FUZZY SLIDING MODE CONTROL FOR SHIP ROLL STABILIZATION

Shyh-Leh Chen and Wei-Chih Hsu

ABSTRACT

A fuzzy sliding mode controller is proposed in this study for ship roll stabilization. Ship dynamic models usually contain large uncertainty. Sliding mode control is well known for its good robustness to large uncertainty. However, the required uncertainty bound is usually difficult to estimate. A fuzzy logic is designed here for the upper bound estimation of the uncertainty coming mostly from the wave excitation. As a result, the uncertainty-related parameters in the sliding mode controller are automatically tuned by fuzzy logic according to the encountered wave amplitude. The present controller has the advantage that smaller control efforts are required for the anti-capsizing purpose under the same sea states. A numerical example is investigated to confirm the analysis.

KeyWords: Fuzzy logic, sliding mode control, ship stabilization.

I. INTRODUCTION

Rolling is an undesirable motion for most vessels. Large amplitude ship rolling can easily lead to capsizing which will cause the loss of life and property. Controlling or reducing rolling motions is thus an important problem. There exist several active anti-roll devices, including fin stabilizers [2,4], gyroscopes [26], rudder-roll stabilizers [28], moving weights [17], and activated tanks [2,10,17]. With these devices, many control algorithms have been realized and implemented, ranging from classical methods, such as PID [21] and optimal control [10], to modern approaches, such as adaptive control based on gain scheduling [27], neural networks control [9], and fuzzy logic control [13]. However, most studies assumed small amplitude motions, which allowed them to use a linear ship model and to consider wave forcing as a small disturbance. Hence the results can not be applied to large amplitude motions like anti-capsizing control.

It has been shown that ship rolling dynamics are strongly nonlinear under large wave excitation [6,14,22, 27]. In such cases, conventional linear control cannot prevent a ship from capsizing, and nonlinear control methods must be employed. A controlling chaos technique was proposed in [8] as an anti-capsizing controller. However, the effectiveness of the controller depends strongly on the precise prediction of the wave excitation force, which is almost impossible in practice.

One distinct feature of the ship rolling dynamics is its potentially large uncertainty. The uncertainty comes mainly from the less understood hydrodynamic effects. The wave motions provide not only external excitation, but also parametric excitation [6,27]. Such a nonlinear control problem with possibly large uncertainty is well suited for the method of sliding mode control [15]. Chen et al. [5] designed a robust anti-capsizing controller using sliding mode control. The controller was proved to be able to survive bad sea states, e.g. 10m of wave height.

Although the sliding mode controller can allow for large uncertainty, the upper bound on the uncertainty is required. It is in general very difficult, if not impossible, to estimate the uncertainty bound. A common practice is by trial and error, and high gains are usually chosen for the uncertainty-related control parameters. As a result, excessive control efforts are often applied in small waves where the uncertainty is small. In other words, the robustness is achieved at the expense of control efforts. This may easily lead to the saturation of actuators and wasting of control energy. Energy is very limited for most ships in the open sea. To circumvent the problem, good estimate on the uncertainty bound is necessary so that the control gains can be tuned accordingly. A proper tool for the estimation is fuzzy logic since we do not even know the function form for the uncertainty in many systems (e.g., the ship dynamics).

The objective of this paper is to design a fuzzy
sliding mode controller for anti-capsizing purposes. The approach of [5] will be followed and generalized. In particular, the uncertainty-related parameters in the sliding mode controller will not be constant. Instead, depending on the wave amplitudes, they are determined by some fuzzy logic rules. The inclusion of fuzzy logic allows the controller to incorporate with the sailor’s experiences. It will be shown that the proposed controller can work in regular or irregular waves\(^1\) with more reasonable control efforts.

