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A Novel Control System of Ship Fin Stabilizer Using Force Sensor to Measure Dynamic Lift

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ABSTRACT For improving the control system of traditional ship fin stabilizers, this paper presents a novel control system to solve the deviation of system feedback by using dynamic lift. However, the difficulty of lift feedback control is lift measurement technology. Thus, the lift sensor is designed using Spoke-type structure and Wheatstone bridge-type strain conversion circuit. The design of compensation resistors is adopted in the circuit, which effectively reduces the effect of temperature on zero drift and sensitivity for the lift sensor. In addition, the mechanical decoupling method of fin hydrodynamic force and the lift measurement method based on double bearing load are proposed. In addition, then, the control system of ship fin stabilizer is improved in order to solve the main technical problems. Finally, the experimental results demonstrate that the designed sensors meet the requirements of lift measurement. In addition, the simulation results of the ship at different situations show that the effect of roll stabilization (86.803%–93.858%) is improved effectively.

INDEX TERMS Ship fin stabilizer, roll stabilization, lift feedback control system, force sensor, dynamic lift measurement.

I. INTRODUCTION

Since a ship is always sailing at sea, the severe wave disturbances will affect the ship motions, especially on the roll motion. It can seriously affect the ship stability and seakeeping performance. Ship roll stabilization is very important for the human comfort, the crew work efficiency, cargo safety, and the operation of certain equipment on board. Thus, many stabilization equipments with their associated control systems have been invented to reduce roll motion [1], [2].

During the last decades, several devices for ship roll reduction include bilge keels, anti-rolling tanks, gyroscopic stabilizers, rudder-roll stabilization, moving weights, and active fins. Among the special auxiliary equipments, the fin stabilizer is the most attractive device because of its feasibility and high effectiveness. Especially for the ship at high speed, the fin stabilizer can provide considerable damping to accomplish roll reduction [3], [4].

For roll stabilization performance of the fin stabilizer, its control system and hydrodynamic application of fin are the key factors. When shape structure of the fin is determined, appropriate control system and control method can play a very important role in hydrodynamic characteristics. Under certain circumstances, the theoretical anti-rolling effect of software simulation can reach about 90% [5]. But the actual effect is far behind the expected indicators since many interference factors make system deviation too large. There are two main reasons for this deviation. Firstly, the conventional fin stabilizer estimates lift using fin angle measurement as system feedback. However, the system feedback of fin stabilizer should be the actual dynamic lift. Therefore, the accuracy of the system feedback is not high. That is, the measurement of dynamic lift is not accurate. Secondly, the structure of the control system needs to be improved and the controller is not suitable for the fin stabilizer [6], [7]. Thus, the accuracy of the actual measurement and nonlinearity of the system should be considered especially in practical engineering application.

The control moment of the fin stabilizer actually needs the dynamic lift obtained from the hydrodynamic force. The control moment is the moment preventing ship rolling. However, the marine environment is very complex. In addition, the sensor mounted on the fin should be considered in practical projects. So it is very difficult to measure the dynamic lift directly. For the conventional fin stabilizer, fin angle feedback is generally adopted [8]. Fin angle is easy to measure, and fin angle sensor is located inside the hull, which is convenient to install. The lift is not measured directly in the fin angle feedback, and it is indirectly calculated on the basis

TABLE 1. Comparison of different lift measurement scheme.

Attribute	Scheme 1	Scheme 2	Scheme 3	Scheme 4
Characteristic	contact displacement	cross shaft	pressure difference	non-contact displacement
Force decoupling	able	able	unable	able
Change of original device	biggest	big	little	bigger
Installation of the sensor	difficult	difficult	convenient	convenient
Maintenance of the sensor	difficult	difficult	convenient	convenient
Accuracy	accurate	accurate	inaccurate	accurate
Generality	universal	special	universal	universal

of fin angle. The theoretical value of lift is approximately calculated based on an ideal linear relationship. In fact, the dynamic lift is much more complicated than that of the corresponding static. In addition, the relationship is seriously nonlinear and uncertain between fin angle and actual lift due to many disturbances [9]. There is a big deviation between theoretical value and actual value for lift. So the system feedback is inaccurate. Therefore, the direct measurement of the actual dynamic lift is the key factor to solve the system deviation.

In recent decades, although the lift feedback concept of fin stabilizers has been proposed, it has not been popularized and applied. The fundamental reason is that it is more difficult to measure lift than to measure fin angle. Due to the problems of sensor type, sensor installation, maintenance and decoupling of force, lift measurement has become a popular research topic in the field of ship motion control. Whether it is theoretical research or engineering application, lift measurement has good research value. The advantages and disadvantages of lift measurement schemes are analyzed and compared for different research institutions in the world. A novel lift measurement scheme is designed on the basis of this analysis.

(1) Lift measurement scheme based on sensor mounted in hollow shaft.

American Sperry-Marine Company [10] designed a novel fin shaft structure. The contact displacement sensor was mounted in the hollow shaft. And the lift of fin was translated by measuring the deformation of fin shaft. However, the stiffness of fin shaft was very large, so the strain was very small and difficult to measure. And the different diameters and materials directly affected the deformation of fin shaft. For different devices, the coefficients were different, so it was not universal. What is more difficult is that if the sensor was damaged, it was difficult to repair on board timely.

(2) Lift measurement scheme based on force sensor of cross shaft.

British Rolls-Royce Company [11] designed strain gauges installed in the cross shaft. The lift of the fin was acquired by measuring force of cross shaft. Because the deformations of different types of cross shaft were different, the method was not universal. At the same time, because the strain gauge was not easy to replace, the maintainability of the method was very low.

(3) Lift measurement scheme based on pressure sensor installed in hydraulic cylinder.

Japanese Mitsubishi Company [12] improved the hydraulic servo actuator of fin stabilizers. Measurement of lift was realized by pressure sensor installed in the hydraulic cylinder of driving fin. The moment of fin shaft could be measured by pressure sensor. If the fin pressure center was known, the lift on fin could be obtained. The scheme was simple in structure and easy to maintain. But there was an inescapable problem: the pressure center of the fin, which was related to fin type, angle, angular velocity and speed. In addition, it was highly nonlinear and could not be measured. Therefore, the accuracy of the method was not high, and the obtained lift values could only be used as reference variables in fin angle feedback control system.

(4) Lift measurement scheme based on sensors mounted on the outer end of fin shaft.

A novel hollow shaft with shaft core was designed by Ship stabilization and Control Research Institute of Harbin Engineering University [13]. Lift measurement was implemented by mounting non-contact displacement sensor at the end of fin shaft.

