$$

where $\hat{\boldsymbol{\alpha}}$ is the estimated value of $\boldsymbol{\alpha}^*$.

Switching control law u_{vs} is used to compensate for the uncertainties between $u^*(t)$ and u_{fz}:

$$u(t) = u_{fz} + u_{vs}. \quad (16)$$

The structure of adaptive fuzzy sliding mode controller is shown in Figure 5.

3.3. Adaptive Control Algorithm Design. According to (14), we can get

$$\tilde{u}_{fz} = \hat{u}_{fz} - u^* = \hat{u}_{fz} - u_{fz}^* - \varepsilon. \quad (17)$$

Define

$$\tilde{\alpha} = \hat{\alpha} - \alpha^*. \quad (18)$$

Equation (17) transforms into

$$\tilde{u}_{fz} = \tilde{\boldsymbol{\alpha}}^T\boldsymbol{\xi} - \varepsilon. \quad (19)$$

It can be obtained by (15) as follows.

$$\dot{s}(t) = \ddot{e}(t) + k_1\dot{e}(t) + k_2 e(t). \quad (20)$$

Then (12) is transformed into

$$\begin{aligned} u^*(t) &= g(\theta,t)^{-1}\left(-f(\theta,t) - d(t) + \ddot{\theta}_d + \ddot{e}(t) - \dot{s}(t)\right) \\ &= g(\theta,t)^{-1}\left(-f(\theta,t) - d(t) + \ddot{\theta}(t) - \dot{s}(t)\right) \\ &= g(\theta,t)^{-1}\left(g(\theta,t)u(t) - \dot{s}(t)\right). \end{aligned} \quad (21)$$

It can be obtained by (16) and (21) as follows:

$$\begin{aligned} \dot{s}(t) &= g(\theta,t)\left(u(t) - u^*(t)\right) \\ &= g(\theta,t)\left(u_{fz} + u_{vs} - u^*(t)\right). \end{aligned} \quad (22)$$

Define Lyapunov function as

$$V_1(t) = \frac{1}{2}s^2(t) + \frac{g(\theta,t)}{2\eta_1}\tilde{\boldsymbol{\alpha}}^T\tilde{\boldsymbol{\alpha}}, \quad (23)$$

where η_1 is a positive constant.

Then

$$\begin{aligned} \dot{V}_1(t) &= s(t)\dot{s}(t) + \frac{g(\theta,t)}{\eta_1}\tilde{\boldsymbol{\alpha}}^T\dot{\tilde{\boldsymbol{\alpha}}} \\ &= s(t)g(\theta,t)\left(u_{fz} + u_{vs} - u^*(t)\right) + \frac{g(\theta,t)}{\eta_1}\tilde{\boldsymbol{\alpha}}^T\dot{\tilde{\boldsymbol{\alpha}}} \\ &= s(t)g(\theta,t)\left(\tilde{u}_{fz} + u_{vs}\right) + \frac{g(\theta,t)}{\eta_1}\tilde{\boldsymbol{\alpha}}^T\dot{\tilde{\boldsymbol{\alpha}}} \\ &= s(t)g(\theta,t)\left(\tilde{\boldsymbol{\alpha}}^T\boldsymbol{\xi} - \varepsilon + u_{vs}\right) + \frac{g(\theta,t)}{\eta_1}\tilde{\boldsymbol{\alpha}}^T\dot{\tilde{\boldsymbol{\alpha}}} \\ &= g(\theta,t)\tilde{\boldsymbol{\alpha}}^T\left(s(t)\boldsymbol{\xi} + \frac{\dot{\tilde{\boldsymbol{\alpha}}}}{\eta_1}\right) \\ &\quad + s(t)g(\theta,t)\left(u_{vs} - \varepsilon\right). \end{aligned} \quad (24)$$

In order to achieve $\dot{V}_1(t) \le 0$, the following adaptive law and the switching control are adopted:

$$\dot{\tilde{\boldsymbol{\alpha}}} = \dot{\hat{\boldsymbol{\alpha}}} = -\eta_1 s(t)\boldsymbol{\xi}, \quad (25)$$

$$u_{vs} = -E(t)\,\mathrm{sgn}(s(t)). \quad (26)$$

Then

$$\begin{aligned} \dot{V}_1(t) &= -E(t)|s(t)|g(\theta,t) - \varepsilon s(t)g(\theta,t) \\ &\le -E(t)|s(t)|g(\theta,t) - |\varepsilon||s(t)|g(\theta,t) \\ &= -(E(t) - |\varepsilon|)|s(t)|g(\theta,t) \le 0. \end{aligned} \quad (27)$$

In the switching controller, the switching gain $E(t)$ is difficult to be determined, so the actual control is often determined by experience. If $E(t)$ is chosen too large, then system will have a large chattering. Conversely, if $E(t)$ is chosen too small, then control system is unstable.

$\widehat{E}(t)$ is used to replace $E(t)$; then (26) is transformed into

$$u_{vs} = -\widehat{E}(t)\operatorname{sgn}(s(t)), \tag{28}$$

where $\widehat{E}(t)$ is estimated switching gain.