The idea of integrating sliding mode control and fuzzy logics is not new. Roughly speaking, there have been three types of fuzzy sliding mode control in the literature. The first type [e.g., 19,30] employs the fuzzy rules to approximate the equivalent control. The second type [e.g., 1,7,16,24,25,31] fuzzifies the sliding manifold to improve the chattering problem caused by the switching in the control input. The last type [e.g., 12,18] uses the fuzzy logic to estimate the uncertainty, which is exactly the same idea applied in the present work. These studies also incorporate adaptation laws to automatically update the associated membership functions or fuzzy rules for optimal estimation of the uncertainty. However, adaptive control is not suitable for the current system since the adaptation speed may not be able to catch up the fast-varying and irregular wave excitation.

The paper is organized as follows. After the introduction, a nondimensional, 1-DOF ship roll model with regular or irregular wave excitation is briefly described in Section II. The capsizing probabilities under different sea states are presented in Section III. In Section IV, the fuzzy sliding mode controller is designed, where the fuzzification of the wave amplitude will be based on the capsizing probability given in Section III. It is shown in Section V that the proposed controller can achieve the anti-capsizing objective. Simulation results and discussions are provided in Section VI, and conclusions are drawn in Section VII.

II. THE SHIP ROLL MODEL

A nondimensional, 1-DOF ship roll dynamics can be modeled as

\[ \dot{x}_1 = x_2 \]
\[ \dot{x}_2 = f_1(x_1) + f_2(x_1)x_2 + \varepsilon g(x_1, x_2, x_3, t) \]
\[ \dot{x}_3 = 0 \]

where \(x_1\) and \(x_2\) are roll angle and velocity, respec-

\(^1\) When the wave motion is a single harmonic function, it is called regular wave. Irregular wave is when the wave motion is a stochastic process.

For most ships the functions in equation (2) can be expressed by the approximate form

\[ f_1(x_1) = -x_1 + \alpha_1 x_1^2 \]
\[ f_2(x_1) = -1 + \alpha_2 x_1^2 \]
\[ g(x_1, x_2, x_3, t) = -\delta_1 x_2 - \delta_2 x_1 x_3 + h(t) \left[ f_1(x_1) + f_2(x_1)x_3 \right] + h_2(t) \]

The functions \(f_1\) and \(f_2\) represent hydrostatic and inertial contributions which are dominant roll moments. \(\delta_1 x_2 + \delta_2 x_1 x_3\) is the hydrodynamic damping, and the contribution from wave excitation is given by \(h(t)\) and \(h_2(t)\).

For vessels in regular sea, where the waves are assumed to be purely sinusoidal, \(h(t)\) and \(h_2(t)\) can be taken as [6]

\[ h(t) = \gamma_1 \cos \Omega t \]
\[ h_2(t) = \gamma_2 \sin \Omega t \]

where \(\Omega\) is the wave frequency and the \(\gamma_i\)’s depend on the wave amplitude. On the other hand, if the sea is irregular, i.e., the waves are random processes, then \(h(t)\) and \(h_2(t)\) can be approximated by [11]

\[ h(t) = \sum_{j=1}^{N} \gamma_{j1} \cos(\Omega_j t + \Psi_{j1}) \]
\[ h_2(t) = \sum_{j=1}^{N} \gamma_{j2} \sin(\Omega_j t + \Psi_{j2}) \]

where \(N\) is some positive integer, the \(\Omega_j\)’s are grid wave frequencies, the \(\Psi_{j1}\)’s are random variables uniformly distributed in \([0,2\pi]\), and the \(\gamma_{j1}\)’s depend on the spectral density function of the waves. It should be emphasized that this roll model has been extensively studied in the literature (see, e.g [5,6,11]). Hence we shall not repeat the detailed derivation, but concentrate on the controller design.