Stiffness matrix of Timoshenko beam was used to carry out theoretical analysis. It transformed lift into easily measurable displacement and established the quantitative relationship between them. But the disadvantage was due to the hollow shaft, as the shaft must be large in order to ensure the strength requirements, which took up valuable space inside the ship. At the same time, due to the large deflection of the shaft, the material fatigue and sealing requirements were higher. In addition, the processing was relatively difficult [14].

The advantages and disadvantages of the above lift measurement scheme were compared as shown in Table 1.

For dynamometric system, the performance and application of the force sensors are very important, which will sometimes play a decisive role in the system. In order to adapt to the particularity of institutions in the complex dynamic systems, some advanced force sensors have been designed and demonstrated better measuring effect than the conventional methods [15]–[17].

In the development research of lift feedback for the fin stabilizer, the decoupling of the hydrodynamic force, the lift measurement, the design and installation of the sensor are the main technical problems. Therefore, inspired by the above designed force sensors with high performance, the lift sensor was designed and analyzed. Furthermore, the controller of the fin stabilizer should also be improved accordingly. In the design process of the conventional fin stabilizer, the linear roll model is adopted, so the serious nonlinear and uncertain factors are neglected. Since a severe disturbance is difficult to predict, a fixed controller is not suitable for the system. For these reasons, a novel force sensor was designed using Spoke-type structure and Wheatstone bridge-type strain measurement conversion circuit. And the novel mechanical decoupling method of fin hydrodynamic force and the lift measurement method are proposed. And then the lift measurement position and the installation of the sensors are given, which fundamentally solve the problem of installation and maintenance. The control system and controller of the fin stabilizes are improved.

The structure of the paper is organized as follows. Section 2 presents the lift problem formulation of ship fin stabilizer and introduces a novel lift measurement scheme. Section 3 details the design and calibration of the novel lift sensors. In Section 4, the control system of fin stabilizer is improved, including system structure and controller design. The simulation and experimental results and discussion are presented in Section 5. Finally, conclusions and future works are summarized in Section 6.

II. PROBLEM FORMULATION FOR LIFT OF THE FIN STABILIZER

A. ANALYSIS OF SHIP ROLLING MODEL

For a ship equipped with fin stabilizer, the nonlinear equation of ship rolling motion is represented by the Conolly nonlinear model [18], which is described as follows:

$$(I_x+\Delta I_x)\frac{d^2\phi}{dt^2}+B_1\frac{d\phi}{dt}+B_2\left|\frac{d\phi}{dt}\right|\frac{d\phi}{dt}+C_1\phi+C_3\phi^3+C_5\phi^5 = -K_D-K_C \quad (1)$$

where I_x is the moment of inertia by the vertical axis of ship barycentre, ΔI_x is the additional moment of inertia by the vertical axis of ship barycentre, ϕ is the roll angle of ship. B_1 and B_2 are the damping moment coefficients, C_1, C_3 and C_5 are the restoring moment coefficients, $B_1\frac{d\phi}{dt}+B_2|\frac{d\phi}{dt}|\frac{d\phi}{dt}$ is the damping moment caused by the roll angular velocity, $C_1\phi+C_3\phi^3+C_5\phi^5$ is the restoring moment of ship, K_D is the disturbing moment caused by the waves and K_C is the control moment provided by fin stabilizer.

Here, wave disturbance moment K_D can be expressed as

$$K_D = Dh_e\beta_{\max}\sin\omega_e t \quad (2)$$

where D is the ship displacement, h_e is the metacentric height, $\beta_{\max}$ is the maximum significant wave angle and ω_e is the ship's encounter frequency.

For a pair of fin stabilizers, the control moment K_C can be expressed as:

$$K_C = 2W_c l_f F_l \quad (3)$$

where W_c is the transfer function from the control command to lift, l_f is the arm of lift and F_l is the lift of fin.

The ship equipped with a pair of fin stabilizers is shown in Figure 1.

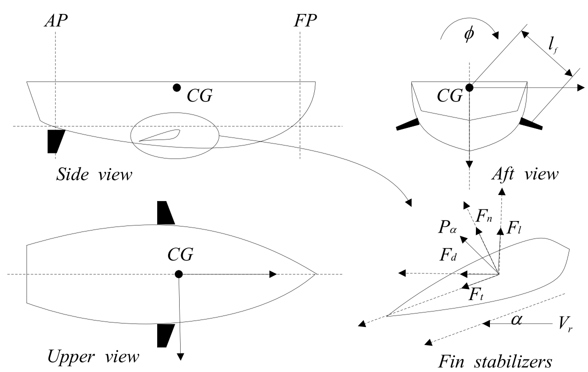

FIGURE 1. The ship equipped with a pair of fin stabilizers.

Equation (1) implies that if the control moment K_C provided by fin stabilizers equals the disturbance moment K_D caused by the waves, the right side of Equation (1) is zero. Namely, the ship stops rolling. But the marine environment is very complex, and the interference of waves is difficult to measure. Therefore, the control moment K_C provided by fin stabilizers needs dynamic adjustment. According to Equation (3), the accuracy of control moment K_C depends on the accuracy of the measured lift F_l when W_c and l_f are known. How to solve the problem of wave resistance is focused on the measurement and control of lift F_l.

B. ANALYSIS OF LIFT DEVIATION

The lift measurement in a direct way is very difficult to obtain. However, the fin angle is easy to measure. Fin angle is usually used as the feedback of servo system for conventional fin stabilizer, and fin angle is controlled according to real-time measured ship rolling signal. Usually, the theoretical value of lift is estimated by the relation between lift and fin angle. But the actual value of lift is constantly changing, and there is a deviation between them. The deviation can directly affect the control precision of fin stabilizer system and reduce the rolling effect. Therefore, it is necessary to study it deeply.

Because the characteristic of lift F_l in fluid is very complex, it is necessary to analyse it theoretically and experimentally. According to the force coefficient calculation theory for fin in viscous fluid, when fin and flow have a fixed angle, lift is approximately kept constant, which is shown as follows:

$$F_l = \frac{1}{2}\rho V^2 A C_0 (1-\frac{A_2}{A_0})(1+k)\alpha \quad (4)$$

where ρ is the fluid density, V is the fluid velocity, A is the projection area of fins and α is the fin angle.

$k = \tan\frac{\beta}{\alpha} \cdot C_0 = \frac{4\mu A_0}{\rho V^2}, A_0 = \frac{\sqrt{2}a_0}{\delta_1}\theta_0, A_2 = \frac{\sqrt{2}a_0}{\delta_1}n\pi, (n = 0, 1, 2, \cdots), \theta_0 = \arctan\frac{V\cos\alpha}{\sqrt{2}a_0} - \arctan\frac{\mu}{\sqrt{2}a_0}$. 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 and δ_1 is the thickness of boundary condition.