Define estimation error as

$$\widetilde{E}(t) = \widehat{E}(t) - E. \tag{29}$$

The Lyapunov function of the closed loop system is defined as

$$\begin{aligned} V(t) &= V_1(t) + \frac{g(\theta,t)}{2\eta_2}\widetilde{E} \\ &= \frac{1}{2}s^2(t) + \frac{g(\theta,t)}{2\eta_1}\widetilde{\boldsymbol{\alpha}}^T\widetilde{\boldsymbol{\alpha}} + \frac{g(\theta,t)}{2\eta_2}\widetilde{E}^2, \end{aligned} \tag{30}$$

where η_1 and η_2 are positive constants.

Then

$$\begin{aligned} \dot{V}(t) &= s(t)\dot{s}(t) + \frac{g(\theta,t)}{\eta_1}\widetilde{\boldsymbol{\alpha}}^T\dot{\widetilde{\boldsymbol{\alpha}}} + \frac{g(\theta,t)}{\eta_2}\widetilde{E}\dot{\widetilde{E}} \\ &= g(\theta,t)\widetilde{\boldsymbol{\alpha}}^T\left(s(t)\boldsymbol{\xi} + \frac{1}{\eta_1}\dot{\widetilde{\boldsymbol{\alpha}}}\right) \\ &\quad + s(t)g(\theta,t)(u_{vs} - \varepsilon) + \frac{g(\theta,t)}{\eta_2}\widetilde{E}\dot{\widetilde{E}} \\ &= -\widehat{E}(t)|s(t)|g(\theta,t) - \varepsilon s(t)g(\theta,t) \\ &\quad + \frac{g(\theta,t)}{\eta_2}\left(\widehat{E}(t) - E\right)\dot{\widehat{E}}(t). \end{aligned} \tag{31}$$

In order that $\dot{V} \le 0$, adaptive law is defined as

$$\dot{\widehat{E}}(t) = \eta_2|s(t)|. \tag{32}$$

Then

$$\begin{aligned} \dot{V}(t) &= -\widehat{E}(t)|s(t)|g(\theta,t) - \varepsilon s(t)g(\theta,t) \\ &\quad + \left(\widehat{E}(t) - E\right)|s(t)|g(\theta,t) \\ &= -\varepsilon s(t)g(\theta,t) - E|s(t)|g(\theta,t) \\ &\le |\varepsilon||s(t)|g(\theta,t) - E|s(t)|g(\theta,t) \\ &= -(E - |\varepsilon|)|s(t)|g(\theta,t) \le 0. \end{aligned} \tag{33}$$

Thus, when $\dot{V}(t) \equiv 0$, $s(t) \equiv 0$, according to the LaSalle invariant set theory, $t \to \infty$, $s \to 0$.

In view of the serious nonlinearity and uncertainty of system, the fuzzy input-based adaptive sliding mode controller is designed for fin stabilizer according to the proposed control law. Because of the uncertainty of wave disturbance $d(t)$ and system parameters $f(\theta,t)$ and $g(\theta,t)$, the membership functions are used to make control input fuzzy. Therefore, the uncertain system parameters of fin stabilizer and the sensitivity of external disturbance need to be verified.

The input is $s(t)$ and the output is $u^*(t)$.

The fuzzy rules are as follows:

(1) If $s(t)$ is NM, then $u^*(t)$ is PM.

(2) If $s(t)$ is NS, then $u^*(t)$ is PS.

(3) If $s(t)$ is Z, then $u^*(t)$ is Z.

(4) If $s(t)$ is PS, then $u^*(t)$ is NS.

(5) If $s(t)$ is PM, then $u^*(t)$ is NM.

4. Results and Discussion

4.1. Experimental Equipment and Steps. The test platform system of fin stabilizer and vessel roll simulator are used to simulate the motion of the vessel equipped with fin stabilizer in shock and vibration of waves. And the composition of the experimental system is shown in Figure 6.

The main steps of the experiments are as follows.

Step 1 (vessel motion is simulated in waves). According to the mathematical model of the waves and vessel, the vessel roll simulator is driven by industry personal computer (IPC) to simulate the motion of the vessel in shock and vibration of waves. Then the speed, roll angle, roll velocity, and roll angular velocity are fed back to IPC.

Step 2 (fin stabilizer is driven to reduce roll). The state of the vessel is computed by IPC. Then the fin stabilizer is driven through the controller. The fin stabilizer can reduce the roll angle dynamically according to the real-time feedback.

Step 3 (running states are controlled in IPC). The running states of vessel roll simulator and fin stabilizer are transmitted to IPC and controlled in real time. And the experimental results can be obtained on the monitor.