III. CAPSIZING PROBABILITY

In this section, the ship rolling dynamics will be reviewed as a preparation for the controller design. In particular, the capsizing probability under regular and irregular waves will be discussed. The unperturbed system, where \(\varepsilon = 0\) in equation (2), corresponds to the situation of calm water and no damping. In this case, the system
possesses 3 equilibrium states: a center at the origin representing the upright position, and two saddles representing the angles of vanishing stability, as shown in Fig. 1. The two saddles are connected by a heteroclinic cycle [27] whose interior is referred to as the safe region, denoted by \( S \). As one can see clearly from Fig. 1, the system will remain inside the safe region whenever the initial condition is in the region.

As the wave excitation is introduced, i.e. \( \varepsilon \neq 0 \), the heteroclinic cycle will break down and some initial conditions inside the safe region will generate unstable trajectories resulting in the capsizing of the ship [27]. Therefore, the capsizing probability can be defined as the percentage of the initial conditions in the safe region that will become unsafe after a prescribed time period when the wave excitation is present.

IV. DESIGN OF THE FUZZY SLIDING MODE CONTROLLER

For anti-capsizing purposes, a fuzzy sliding mode controller that employs activated tanks or moving weights will be designed in this section. The main goal of the anti-roll tanks and moving weights is to dynamically adjust the horizontal position of the vessel’s center of gravity to counteract the rolling motions. With such devices, \( x_3 \) in the ship model is no longer constant and equations (1)-(3) are modified as

\[
\dot{x}_1 = x_2 \\
\dot{x}_2 = f_2(x_1) + \Delta f(x_1, x_2, x_3, t) \\
\dot{x}_3 = u
\]

where \( u \) is the flow rate or mass speed which can be varied instantaneously by the actuator. In other words, the dynamics of the actuator are assumed to be fast enough and are negligible. Also, note that \( \Delta f(x_1, x_2, x_3, t) \) in equation (12) represents the uncertainty containing unmodeled dynamics and disturbances (e.g., wind loads) as well as the wave excitation and damping (i.e., \( \varepsilon\dot{g}(x_1, x_2, x_3, t) \)). The uncertainty function \( \Delta f(x_1, x_2, x_3, t) \) is assumed to be continuous in its arguments. It should be pointed out that compared to the well known hydrostatic forces, the hydrodynamic contributions are less understood. This is primarily due to the difficulties involved in solving the free-surface hydrodynamic problem encountered when computing the forces acting on a vessel. Therefore, ship model uncertainties always exist and can be substantial in magnitude.

The nonlinear system given by (11)-(13) with possibly very large uncertainty is well suited for the method of sliding mode control [15]. However, the uncertainty \( \Delta f(x_1, x_2, x_3, t) \) does not satisfy the matching condition and hence the backstepping process needs to be incorporated [15]. In other words, there will be two levels of sliding mode control. Taking

\[
\sigma_1 = x_2 + \beta x_1 = 0, \quad \beta > 0
\]

as a sliding manifold, the virtual control input \( x_3 \) is first designed to yield

\[
x_3 = \psi(x_1, x_2)
\]

\[
= -\frac{1}{f_2(x_1)} f_1(x_1) + \beta x_2 + \frac{\lambda_1 + \rho_2}{\tanh(1)} \left( \frac{\sigma_1}{\varepsilon_1} \right)
\]

(14)

Next,

\[
\sigma_2 = x_3 - \psi(x_1, x_2) = 0
\]

is taken as the other sliding manifold to design the real control input \( u \), yielding

\[
u = \frac{\partial \psi}{\partial x_1} x_2 + \frac{\partial \psi}{\partial x_2} f_2(x_1) + \frac{\partial \psi}{\partial x_3} \Delta f(x_1, x_2, x_3, t) \left( \frac{\lambda_2 + \rho_3}{\tanh(1)} \right) \left( \frac{\sigma_2}{\varepsilon_2} \right)
\]

(15)

In equations (14) and (15), \( \lambda_1, \lambda_2, \varepsilon_1 \), and \( \varepsilon_2 \) are positive parameters. On the other hand, \( \rho_1 \geq 0 \) and \( \rho_2 \geq 0 \) are estimates on the upper bounds of the uncertainties:

\[
|| \Delta f(x_1, x_2, x_3, t) || \leq \rho_1 \quad \text{and} \quad || \Delta \psi(x_1, x_2, x_3, t) || \leq \rho_2
\]

(16)

where the function norm \( || \cdot || \) is taken over \( t \in [0, \infty) \) and a compact domain of interest \( D \) (in the state space) to be defined in Section V. It is hoped that \( \rho_{1u} = \rho_{1} \) and \( \rho_{2u} = \rho_{2} \).