Then, the lift coefficient $C_L(\alpha)$ is defined as

$$C_L(\alpha) = C_0(1 - \frac{A_2}{A_0})(1 + k)\alpha \tag{5}$$

Equation (4) shows that it is approximately linear between lift F_l and fin angle α. The static approximation is used as the linear input of the control system when designing the conventional fin stabilizers in engineering applications. Aiming at the hydrodynamic characteristics of fin, dynamic experiments of fins in water tank were carried out by Ship Stabilization and Control Technology Research Institute [10].

The main experimental fins are shown in Figure 2.

FIGURE 2. Main experimental fins.

In Figure 2, the experimental fins are made based on the actual fin stabilizers, which satisfy geometric similarity, kinematic similarity and dynamic similarity. They are different in character of chord length of root, chord length of tip, fin height, shaft distances root, shaft distances tip, sweepback and shaft coordinate. Taking NACA0015 fin as an example, its dynamic lift characteristics and static lift characteristics were studied. The main model parameters of fin are shown in Table 2.

TABLE 2. 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

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

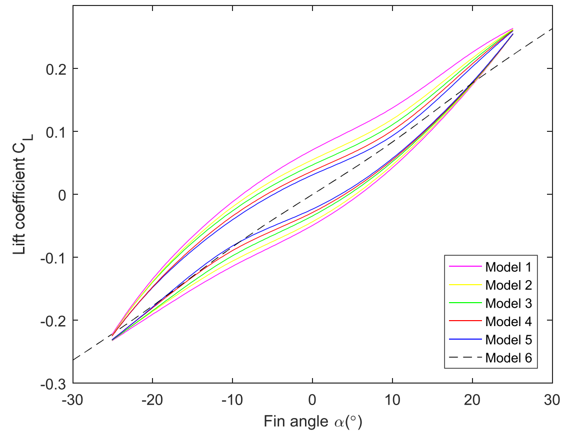

FIGURE 3. Relationship between lift coefficient and fin angle.

From the experimental results shown in Figure 3, the dynamic characteristic of lift coefficient and fin angle is not an ideal straight line, but a closed curve shape. The closed curve shape means a nonlinear relationship. Therefore, there is a large deviation between theoretical estimation value and actual measurement value for the lift, which was analyzed and the main factors are as follows.

(1) Reynolds number, the fin heave Froude number and fin length Froude number are changing in real time with the ship and the sea, which can influence the uncertainties and nonlinearities of the lift coefficient.

(2) In the design of fin stabilizers, the lift coefficient of fin is the static hydrodynamic coefficient, not the dynamic hydrodynamic coefficient of actual fin, which makes the control system of fin stabilizers not optimal. At the same time, beyond the stall angle range, the linear characteristics will be further weakened.

In addition, fin angle feedback control has the following problems:

(1) In Equation (4), the velocity used for calculating control moment should be the flow velocity, that is, the velocity of fin to flow field. For a fin moving with the ship, the velocity should be the relative velocity between the sea and the ship itself, which is affected by the ship and the bilge keel. In practical applications, it is difficult to measure this relative velocity. Thus the ship's speed is generally used to approximate it.

(2) The fin angle α used for calculating control moment should be the effective angle of attack $\alpha_e(\alpha_e = \alpha_{\mathrm{m}}+\alpha_\omega+\alpha_{\mathrm{p}})$, i.e., the included angle between fin and flow field. α_{m} is the angle of attack of fin, which equals the mechanical angle of fins α. α_ω is the additional angle of flow velocity and change of flow direction caused by ship. α_{p} is the velocity angle caused by roll motion and forward motion. α_ω can be seen as a random disturbance angle, which varies greatly in high sea states and low encounter frequencies. Since α_e cannot be measured, the deviation lift is caused by replacing α_{m} with α_e in practical applications.

TABLE 3. Experimental status of fin.

Model	Speed (Kn)	Swing (°)	Swing period (s)	Dimensionless frequency
Model 1	3.0	25	2.467	0.06
Model 2	3.0	25	3.700	0.04
Model 3	3.0	25	4.933	0.03
Model 4	3.0	25	6.167	0.024
Model 5	3.0	25	7.400	0.02

(3) The ship itself, and auxiliary devices such as bilge keel, will also have certain effects on the lift characteristics of fin stabilizers. In fact, part of fin is in the boundary layer of the ship. Due to the ship's influence, the velocity of this part is lower than that of the inflow, which increases the hydrodynamic characteristics of the fin. However, fin angle feedback can't reflect these interference factors.

(4) For dual fin stabilizer system, the front fin and the rear fin interacts with each other. The rear fin is disturbed by the front fin, and the impact is very complex. Actual lift characteristics usually decrease, but sometimes rise.

(5) When the ship is sailing on the sea, the fin's movement consists of the rotating shaft, the forward motion and the rolling, heaving and pitching of the ship. These complex motions cause a large change in lift, which is related to fin angle, fin angular velocity and fin angular acceleration.

If the lift generated on the fin is used as system feedback, W_c in Equation (3) becomes a scale factor k.

Then

$$K_C = 2klu \tag{6}$$

Ignoring the influence of servo system, it is considered that output tracks input in the working frequency band. Lift command u equals the lift of fin. The above problems are overcome, which can improve the accuracy of lift and enhance the anti-rolling effect. In addition, the lift feedback is also beneficial to the power design of driving fin servo system and can improve the comprehensive control performance of fin stabilizers.

C. DESIGN AND ANALYSIS OF NOVEL LIFT MEASUREMENT SCHEMES

In recent decades, conventional fin stabilizers have been widely used in the field of ship stabilization control, and its excellent performance is confirmed. Lift measurement can be achieved by improving the original fin shaft mechanism. This can inherit practicability and reliability of conventional fin stabilizers, and compensate the inaccurate system feedback. Improving the original fin shaft mechanism enables real-time lift measurement, which does not depend on the fin angle to estimate lift. The lift feedback is designed to reduce the overall error of the system.

1) STRUCTURE OF NOVEL FIN SHAFT DEVICE

Compared with the conventional fin stabilizers, the improved design of the novel fin shaft device is to install two force sensors at the upper bearing. Through the force analysis and measurement of the fin axis structure, the actual lift on the fin is obtained. The changes in the novel scheme are very small, which does not affect the original strength and sealing requirements. Thus, the novel scheme is easy to realize in practical engineering applications. The structure of novel fin stabilizer shaft device is shown in Figure 4.

In the design, the upper support bearing is set as the force measuring point, which has the following advantages.

(1) The installation, testing and maintenance of sensors are convenient in the ship cabin.