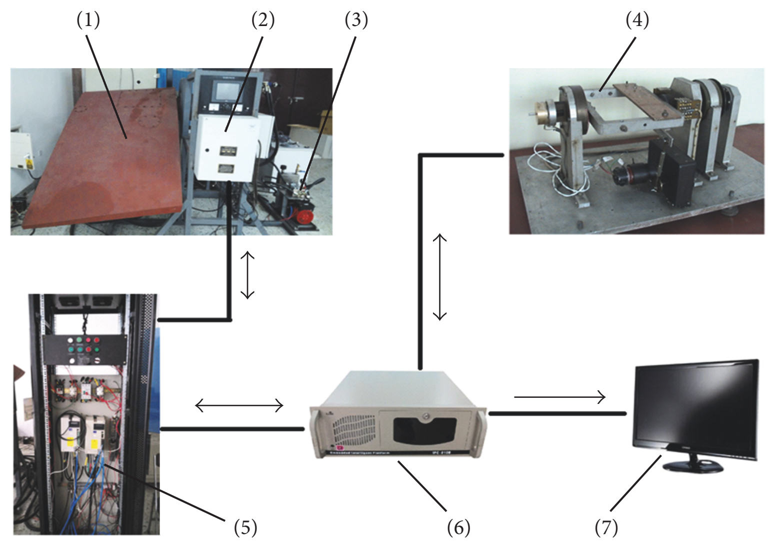
Step 4 (dynamic response is analyzed). The fuzzy input-based adaptive sliding mode controller and PID controller are carried out. And the dynamic response of the two is compared.

Step 5 (the antirolling performances are compared). At different speeds and encounter angle, the antirolling performances of two controllers are carried out in the shock and vibration of random waves.

4.2. Experimental Parameters. Here, the waves are simulated using ITTC double parameter spectrum, which are adopted to recreate the fully developed sea environment. The spectral density formula is presented as follows:

$$S(\omega_i) = \frac{173H_{1/3}}{T^4\omega_i^5}\exp\left(-\frac{691}{T^4\omega_i^4}\right), \tag{34}$$

where $H_{1/3}$ is the significant wave height. T is the wave period. ω_i is the wave frequency of the ith regular wave component.



(1) Fin
(2) Hydraulic servo starter
(3) Hydraulic servo system
(4) Vessel roll simulator
(5) Controller of fin stabilizer
(6) Industry personal computer
(7) Monitor

Figure 6: Structure of experimental system.

The irregular waves are formed by 60 regular wave components. The amplitude of each regular wave component ζ_i and the resultant wave elevation ζ can be obtained by

$$\zeta_i = \sqrt{2S(\omega_i)\Delta\omega},$$
$$\zeta = \sum_{i=1}^{60} \zeta_i \cos(\omega_i t + \varepsilon_i), \tag{35}$$

where ε_i is the random phase angle of the ith regular wave, which ranges from 0 to 2π. In this paper, the resultant wave elevation ζ is adopted to calculate the external forces.

In order to test the performance of the designed controller, a real vessel is simulated in the experimental system as an example.

The vessel parameters are displacement $D = 1500\,\mathrm{t}$, length $L = 98.0\,\mathrm{m}$, beam $B = 10.2\,\mathrm{m}$, draft $T = 3.1\,\mathrm{m}$, metacenter height $h = 1.15\,\mathrm{m}$, and resonant period $T = 7.8\,\mathrm{s}$. And the disturbance moment acting on the vessel is about $10^5 \sim 10^6$ N·m. The rolling motion of the vessel is stopped when the control moment of the fin stabilizer is equal to the disturbance moment of the waves. So the control moment should be as close as possible to the disturbance moment.

According to (8) and (9), the nonlinear roll model with the corresponding lift-feedback fin stabilizer is

$$\ddot{\theta} = -0.25174\dot{\theta} - 0.7056\left|\dot{\theta}\right|\dot{\theta} - 0.648366\theta + 15.7696\theta^3 - 20.65\theta^5 - 0.00973u - 0.39479e^{-7}K_{\omega}. \tag{36}$$

According to (15), (16), (25), (28), and (32), then the control law is

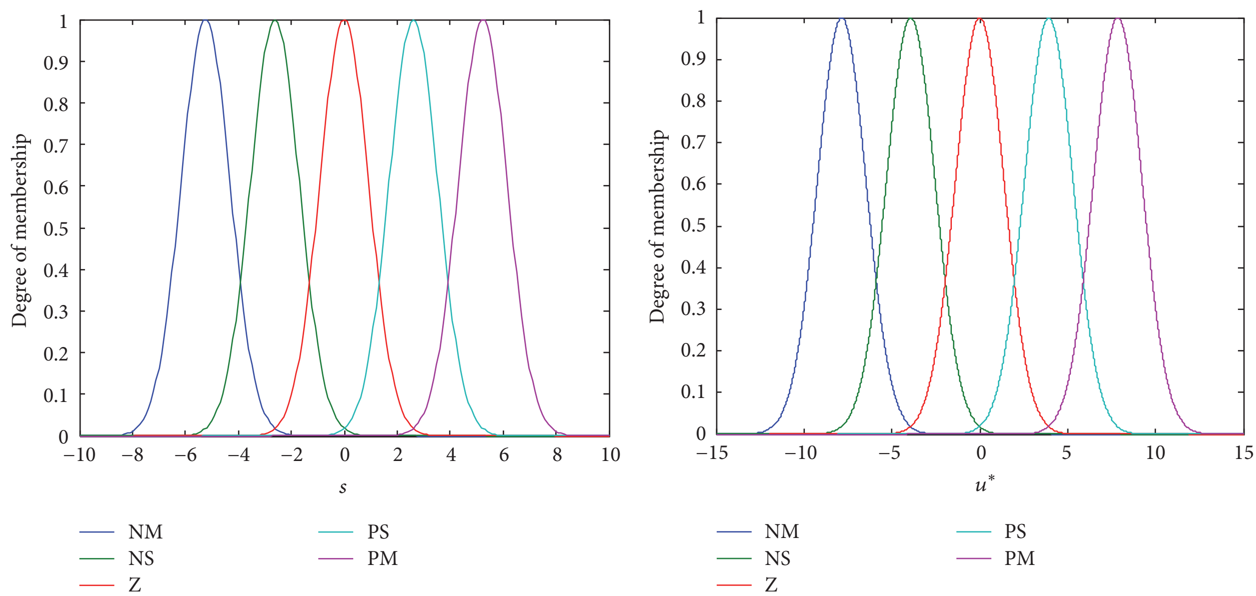
$$u(t) = \hat{\boldsymbol{\alpha}}^T \boldsymbol{\xi} - \hat{E}(t)\operatorname{sgn}(s(t)). \tag{37}$$