One common problem in using the sliding mode controller is that it is very difficult, if not impossible, to estimate the uncertainties, i.e., to estimate \( \rho_1 \) and \( \rho_2 \) in (16). Large values of \( \rho_1 \) and \( \rho_2 \) are usually applied to cover the possibly worst uncertainty. However, it may easily lead to saturation of the actuators. Moreover, in cases where the uncertainty is small, smaller values for \( \rho_1 \) and \( \rho_2 \) can be used to save control effort. Although \( \rho_1 \) and \( \rho_2 \) could be taken as functions of the states and time, some systems (like the present one) does not even

Fig. 1. The roll dynamics in calm water.
possess the function form for the uncertainty. For the present system, the uncertainty comes mostly from the less understood hydrodynamic effects caused by the sea wave excitation. Hence, \( \rho_1 \) and \( \rho_2 \) can be considered as functions of the wave amplitude only. When the sea state is nice or moderate, the system uncertainty will not be so large that \( \rho_1 e \) and \( \rho_2 e \) can be tuned smaller to save control energy. Energy is crucial to most ships.

To the aim, we shall modify the control law (14) and (15) to that shown in Fig. 2. Instead of using fixed and large values of \( \rho_1 e \) and \( \rho_2 e \), they will now be determined by the sea states. When the wave amplitudes (or significant wave amplitudes in the case of random seas) are large, \( \rho_1 e \) and \( \rho_2 e \) are taken to be large. On the contrary, small values of \( \rho_1 e \) and \( \rho_2 e \) are taken for small wave amplitudes. Here the concept of "large" and "small" will be realized by fuzzy logic. The fuzzification of the wave amplitude will be based on the capsizing probability defined in the previous section. For example, the wave amplitude is considered "large" when the corresponding capsizing probability is greater than 0.5. On the other hand, \( \rho_1 e \) and \( \rho_2 e \) will be fuzzified by trial and error through numerical simulations. This trial and error process may include the sailor’s experience in reality.

If the fuzzy logics are properly designed, one should have that for any sea state

\[
|\rho_1 - \rho_1 e| \leq e_1 \\
|\rho_2 - \rho_2 e| \leq e_2
\]  

where \( e_1 \) and \( e_2 \) are small positive constants. This is due to the fact that fuzzy logics can be served as universal approximators [3,29].

V. STABILITY ANALYSIS

In this section, it is desired to show that the fuzzy sliding mode controller proposed in the previous section can indeed achieve the anti-capsizing objective. In other words, for any initial condition in the safe region, all state variables are bounded for all forward time. Moreover, the roll angle and velocity will approach a small neighborhood of the origin at the steady state. Let

\[
V_1 = \frac{1}{2} x_1^2 + \frac{1}{2} \sigma_1^2 = \frac{1}{2} x_1^2 + \frac{1}{2} (\beta x_1 + x_2)^2
\]

and let \( \Omega = \{ |V_1| \leq c_1 \} \), where \( c > 0 \) be such that \( S \subset \Omega \). Define the admissible set for initial states by

\[
I = \{ x \in \mathbb{R}^3 | (x_1, x_2) \in S, |x_2| \leq L \}
\]

and the compact domain of interest by

\[
D = \{ x \in \mathbb{R}^3 | (x_1, x_2) \in \Omega, |x_2| \leq L \}
\]

where \( L \) represents an upper bound on \( x_2 \), which depends on the allowable range for the center of gravity \( x_3 \). Due to practical geometric constraints, there usually exists a maximum bound for \( x_3 \). \( L \) is chosen such that \( \forall x \in D, x_3 \) is within the maximum bound. It is now ready to state and prove the main result.