(2) According to the principle of moment balance, the bearing force of the upper support bearing is smaller than that of the lower support bearing. The range and volume of the sensors are correspondingly reduced. Therefore, the range of possible sensors is extensive.

2) MEASUREMENT AND ANALYSIS OF LIFT

The fin shaft can be simplified into a slender beam. The angular contact roller bearing with upper support and spherical roller bearings with lower support can be approximated into two simple supports B and C. BC of fin shaft can be approximately equivalent to a double fulcrum simple beam. The hanging dimension of fin shaft CD is longer, which can be approximated as a cantilever beam. Therefore, the overall structure of fin shaft can be approximated as the composite beam structure of a double support beam and a cantilever beam. The fin shaft and its simplified form are shown in Figure 5.

When the fin rotates in sea, the hydrodynamic force can be approximated as the resultant force F_D acting on the pressure center point D of the fin. The support reaction force of angular contact roller bearing with upper support point B is F_B, and support reaction force of spherical roller bearings with lower support point C is F_C. The unbalance force of the hydraulic cylinder driving the fin rotation at point A is F_A. l, m and n are the length of each segment as shown in Figure 5, respectively.

According to moment balance, it can be obtained:

$$F_D m - F_A(l+n) + F_B n = 0 \tag{7}$$

Then

$$F_D = \frac{F_A(l+n) - F_B n}{m} \tag{8}$$

Therefore, the hydrodynamic force can be obtained by measuring the support force F_B of the point B and F_A.

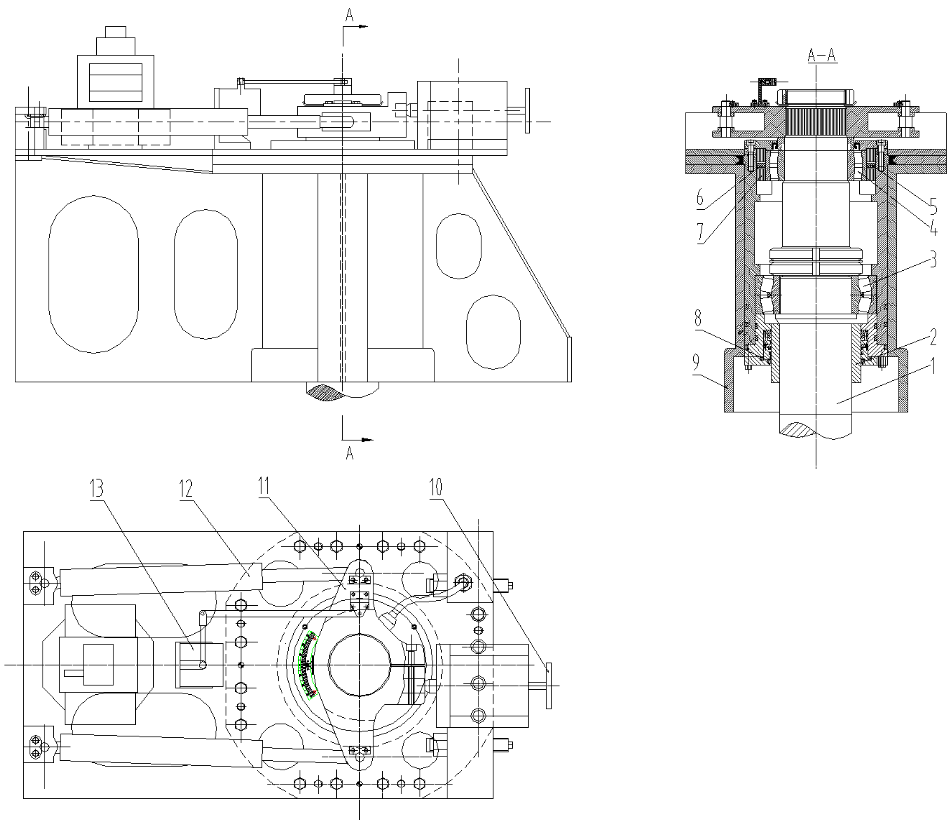

FIGURE 4. Novel fin shaft structure. 1-fin shaft, 2-shaft sleeve, 3-spherical roller bearings with lower support, 4-angular contact roller bearing with upper support, 5- 1# sensor, 6- 2# sensor, 7- sensor base, 8-sealing ring, 9-case body, 10-brake, 11-bearing seat of micro motion, 12-hydraulic cylinder, 13-fin angle feedback device.

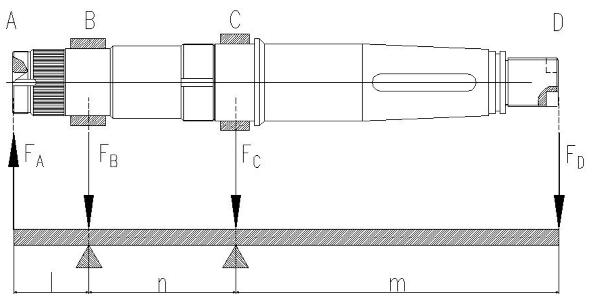

FIGURE 5. Fin shaft and its simplified form.

But how to decompose lift from the hydrodynamic force conveniently and effectively, which is necessary to design the installation mode of the sensors.

For Fin shaft, its diameter is much smaller than its length. So it is reasonable to simplify it into an ideal slender rod. However, its diameters are not the same everywhere. So it may touch the fin box when the hydrodynamic force is maximum. Then the lift of measurement is disturbed. The potential errors may be caused.

III. DESIGN AND CALIBRATION OF THE LIFT SENSOR

In the process of control system design, the sensors are very important as the feedback element. Performance of the sensors will sometimes play a decisive role in the system. Similarly, in the fin stabilizer control system, in order to achieve lift feedback, the sensor to meet lift measurement requirements is the key factor. Due to the unique structure and function of the fin stabilizers, the existing force sensors can hardly satisfy the lift measurement of the fin stabilizers in shape or size. Hence, a novel sensor is designed.

A. DESIGN AND ANALYSIS OF THE SENSORS INSTALLATION METHOD

Because of the bearing with fin shaft movement, the movements of rotation and translation exist at the same time. To ensure lift measurement is accurate and not affected by other forces, a specialized sensor installation method needs to be designed. Then a square micro bearing seat is designed. When the fin is subjected to dynamic hydrodynamic force, the fin shaft is driven, and the micro bearing seat generates a micro motion along the lift direction. The corresponding force sensor can measure the lift on the upper support point. When the fin is under resistance, the micro bearing seat will lean against the case body due to the vertical lift direction. So the resistance does not interfere with lift measurement. The sensors installation method is shown in Figure 6.