Aiming at the sliding mode surface, five kinds of membership functions for input $s(t)$ are used to fuzzy as follows and shown in Figure 7(a):

$$\mu_{\mathrm{NM}}(s) = \exp\left[-\left(\frac{(s+5\pi/3)}{(5\pi/12)}\right)^2\right],$$
$$\mu_{\mathrm{NS}}(s) = \exp\left[-\left(\frac{(s+5\pi/6)}{(5\pi/12)}\right)^2\right],$$
$$\mu_{\mathrm{Z}}(s) = \exp\left[-\left(\frac{s}{(5\pi/12)}\right)^2\right], \tag{38}$$
$$\mu_{\mathrm{PS}}(s) = \exp\left[-\left(\frac{(s-5\pi/6)}{(5\pi/12)}\right)^2\right],$$
$$\mu_{\mathrm{PM}}(s) = \exp\left[-\left(\frac{(s-5\pi/3)}{(5\pi/12)}\right)^2\right].$$

Five kinds of membership functions for output $u^*(t)$ are used to fuzzy as follows and shown in Figure 7(b):

$$\mu_{\mathrm{NM}}(u^*) = \exp\left[-\left(\frac{(s+15\pi/6)}{(15\pi/24)}\right)^2\right],$$
$$\mu_{\mathrm{NS}}(u^*) = \exp\left[-\left(\frac{(s+15\pi/12)}{(15\pi/24)}\right)^2\right],$$



(a) Input

(b) Output

FIGURE 7: The degree of membership.

$$\mu_Z(u^*) = \exp\left[-\left(\frac{s}{(15\pi/24)}\right)^2\right],$$
$$\mu_{PS}(u^*) = \exp\left[-\left(\frac{(s-15\pi/12)}{(15\pi/24)}\right)^2\right],$$
$$\mu_{PM}(u^*) = \exp\left[-\left(\frac{(s-15\pi/6)}{(15\pi/24)}\right)^2\right]. \quad (39)$$

The five rules are used to approach $u^*(t)$. The initial values of $\hat{\boldsymbol{\alpha}}$ and $\widehat{E}$ are 0.20. The parameters of controller set $\eta_1 = 200$ and $\eta_2 = 0.50$. The roll angle command is $\theta_d(t) = 0$, and $k_1 = 10$, $k_2 = 25$.

Then the sliding surface is

$$s(t) = \dot{\theta}(t) - \int_0^t \left[\ddot{\theta}_d(t) - 10\dot{e}(t) - 25e(t)\right] dt. \quad (40)$$

At present, the actual fin stabilizers generally use the traditional PID controller, which has the advantages of being intuitive, simple, and robust. However, the complex random waves, the varying motion parameters, and different courses can affect the roll. In practical engineering, PID parameters often need to be adjusted by experienced professionals. Here, the best parameters of the actual fin stabilizer are simulated and compared.

As a comparison, the PID controller is proposed. In the simulation, since the practicality of fin stabilizers is considered, the PID controller in actual application is adopted as follows:

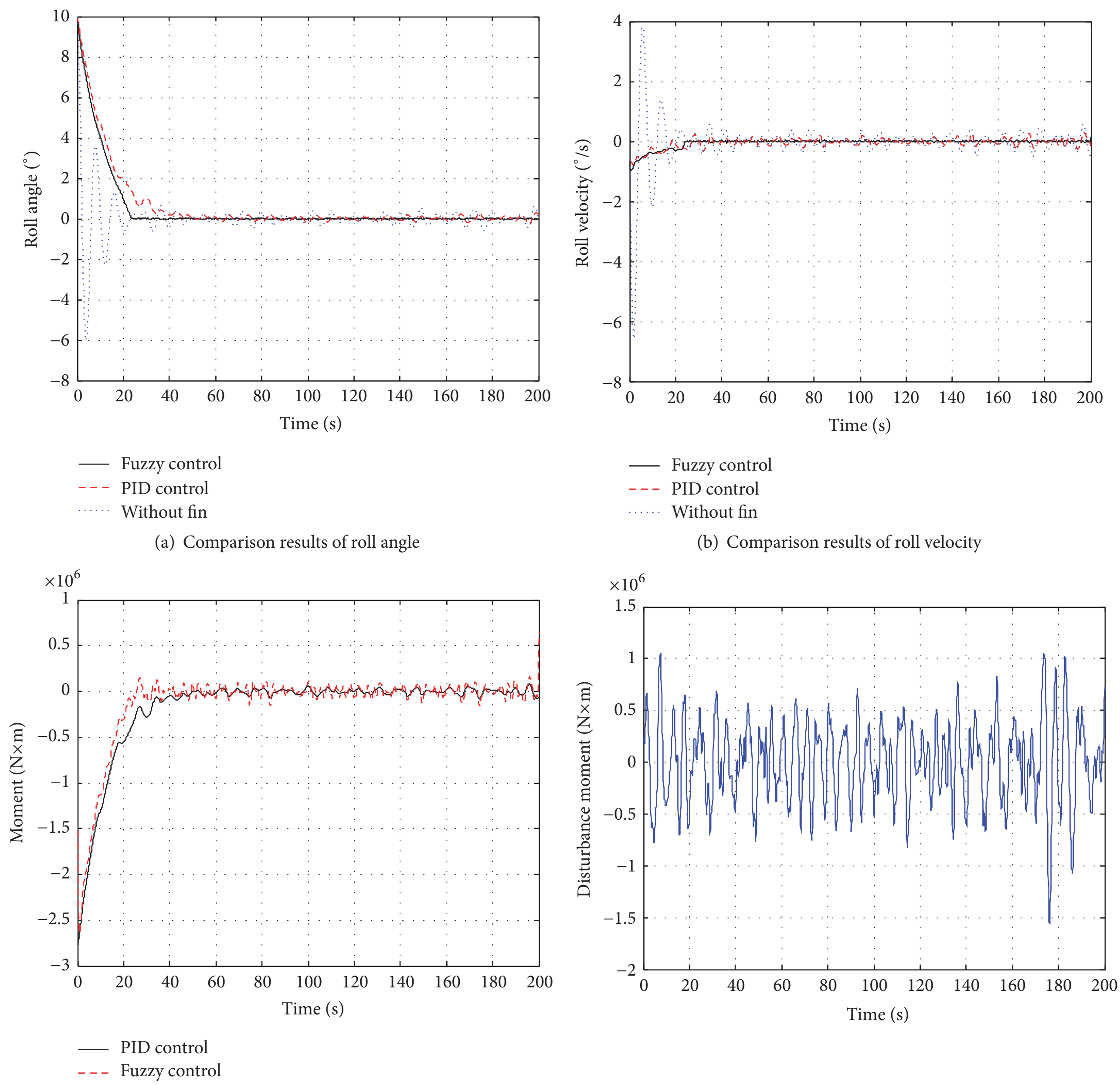
$$u_{PID}(s) = \left(k_I\frac{1}{T_I s+1} + k_D\frac{T_{D1}s}{(T_{D1}s+1)(T_{D2}s+1)} + k_p\right) \cdot \phi(s), \quad (41)$$

where k_p, k_I, and k_D are the adjustment coefficients of proportion, integral, and differential in the controller, respectively. In order to solve the integral drift, the integral link is approximated by the inertia link. T_I is time constant. In order to avoid the high frequency disturbance, the differential equation is replaced by the indirect differential link. T_{D1} and T_{D2} are the corresponding time constants. θ is the roll angle. Here $k_p = 6.89$, $k_I = 38.5$, $k_D = 2.05$, $T_I = 24.610$, $T_{D1} = 0.063$, and $T_{D2} = 0.17$.

4.3. Convergence Analysis. According to the system model and parameters of the controller in Section 4.1, the fuzzy input-based adaptive sliding mode controller and PID controller are applied to carry out simulation using MATLAB in IPC.

Assuming that there is an initial roll deviation angle of 10°, the simulation results of roll angle, roll rate, control moment, and the disturbance moment induced by waves can be obtained, as shown in Figure 8.

As can be seen from Figure 8, the fuzzy input-based adaptive sliding mode control can effectively reduce the



(a) Comparison results of roll angle

(b) Comparison results of roll velocity

(c) Comparison results of control moment

(d) Comparison results of disturbance moment

Figure 8: Experimental results of dynamic response.

roll motion. Compared with the PID controller, better performance of roll stabilization is achieved by adopting the adaptive sliding mode controller designed, in terms of the rate of convergence of roll angle. For roll angle, it is convergent at 25 s using the adaptive sliding mode controller. Comparatively, roll angle can just converge at 50 s using the PID controller. Similarly, the simulation results of roll rate and control moment show that the designed controller is better than the PID controller adopted in actual application for the convergence rate.