Theorem. Consider the 1-DOF ship rolling dynamics (11)-(13), with the sliding mode controller given by (14) and (15), where the uncertainty-related parameters \( \rho_1 e \) and \( \rho_2 e \) are tuned by fuzzy logics according to sea wave amplitude. Suppose that the fuzzy logics are properly designed so that \( \rho_1 e \) and \( \rho_2 e \) well approximate the actual uncertainty bounds \( \rho_1 \) and \( \rho_2 \) in the sense of (17) and (18). Then, for \( \lambda_1 \) and \( \lambda_2 \) sufficiently large and for \( \epsilon_1 \) and \( \epsilon_2 \) sufficiently small, the closed-loop rolling dynamics will satisfy that for any initial states in the admissible set, i.e., \( \forall x(0) \in I \),

(i) \( |x| \) is bounded \( \forall t \geq 0 \),

(ii) \( x_1, x_2 \) are ultimately bounded with the bound depending on \( \epsilon_1 \).

Proof. The first step is to show that

\[
\dot{V}_1 < 0 \quad \forall x \in \{ \epsilon^2 \leq V_1 \leq c \} \times \{ |x_2| \leq L \}
\]

where \( \epsilon \geq \epsilon_1 \). To this aim, we compute

\[
\dot{V}_1 = -\beta x_1^2 + x_1 \sigma_1 - (\lambda_1 + \rho_2) \sigma_1 \frac{\tanh(\sigma_1 / \epsilon_1)}{\tanh(1)}
\]

\[
+ f_2(x_1) \sigma_2 + \sigma_2 \Delta f
\]

Assume that within the domain of interest \( D \)

\[
|f_2(x_1) \sigma_2| \leq l, \quad l > 0
\]

Note that \( \forall \xi \in \mathbb{R} \).
Case 2. \( |\sigma_1| \geq \epsilon_2 \). If \( \lambda_1 > \sqrt{2c} + l + \epsilon_2 \), we can obtain
\[
\dot{V}_1 \leq -\beta x_1^2 + |\sigma_1| \|x_1\| + l + \epsilon_2 - \lambda_1 < 0
\]
Note here that \( |x_1| \geq \sqrt{2c}, \forall (x_1, x_2) \in \{V'_1 \leq \epsilon_1\} \).

Case 2. \( |\sigma_1| < \epsilon_2 \) and \( |x_1| \geq \epsilon \). Then
\[
\dot{V}_1 \leq \left[ \frac{\beta}{2} x_1^2 + |\sigma_1| \left| \frac{-\lambda_1 + \rho_{\omega}}{2\epsilon} \right^2 \right] - \frac{\beta}{2} \epsilon^2
\]
\[+ \left[ -\frac{\lambda_1 + \rho_{\omega}}{2\epsilon} \sigma_1^2 + \left( l + \rho_{\omega} + \epsilon_2 \right) |\sigma_1| \right]
\]
Note that if \( a > 0 \) and \( b > 0 \), then \( \forall z \in \mathbb{R} \),
\[-\alpha z^2 + bz \leq \frac{b^2}{4a}\]
Hence, \( \dot{V}_1 < 0 \) provided that
\[
\lambda_1 > \frac{\epsilon_2}{\beta} - \rho_{\omega}, \text{ and } \lambda_2 > \frac{\epsilon_2 (l + \rho_{\omega} + \epsilon_2)^2}{2\epsilon^2} - \rho_{\omega}
\]
From the two cases, it is concluded that
\[
\lambda_1 > \max \left\{ \sqrt{2c} + l + \epsilon, \frac{\epsilon_2}{\beta} - \rho_{\omega}, \frac{\epsilon_2 (l + \rho_{\omega} + \epsilon_2)^2}{2\epsilon^2} - \rho_{\omega} \right\}
\]
then (19) will be true.