A pair of force sensors is installed at the opening of the case body. In order to ensure the accuracy of the measurement, a certain prestress is added to the sensor by the fastener, which

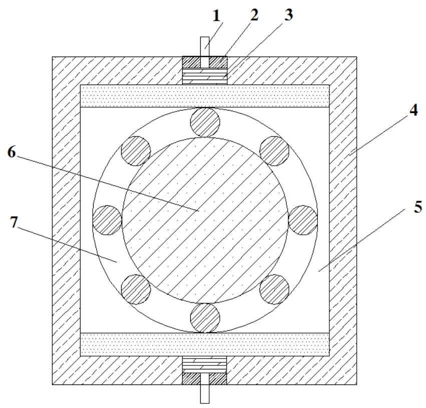

FIGURE 6. Sensors installation method. 1-sensor wiring slot, 2-fastener, 3-force sensor, 4-bearing housing, 5-micro bearing seat, 6-fin shaft, 7-angular contact roller bearing with upper support.

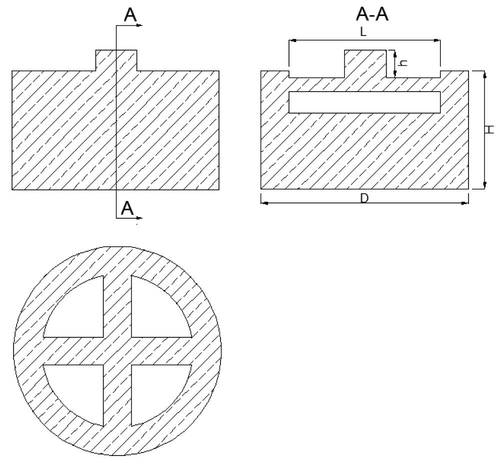

FIGURE 7. Structure of novel Spoke-type sensor.

can avoid the possibility of dead space caused by the gap, meanwhile ensuring the stability of the bearing seat after the balance force.

The designed sensors installation method has the following advantages.

(1) Owing to the specially designed bearing seat, the interference can be avoided when measuring the lift.

(2) Sensor circuit layout is simple, which is convenient for installation and maintenance.

(3) If the force sensor is installed on the resistance side of the bearing seat, the resistance on the fin can be measured simultaneously. So this method can be used to measure the dynamic hydrodynamic of other hydrofoils. Such design can be applied in a wide range of fields.

B. DESIGN AND ANALYSIS OF SENSOR FORCE STRUCTURE

Considering the uniqueness of lift measurement mechanism for the fin stabilizers and the characteristics of various force sensors, the Spoke-type force structure was used to design a special low-cost sensor for lift measurement. The structure of the novel sensors is shown in Figure 7.

Here, l is the length of the stressed beam. h is the height of the spoke. H is the height of the sensor. D is the diameter of the sensor.

In Figure 7, the sensor consists of the hub, rim and spokes. Among them, there are four spokes. And the two spokes on a line constitute a stressed beam. Because the spokes are symmetrical, according to the principle of shearing force, the force of a beam is $F/2$. According to the beam structure fixed at both ends, the bending moment caused by stress F at middle point is as follows.

$$M = FL/8 \tag{9}$$

The moment in the middle section of the spoke is zero. Therefore, a resistance strain sensitive element is installed at the neutral layer of the section. When the sensor is force balanced, the structure is stable, and the deformation space of strain is relatively large, which is convenient to lead out the wire.

C. DESIGN AND ANALYSIS OF SENSOR CIRCUIT

After determining the force structure of the sensor, it is necessary to design an accurate conversion circuit. The corresponding mechanical deformation is converted to electrical signal to achieve the desired purpose.

1) PRINCIPLE OF MEASUREMENT

There are many types of strain gauges used for measuring force, such as metal resistance type, semiconductor, magnetic body, etc. In this paper, the resistance strain sensitive element is adopted, which is the most commonly used materials for the force sensor. The basic principle is to paste the strain gauge on the corresponding elastomeric surface, and then elastomeric surface will produce strain due to force. The strain forms a selective diffusion strain zone on the strain gauge, and the variation of resistance in strain gauge zone with stress in strain crystal. Therefore, the resistance changes can be measured by strain gauge to measure force indirectly.

Usually, the relationship between resistance R and resistance change ΔR of strain zone can be expressed as follows.

$$\frac{\Delta R}{R} = \frac{\Delta \rho}{\rho} = \pi\sigma = \pi E\varepsilon = K\varepsilon \tag{10}$$

where, ρ is resistivity. $\Delta\rho$ is the change of resistivity. π is the piezoresistive coefficient along a certain crystal direction L, and the L denotes crystal direction. E is elastic modulus. σ and ε are stress and strain respectively. K is the sensitivity coefficient of the strain, which polarity depends on the conductivity type and its value varies with the orientation of the crystal.

For resistive strains, the sensitivity coefficient can be described by Equation (11) under ideal conditions.

$$K = \frac{\Delta R}{R} \times \frac{1}{\varepsilon} = \frac{\Delta \rho}{\rho\varepsilon} = (1 + 2\mu_s) + C(1 - 2\mu_s) \tag{11}$$

where, μ_s is Poisson's ratio, which represents the transverse deformation coefficient of metallic materials, $\mu_s \approx 0.3$. And $(1 + 2\mu_s)$ is the change of resistance caused by the shape effect. C is the scale coefficient of metal conductor lattice structure, for commonly used metals or alloys, $C = \{-12, 6\}$.

The deformation of the elastomeric surface is transmitted through the substrate and the binder to the sensitive grid of the strain gauge. And compared with the elastic modulus of strain gauge sensitive grating, the elastic modulus of substrate and binder are often different. So the transfer strain transition zone is formed at the ends of the substrate and the sensitive grid. Due to the existence of the transition region, the actual strain sensitivity coefficient is less than the theoretical value, usually taken as $K \approx 2$.

In addition, according to the mechanical structure of sensor designed in Section 3.2, the strain along 45° direction of the spoke neutral layer can be obtained as follows.

$$\varepsilon = \frac{3F}{8bhG} \tag{12}$$

where G is pressure coefficient.

2) CONVERTING CIRCUIT OF THE SENSOR

In the design of sensors, 2 or 4 strain gauges are usually combined. And they form a Wheatstone bridge circuit, which converts the resistance change caused by strain to voltage change. Its combinational circuit is shown in Figure 8.

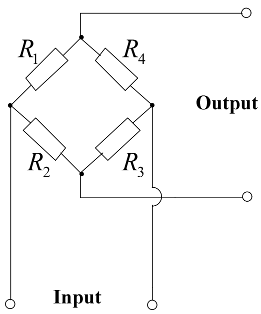

FIGURE 8. Wheatstone bridge circuit.