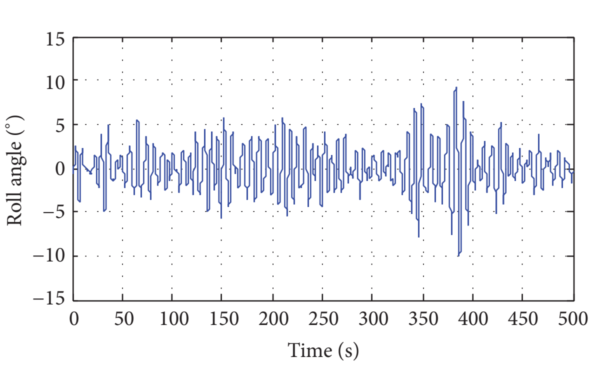
4.4. Simulation at Different Speeds and Wave Directions. In order to approach the practical engineering, the simulation is carried out in random waves. Here, the encounter angles are 45°, 90°, and 135°, respectively, and the speeds of the vessel are 10 Kn, 20 Kn, and 30 Kn, respectively.

The comparative results of the 20 Kn and the encounter angle 45°, for example, are as shown in Figure 9.

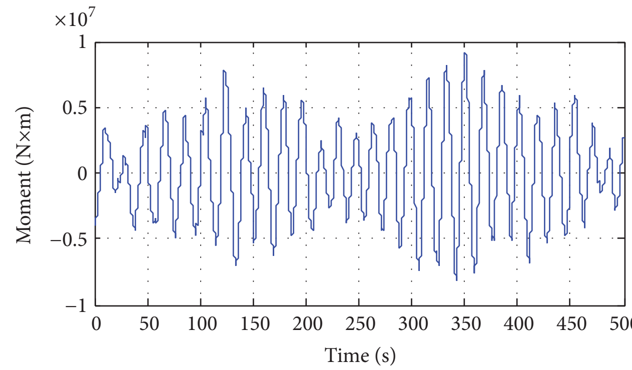
And the statistical results are shown in Table 3. The antirolling effect in Table 3 is calculated as follows:

$$E_{PE} = \frac{O_{AP} - C_{CS}}{O_{CS}} \times 100\%, \quad (42)$$

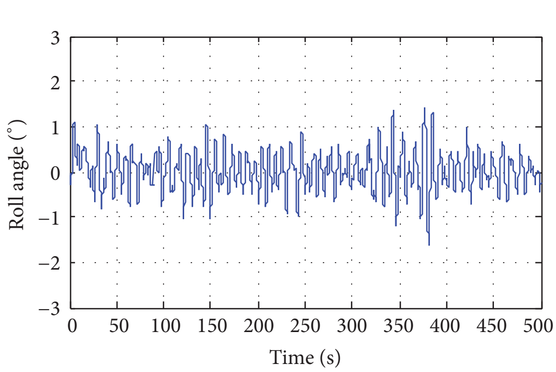
where E_{PE} is antirolling effect. O_{AP} is the variance of the roll angle without fin. O_{AP} and C_{CS} are the variances of the roll angle for two controllers.



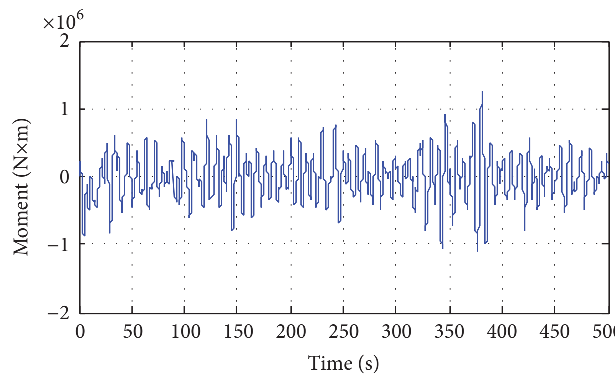
(a) Open-loop roll angle



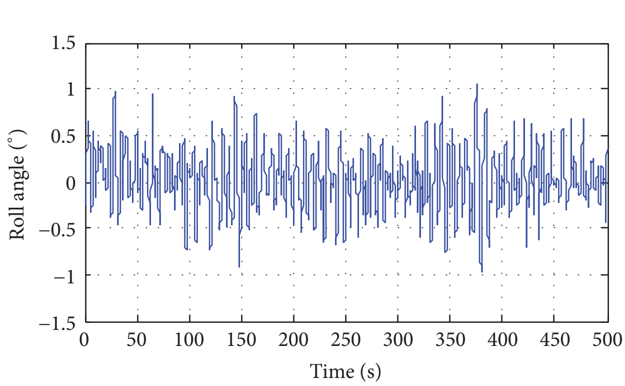
(b) Open-loop disturbance moment



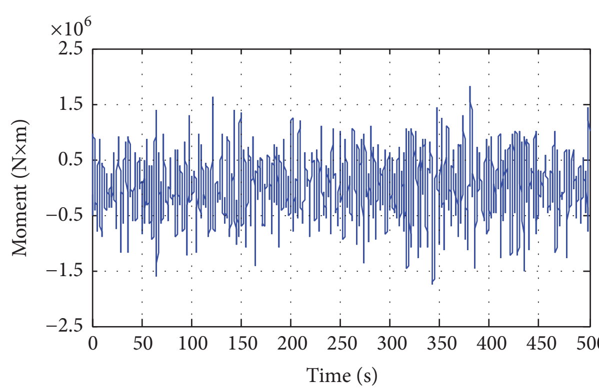
(c) Roll angle using PID controller



(d) Control moment using PID controller



(e) Roll angle using designed controller



(f) Control moment using designed controller

Figure 9: Comparative results of two controllers.