Next, let \( V_2 = \frac{1}{2} \sigma_2^2 \). We shall show that
\[
\dot{V}_2 \leq 0 \quad \forall x \in \{V'_1 \leq \epsilon_1\} \times \{\sigma_2 \geq L\}
\]  
(20)
Since the derivative of \( V_2 \)
\[
\dot{V}_2 = - \left( \lambda_2 + \rho_{\omega} \right) \frac{\sigma_2 \text{tanh}(\sigma_2, \epsilon_2)}{\text{tanh}(\lambda_1)} - \frac{\sigma_2}{\epsilon_2} \frac{\partial y}{\partial x_2} \Delta y
\]
still includes the function \( \text{tanh} (\bullet) \), again two cases are considered.

Case 1. \( L \geq \epsilon_2 \). If \( \lambda_2 \geq \epsilon_2 \), we have
\[
\dot{V}_2 \mid_{\sigma_2 \geq L} \leq - \left( \lambda_2 + \rho_{\omega} \right) L + (\rho_{\omega} + \epsilon_2) L \leq 0
\]

Case 2. \( L < \epsilon_2 \). If \( \lambda_2 \geq (\rho_{\omega} + \epsilon_2) \frac{\epsilon_2}{L} - \rho_{\omega} \), we can get
\[
\dot{V}_2 \mid_{\sigma_2 < L} \leq - \left( \lambda_2 + \rho_{\omega} \right) L + (\rho_{\omega} + \epsilon_2) L \leq 0
\]
Therefore, if \( \lambda_2 \geq \max \left\{ \epsilon_2, \frac{\epsilon_2 (\rho_{\omega} + \epsilon_2)^2}{L^2} - \rho_{\omega} \right\} \), then (20) is true. The inequalities (19) and (20) imply that the domain of interest \( D \) is positively invariant and \( (x_1, x_2) \) will approach the set \( \{V'_1 \leq \epsilon_1\} \) at the steady state. The ultimate bound for \( (x_1, x_2) \) depends on \( \epsilon \) and hence depends on \( \epsilon_1 \). Therefore, \( \forall \sigma(0) \in I \subset D \), the goals (i) and (ii) can be achieved provided that \( \lambda_1 \) and \( \lambda_2 \) are sufficiently large and \( \epsilon_1 \) and \( \epsilon_2 \) are sufficiently small.

VI. NUMERICAL SIMULATIONS AND DISCUSSIONS

To examine the performance of the controller, a specific vessel, the twice-capsized clam dredge Patti-B [23], is investigated in this section. The system parameters for the Patti-B can be found in [6]. We will emphasize on the comparison of open loop system, the closed loop system with sliding mode controller, and that with fuzzy sliding mode controller.

Without any controller, a vessel is susceptible to the danger of capsizing under severe sea states. The capsizing probabilities for Patti-B under regular and irregular seas can be obtained by extensive numerical simulations [11], and are demonstrated in Fig. 3. The regular waves have a fixed frequency at 0.6 rad/s, with amplitudes varying from 0 to 5 m. For irregular waves, the spectral density function of North Atlantic Ocean is adopted here [10]:
\[
S(\omega) = 0.31H^2 \frac{\omega^4}{\omega_0^4} \exp[-1.25(\frac{\omega}{\omega_0})^4]
\]  
(21)
where \( \omega_0 \) is the characteristic wave frequency and is fixed at 0.6 rad/s and \( H \), is the significant wave height varying from 0 to 5 m. For corresponding to wave amplitudes of 0 to 5 m.