The input voltage of the bridge is E, and then the output voltage is:

$$\Delta E = \frac{R_1R_3 - R_2R_4}{(R_1 + R_2)(R_3 + R_4)}E \tag{13}$$

If initial conditions are set as $R_1 = R_2 = R_3 = R_4$, then the output voltage is $\Delta E = 0$. If one of these strain bodies is subjected to strain and causes ΔR change, then the output voltage will change accordingly.

Choosing that $R_1 = R + \Delta R$, $R_4 = R - \Delta R$, $R_2 = R_3 = R$, if the input voltage is E, then the output voltage is $\Delta E = \frac{\Delta R}{2R}E$. If the input current I is constant current source, then the output voltage is $\Delta E = \frac{\Delta R}{2}I$.

Considering the precision of sensor design and the difficulty of realization, the above mode is selected in the paper, and the input current I is constant current source. The relationship between output voltage and input voltage can be simplified as

$$\Delta E = \frac{\Delta R}{2R}E = \frac{K\varepsilon}{2}E \tag{14}$$

D. CORRECTION OF TEMPERATURE COMPENSATION

The biggest disadvantage of using strain gauges to design force sensors is that temperature variations have a great impact on characteristics of strain gauges, which is mainly reflected in the following two aspects.

(1) Change of temperature can cause deflection of bridge zero.

(2) Change of temperature can cause sensitivity change of bridge.

In order to solve the above problems, the corresponding compensation measures are adopted to improve the design of bridge based on Figure 8.

The effect of temperature change on bridge zero is generally required to be as small as possible. Therefore, in addition to the temperature compensation of the strain gauge itself, it is often necessary to correct the different resistance temperature coefficients in the bridge, which achieves the purpose of eliminating the zero drift.

Similarly, the sensitivity of the bridge based on the strain gauge is mainly affected by temperature. The key of sensitivity compensation is to eliminate the influence of temperature change on elastic coefficient and strain coefficient. The solution is generally putting resistances in series in the bridge, which are the same as those used in temperature compensation circuits.

Combined with the above analysis, the improved design of the bridge circuit is shown in Figure 9.

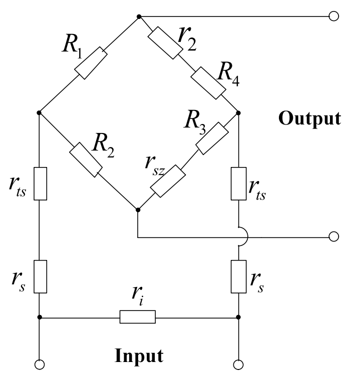

FIGURE 9. Bridge with temperature compensation.

Here, $R_1 \sim R_4$ are the resistances of strain gauges. Bridge balance is adjusted through resistance r_2, which value is generally around 1Ω. r_{sz} is the compensating resistance of zero-temperature characteristic. It usually adopts nickel-copper material and its temperature coefficient is different from strain gauge. The zero change can be reduced due to the difference of temperature coefficients. And its value is generally less than 1Ω. r_{ts} is the compensating resistance of sensitive-temperature characteristic, which material is the same as r_{sz}, and the compensating resistances on both sides of the input are the same. In order to make the bridge reach the initial balanced state, the output adjustment resistance r_s and the input adjustment resistance r_i are added, which are at the kΩ level.

E. DESIGN AND ANALYSIS OF KEY PARAMETERS

Taking the parameters of a certain fin stabilizer installed on ship as an example, the key parameters of the sensor are designed and calculated.

The ship uses a pair of fin stabilizers and provides approximately 1100000 Nm control moment. The distance is 5.56 m between the hydrodynamic point of the fin and the center of gravity of the ship.

According to the above conditions, the control moment required for single fin is 5.5×10^5 Nm, i.e., the hydrodynamic forces on single fin is about 9.90×10^4 N.

In order to make the design allowance and simplify the calculation, in the design of the sensor structural parameters, $F = 1 \times 10^5$ N is set.

The sensor has four spokes, and the force of the center is actually acting on the two fixed end beams. Thus, the maximum bending moment of the spoke root is as follows.

$$M_m = \frac{1}{2}\frac{FL}{8} = \frac{FL}{16} \tag{15}$$

The resistance strain sensitive element is installed at the neutral layer of the middle section of the spokes, and then its shear force is:

$$Q = F/4 \tag{16}$$

The alloy 35CrMnSi is selected as the skeleton material of the spokes, and the corresponding stress is σ_{b}. The allowable stress coefficient of the alloy is λ_σ. Hence the allowable stress of the alloy is:

$$[\sigma_{\mathrm{b}}] = \sigma_{\mathrm{b}} \cdot \lambda_\sigma \tag{17}$$

The safety factor is designed as δ, then the actual allowable stress of the alloy material is:

$$[\sigma] = [\sigma_{\mathrm{b}}]/\delta \tag{18}$$

Therefore, the coefficient of cross section for sensor spoke is:

$$\mathrm{W} = \frac{\mathrm{M}_m}{[\sigma]} \tag{19}$$

Spoke section is rectangular, and then the section size of spoke is calculated as follows:

$$W_b = bh^2 \big/ 6 \tag{20}$$

Then the height of the spoke is:

$$h = \sqrt{\frac{6W_b}{b}} \tag{21}$$

The corresponding shear stress is:

$$\tau_{\max} = \frac{3Q}{2S} = \frac{3Q}{2bh} \tag{22}$$

where S is the area of the shear plane.

$$G = \mu E \tag{23}$$

Therefore, the positive strain coefficient acted on along 45° direction of the spoke neutral layer:

$$\varepsilon = \frac{\tau_{\max}}{G} = \frac{\tau_{\max}}{\mu E} \tag{24}$$

Then the maximum deformation size of the sensor is:

$$\Delta h = \varepsilon h \tag{25}$$

Through the above analysis, the key design parameters of the sensor are shown in Table 4.

Table 4 is the design and calculation results of the sensor parameters according to the practical application of fin stabilizers. The results show that the maximum displacement on the upper support of fin shaft is very small when the hydrodynamic force is measured. It is only 9.055×10^{-2} mm, which can't affect the normal operation of fin shaft.

F. CALIBRATION OF THE SENSOR

According to the above key design parameters, the novel made lift sensor is as shown in Figure 10.

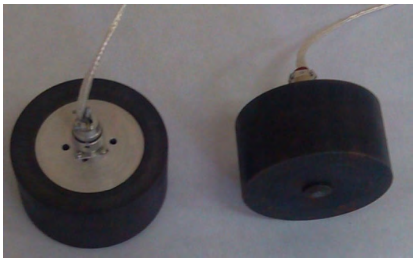

FIGURE 10. Novel made lift sensor.