As can be seen from the statistical results in Table 3, the antirolling effect of fuzzy input-based adaptive sliding mode control is better than the PID control for the vessel at different speeds and wave directions. At the high speed, the antirolling effect reduction is better than that of low speed, and the effect of the front oblique wave is poor. When the vessel navigates at low speed and front oblique wave, the antirolling effect of PID controller becomes worse due to the strong nonlinearity and uncertainty of system. Compared with the conventional PID control, the fuzzy adaptive sliding mode control proposed in this paper has stronger robustness, which can be used to solve the nonlinear and uncertain effects on the system and improve the antirolling effect.

5. Conclusions

Rolling dynamics of the vessels is characterized by strong nonlinearities and uncertainties. The adaptive sliding mode control strategy provides an effective approach to reduce roll motion. In this paper, the research on fin stabilizers for vessel roll stabilization is proposed. The lift-feedback system of the fin stabilizers is designed, and a fuzzy input-based adaptive sliding mode control is obtained by employing Lyapunov function method. Experimental results illustrate the validity of the controller presented.

In this study, lift-feedback of the fin stabilizers is a key. In rough ocean environment, it is a difficult problem how

Table 3: Antirolling effect of two controllers.

Speeds (Kn)	Encounter angle (°)	Without fin		PID controller			Fuzzy input-based adaptive sliding mode controller		
		Mean (°)	Variance (°)	Mean (°)	Variance (°)	Roll reduction rate (%)	Mean (°)	Variance (°)	Roll reduction rate (%)
10	45°	9.0564	4.3714	1.7660	0.7429	80.5	1.4037	0.6671	84.5
	90°	13.7288	6.5789	1.4141	1.0475	89.7	1.2081	1.0322	91.2
	135°	7.9722	3.7722	0.9407	0.4513	88.2	0.7892	0.372	90.1
20	45°	6.1256	2.8846	0.9433	0.345	84.6	0.8515	0.3017	86.1
	90°	10.366	4.5874	0.7982	0.5475	92.3	0.6945	0.4337	93.3
	135°	5.7668	2.7427	0.4729	0.1999	91.8	0.3229	0.1537	94.4
30	45°	4.2979	1.991	0.3653	0.2015	91.5	0.3009	0.1487	93
	90°	8.3652	3.8709	0.5019	0.2341	94.0	0.2510	0.1895	97
	135°	4.0524	1.9457	0.2512	0.0845	93.8	0.1175	0.0654	97.1

to detect the actual lift force. However, specific detection methods have not been proposed. Further study in the respects mentioned will be continued in future work.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this article.

References

[1] T. Perez and G. C. Goodwin, "Constrained predictive control of ship fin stabilizers to prevent dynamic stall," *Control Engineering Practice*, vol. 16, no. 4, pp. 482–494, 2008.

[2] T. Perez, *Ship Motion Control: Course Keeping and Roll Stabilization Using Rudder and Fins*, Springer Publishing Company, Incorporated, 2010.

[3] M. A. Hinostroza, W. Luo, and C. Guedes Soares, "Robust fin control for ship roll stabilization based on L2-gain design," *Ocean Engineering*, vol. 94, pp. 126–131, 2015.

[4] L. Liang, M. Sun, S. Zhang, and Y. Wen, "A integrate control system design of WPC with active T-foil and transom stern flap for vertical motion improvement," *Journal of Computational Information Systems*, vol. 11, no. 9, pp. 3217–3227, 2015.

[5] J. Song, L. Liang, S. Zhang, and Z. Qi, "Design of inverse controller of zero-speed fin stabilizer based on RBF neural network," in *Proceedings of the 33rd Chinese Control Conference, (CCC '14)*, pp. 2111–2115, China, July 2014.

[6] Z. Liu, H. Jin, M. J. Grimble, and R. Katebi, "Ship forward speed loss minimization using nonlinear course keeping and roll motion controllers," *Ocean Engineering*, vol. 113, pp. 201–207, 2016.

[7] Y. Ling, H. Jin, and L. Ling, "Fuzzy-PID controlled lift feedback fin stabilizer," *Journal of Marine Science Application*, vol. 7, no. 2, pp. 127–134, 2008.

[8] L. Liang, M. Sun, S. Zhang, Y. Wen, P. Zhao, and J. Yuan, "Control system design of anti-rolling tank swing bench using BP neural network PID based on LabVIEW," *International Journal of Smart Home*, vol. 9, no. 6, pp. 1–10, 2015.