According to the capsizing probability in Fig. 3, the wave amplitudes are classified into seven linguistic states, namely VS (very small), MS (mid-small), S (small), M (middle), L (large), ML (mid-large), and VL (very large), respectively. The resulting membership function is given through a trial-and-error process. The min-max approach is used for fuzzy inference, and the centroid method is used for defuzzification [20]. In the current system, the input of the fuzzy logic is the wave amplitude and the outputs are \( \rho_{\omega} \) and \( \rho_{\omega_0} \). All inputs and outputs have 7
linguistic states, and we have 7 rules with similar statement:

If the wave amplitude is $X$, then $\rho_{1e}$ and $\rho_{2e}$ are $X$.

Here $X$ stands for one of the 7 linguistic states. Such rules imply that high gains of $\rho_{1e}$ and $\rho_{2e}$ are used only in bad seas.

In the following simulations, the performance of 3 different controllers in both regular and irregular seas is investigated. The 3 controllers discussed are: uncontrolled, sliding mode controller (SMC), and fuzzy sliding mode controller (FSMC). The design parameters for SMC are:

$$\beta = 0.1, \quad \varepsilon_1 = 0.3, \quad \varepsilon_2 = 0.01, \quad \lambda_1 = 0.005, \quad \lambda_2 = 0.01,$$

$$k = 0.5, \quad \rho_{1e} = 0.32, \quad \rho_{2e} = 0.42$$

whereas those for FSMC are the same except for $\rho_{1e}$ and $\rho_{2e}$ which are given by the fuzzy rules described earlier. The initial condition for the simulation is taken as $x(0) = (0, 0, 0)$.

The state trajectories for the 3 cases with wave amplitude of 1 m and wave frequency of 0.6 rad/s are shown in Fig. 7. As one can see from the figure, without control the ship will eventually capsize. Both SMC and FSMC can stabilize the system. The advantage of FSMC over SMC can be seen clearly from the associated control efforts, which are depicted in Fig. 8. The peak control effort needed for FSMC is only about half of that for SMC. The discrepancy could be larger at different wave amplitudes. Therefore, FSMC is easier to realize and can save control energy.

Next, the irregular wave with spectral density given by (21), significant wave height of 2 m, and characteristic wave frequency of 0.6 rad/s is considered. Fig. 9 shows the corresponding state trajectories, and the associated control efforts are plotted in Fig. 10. Again, although both SMC and FSMC can prevent capsizing, the peak control effort for SMC is more than 3 times that for FSMC.

VII. CONCLUSIONS

In this study, we have designed an anti-capsizing fuzzy sliding mode controller for vessels in regular or irregular seas. The controller is modified from a sliding mode controller proposed in a previous work. The design of conventional sliding mode controller is usually conservative, where fixed but large values of uncertainty bounds are used even in a nice sea state. Quite often, excessive control efforts are applied. The proposed controller takes into account the wave dependence of the uncertainties. The key feature is to incorporate fuzzy logics for determining the uncertainty bounds according to wave amplitudes. In this way, the goal of anti-capsizing
can still be achieved, and at the same time, smaller control efforts are needed. The results are verified by numerical simulations for a fishing vessel, the clam dredge Patti-B.

REFERENCES


10. Fossen, T.I., Guidance and Control of Ocean Vehi-

**Shyh-Leh Chen** received B.S and M.S. degrees from National Tsing-Hua University, Hsin-Chu, Taiwan, in 1987 and 1989, respectively, both in power mechanical engineering. He received a Ph.D. degree in mechanical engineering from Michigan State University in 1996.

Since 1996, he has been with National Chung-Cheng University, Chia-Yi, Taiwan, where he is currently an Associate Professor in the Department of Mechanical Engineering. His research interests include nonlinear dynamics and control, wavelet analysis, with application to contouring control of multi-axis systems, active magnetic bearing, and ship stabilization.

**Wei-Chi Hsu** received B.S in aeronautical engineering from Tamkang University, Taiwan, in 1996, and M.S. in mechanical engineering from National Chung-Cheng University, Taiwan, in 1998. Currently, he is a servo engineer for an industrial company in Hsin-Chu. His research interests include ship stabilization and automation.