After the actual test, the relevant working parameters of the sensor are shown in Table 5. The two sensors are calibrated practically, which specific data are shown in Table 6. According to the calibration data of Table 6, the calibration curve is plotted in Figure 11.

The results in Figure 11 show that the two lift sensors have good linear characteristics. The slope of 1# lift sensor is 2.10 mV/10^5N, and slope of 2# lift sensor is 2.20 mV/10^5N. The slopes of two sensors are basically the same, and the signals of the two sensors are synthesized to form a bipolar output. The calibration curve is shown in Figure 12.

The results in Figure 12 show that the slopes of two lift sensors can be considered to be the same within the allowable error range. If the two lift sensors form a pair, they can fully meet the needs of lift measurement when using the installation method in Section 3.1. Compared with other force measuring sensors, the lift sensor has the following advantages:

TABLE 4. Key design parameters of the sensor.

NO.	Symbol	Value	Unit	NO.	Symbol	Value	Unit
1	F	1.0×10^{5}	N	14	$\tau_{\max}$	1.994×10^{8}	N/m^2
2	l	50	mm	15	G	6.3×10^{10}	N/m^2
3	M_m	468.75	Nm	16	ε	3.165×10^{-3}	-
4	σ_b	9.0×10^{8}	N/m^2	17	Δh	9.055×10^{-2}	mm
5	λ_σ	0.55	-	18	R_1	350	Ω
6	$[\sigma_b]$	4.95×10^{8}	N/m^2	19	R_2	350	Ω
7	δ	1.4	-	20	R_3	350	Ω
8	$[\sigma]$	3.535×10^{8}	N/m^2	21	R_4	350	Ω
9	W	1.326×10^{-6}	m^3	22	r_2	1	Ω
10	W_b	1.326×10^{-6}	m^3	23	r_{sz}	0.8	Ω
11	b	10	mm	24	r_{ts}	10 - 30	Ω
12	h	28.21	mm	25	r_s	5×10^{3}	Ω
13	Q	3.75×10^{4}	N	26	r_i	4×10^{3}	Ω

TABLE 5. Relevant working parameters of the sensor.

NO.	Parameter name	Value	Unit
1	Rated load	2×10^{5}	N
2	Rated output	2±0.004	mV/V
3	Rated operating voltage	10	V
4	Input impedance	380±20	Ω
5	Output impedance	350±2.5	Ω
6	Temperature effect on zero	0.005%	FS/℃
7	Permitted overload	120%	FS

TABLE 6. Calibration data of two sensors.

NO.	Applied force	1# sensor	2# sensor
1	0	0	0
2	2	4.172	4.410
3	4	8.344	8.829
4	6	12.501	13.249
5	8	16.702	17.672
6	6	12.496	13.242
7	4	8.338	8.820
8	2	4.168	4.408
9	0	0	0
Unit	10^{5} N	mV	mV

(1) The designed sensor has good natural linearity. The deformation is small enough to ignore under the stress, which can overcome the force error caused by deformation. At the same time, the adverse effect caused by the change of the upper support position on fin stabilizer can be avoided.

(2) According to the principle of shearing force, the position of force measuring point has little effect on the precision of the sensor.

(3) The sensor has good symmetry, which can bear large eccentric force and lateral force.

(4) The sensor is flat in shape. So the deformation under load is small, and the interference between the measured structure and the sensor can be neglected.

(5) The sensor is easy to maintain and replace. It has wide measuring range and high precision. The service life is long and the performance is steady and reliable.

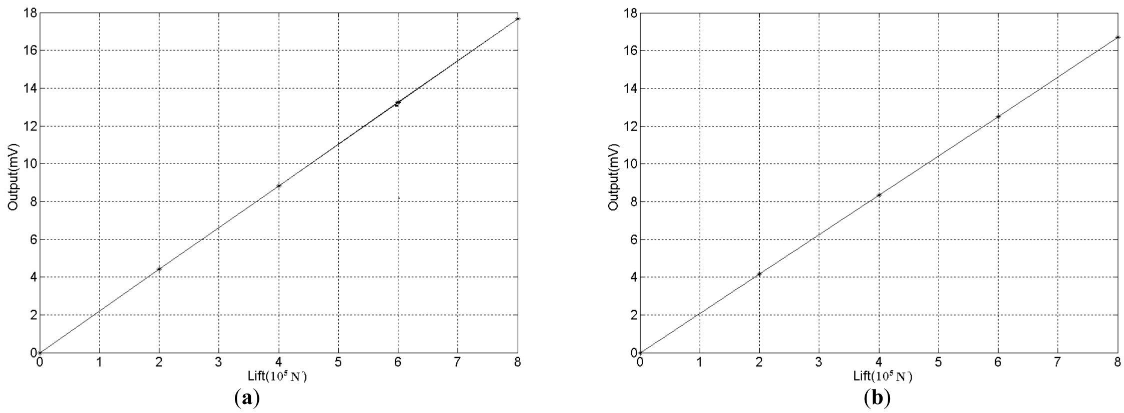

FIGURE 11. **Calibration curve of two lift sensors: (a) Calibration curve of 1# lift sensor; (b) Calibration curve of 2# lift sensor.**

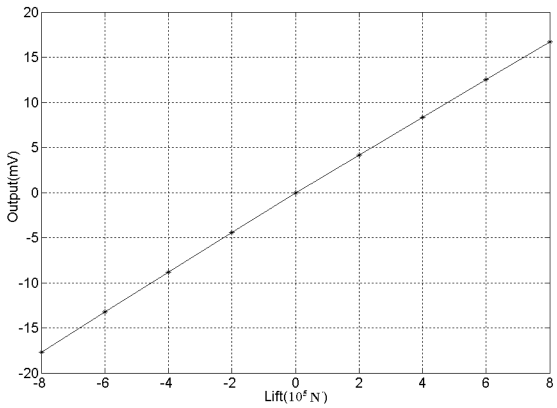

FIGURE 12. **Bipolar calibration curve of two lift sensor.**

IV. IMPROVEMENT OF FIN STABILIZER CONTROL SYSTEM

Real-time measurement of lift is used as the system feedback to the fin stabilizers, compared with the dynamic lift command required for ship roll stabilization. The obtained lift error command will drive the fin to rotate until achieving the desired lift, which can reduce the deviation between actual value and theoretical value. So this improved method can effectively compensate for the lack of conventional fin angle feedback. However, the structure and controller of the control system need to be improved accordingly. Completed the work above, the nonlinearity and uncertainty of rolling motion can be improved, and finally enhance the effect of roll stabilization.

A. IMPROVEMENTS FOR INNOVATION

In order to improve the anti-rolling effect effectively, the requirements of practical application as well as the control system itself should be considered, and the control improvements for innovation are made as follows.