[9] J. C. Yin, Z. J. Zou, F. Xu, and N. N. Wang, "Online ship roll motion prediction based on grey sequential extreme learning machine," *Neurocomputing*, vol. 129, pp. 168–174, 2014.

[10] M.-C. Fang and J.-H. Luo, "On the track keeping and roll reduction of the ship in random waves using different sliding mode controllers," *Ocean Engineering*, vol. 34, pp. 479–488, 2007.

[11] L. Liang, M. Sun, H. Shi, T. Luan, and J. Xu, "Design and analyze a new measuring lift device for fin stabilizers using stiffness matrix of euler-bernoulli beam," *PLOS ONE*, vol. 12, no. 1, pp. 1–13, 2017.

[12] M.-C. Fang, Y.-Z. Zhuo, and Z.-Y. Lee, "The application of the self-tuning neural network PID controller on the ship roll reduction in random waves," *Ocean Engineering*, vol. 37, no. 7, pp. 529–538, 2010.

[13] S. Surendran, S. K. Lee, and S. Y. Kim, "Studies on an algorithm to control the roll motion using active fins," *Ocean Engineering*, vol. 34, pp. 542–551, 2007.

[14] P. Crossland, "The effect of roll-stabilisation controllers on warship operational performance," *Control Engineering Practice*, vol. 11, no. 4, pp. 423–431, 2003.

[15] M.-C. Fang, Y.-H. Lin, and B.-J. Wang, "Applying the PD controller on the roll reduction and track keeping for the ship advancing in waves," *Ocean Engineering*, vol. 54, pp. 13–25, 2012.

[16] G. Xia, C. Pang, and J. Xue, "Fuzzy neural network-based robust adaptive control for dynamic positioning of underwater vehicles with input dead-zone," *Journal of Intelligent & Fuzzy Systems*, vol. 29, no. 6, pp. 2585–2595, 2015.

[17] N. Wang and X.-Y. Meng, "Fuzzy modeling and control of ship-lift-feedback fin stabilizer system," *Journal of Harbin Engineering University*, vol. 28, pp. 22–26, 2007.

[18] Z.-H. Xiu and G. Ren, "Fuzzy controller design and stability analysis for ship's lift-feedback-fin stabilizer," in *Proceedings of 2003 IEEE International Conference on Intelligent Transportation Systems, (ITSC '03)*, pp. 1692–1697, China, October 2003.

[19] L.-X. Wang, "Fuzzy systems are universal approximators," in *Proceedings of the IEEE International Conference on Fuzzy Systems*, pp. 1163–1170, 1992.

[20] B. Yoo and W. Ham, "Adaptive control of robot manipulators using fuzzy compensator," in *International Conference on Intelligent Robots and Systems*, vol. 8, pp. 186–199, 2000.

[21] Y. Yang, C. Zhou, and X. Jia, "Robust adaptive fuzzy control and its application to ship roll stabilization," *Information Sciences*, vol. 142, pp. 177–194, 2002.

[22] R. Li, T. Li, W. Bai, and X. Du, "An adaptive neural network approach for ship roll stabilization via fin control," *Neurocomputing*, vol. 173, pp. 953–957, 2016.

[23] F. Alarçïn, "Internal model control using neural network for ship roll stabilization," *Journal of Marine Science & Technology*, vol. 15, no. 2, pp. 141–147, 2007.

[24] S. Lee, K.-P. Rhee, and J.-W. Choi, "Design of the roll stabilization controller, using fin stabilizers and pod propellers," *Applied Ocean Research*, vol. 33, no. 4, pp. 229–239, 2011.

[25] W. Luo, W. Lv, and Z. Zou, "Robust fin control for ship roll stabilization by using functional-link neural networks," in *Proceedings of International Conference on Advances in Neural Networks*, pp. 228–237, 2013.

[26] M. Moradi and H. Malekizade, "Robust adaptive first-second-order sliding mode control to stabilize the uncertain fin-roll dynamic," *Ocean Engineering*, vol. 69, pp. 18–23, 2013.

[27] T. Perez and M. Blanke, "Ship roll damping control," *Annual Reviews in Control*, vol. 36, no. 1, pp. 129–147, 2012.

[28] G. Q. Xia, T. T. Luan, M. X. Sun, and Y. W. Liu, "Dynamic analysis of catapult availability based on CBM," *International Journal of Hybrid Information Technology*, vol. 9, no. 10, pp. 31–42, 2016.

[29] G. Q. Xia, T. T. Luan, and M. X. Sun, "Research on Uncertain and Nonlinear Dynamic Weapons Dispatching System Using Fuzzy Robust Control," in *International Conference on Natural Computation, Fuzzy Systems and Knowledge Discovery*, pp. 447–452, IEEE, Changsha, China, 2016.

[30] L. H. Liang, M. X. Sun, and T. T. Luan, "Design sliding mode control of fuzzy switching gain for lift-feedback fin stabilizers," in *Proceedings of IEEE International Conference on Mechatronics and Automation*, pp. 1412–1417, Harbin, Heilongjiang, China, August 2016.