(1) The effect of ship roll stabilization in the range of different ship speeds and heading waves is improved.

The conventional fin stabilizer is designed for the specific ship speed and heading wave and its scope of application is limited. The control system is improved so that it can work effectively at all speeds, and meet the requirements of different heading waves. And the power consumption is controlled within a permissible range, which can control fin angle, angular velocity and angular acceleration to achieve the best effect.

(2) The lift and fin angle saturation should not exceed the set value.

In high sea conditions, regardless of whether the anti-rolling effect is active or not, lift and fin angle saturation rate should be limited in a certain range to protect mechanical and hydraulic system.

(3) Stall should be avoided in high sea conditions.

In high sea conditions, mechanical angle of fin will increase. Especially in medium/low speed, dynamic stall should be avoided.

(4) Switching should be smooth and frequent switching should be avoided.

The switching of different working modes should be smooth, and the crew shouldn't feel the switching of the working modes. At the same time, the switch shouldn't be carried out frequently. Consequently, anti-rolling effect as well as the specific situation of the ship should be studied synthetically to design power restrictions, fin angle and angular velocity.

Therefore, the structure and controller of the conventional fin stabilizer system need to be improved.

B. COMPOSITION OF LIFT FEEDBACK CONTROL SYSTEM

(1) System structure

Compared with the fin angle feedback of the conventional fin stabilizer, the lift is used as system feedback of the system for improved fin stabilizer. So the system constitution is changed accordingly. The structure of lift feedback fin stabilizer is composed of three parts: integrated controller, electro-hydraulic servo system and motion feedback of the ship, which are shown in Figure 13.

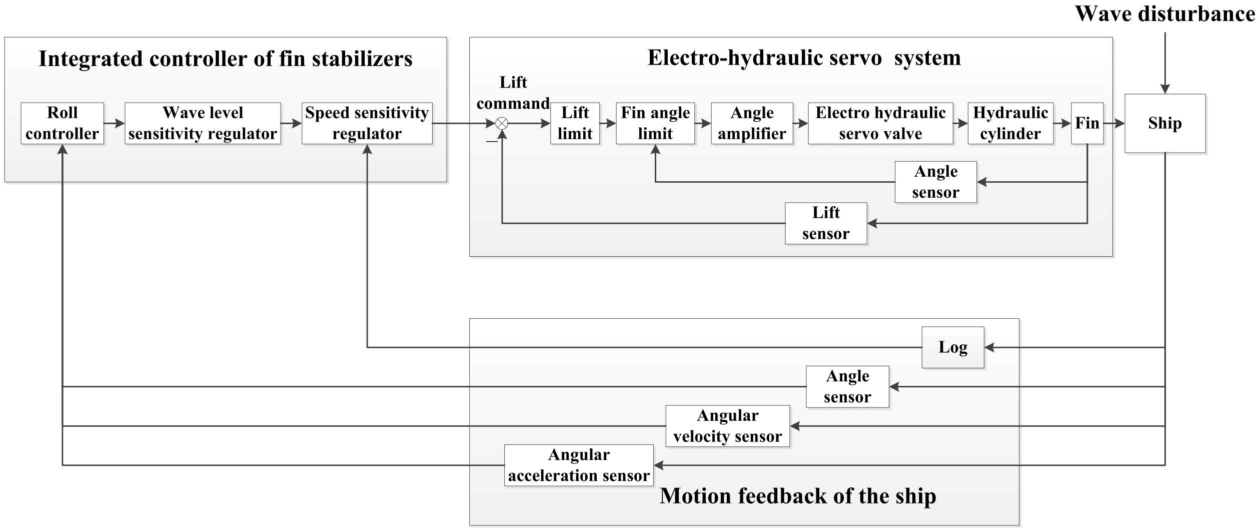

FIGURE 13. Structure of lift feedback fin stabilizer.

(2) Operation process

When the ship is disturbed by waves to produce rolling, the sensors of ship motion feedback can measure the roll angle, roll angular velocity and roll angular acceleration. They are input to the data processor. The processor processes the signal and then sends it as an input to the integrated controller. These data are transmitted to the data processor, and then sent to the integrated controller after processing as an input.

The integrated controller performs the operation according to the designed control strategy. The signals of the wave level sensitivity regulator and the speed sensitivity regulator are considered. Then the lift required by the fin stabilizer against the wave moment is obtained, which is used as a control signal to the electro-hydraulic servo system.

The hydraulic cylinder drives the fin shaft mechanism to the corresponding angle. Under the action of hydro-kinetic force, the control moment generated by the fin resists the disturbance moment. At the same time, lift sensor is used to measure the actual lift, which is transmitted to the electro-hydraulic servo system controller as a feedback signal. Finally, loops are formed and controlled.

(3) Controller design

The ship is subjected to severe external disturbances such as waves and the roll motion model is nonlinear and uncertain. Therefore, it is difficult to design the controller. In practical engineering applications, the nonlinearity and uncertainty of the system are generally ignored. And the linear system is designed according to the specific sea conditions, which anti-rolling effect is not ideal. Sliding mode variable structure control is a robust control strategy, which can be used to solve the nonlinear and uncertain problem.

In general, the sliding mode variable structure control requires that the upper bounds of the uncertainties for the system are known. However, for the actual fin stabilizer system, the upper bound value can't be accurately determined because of the strong external disturbances and its own complexity. RBF neural networks can be used to study the upper bound of interference, which can reduce the output chattering of the system.

V. EXPERIMENTAL RESULTS AND DISCUSSION

A. COMPOSITION OF EXPERIMENTAL SYSTEM

The experimental device is the novel experimental system of the lift feedback control fin stabilizer, which is mainly divided into three parts. The Composition of experimental system is shown in Figure 14.

(1) Electro-hydraulic servo actuator and measurement system

This part is mainly composed of servo controller, hydraulic actuator device, fin shaft, fin, and lift sensors device etc, and it is the main part of the experimental system. In this part, the nonlinear and uncertain relationship between the fin angle and the actual lift of the fin are encapsulated. And the fin is driven directly by the lift command. Depending on the actual hydrodynamic force, the fin is rotated to a suitable angle, which ensures that the fin can get the desired lift under various sea conditions.

(2) Lift load system

Although the lift load system is an auxiliary part of the experimental system, it is used to simulate the dynamic lift characteristics in the whole experiment. Therefore, the performance of this system has a direct impact on the research of lift feedback control fin stabilizer.

(3) System controller of lift feedback fin stabilizer

From the structural form, the lift feedback control is similar to the fin angle feedback control. But there are essential differences on the control characteristics. It is shown that the dynamic adaptive control is adjusted according to different situations. The design and detailed features of the controller are shown in [19]–[22].

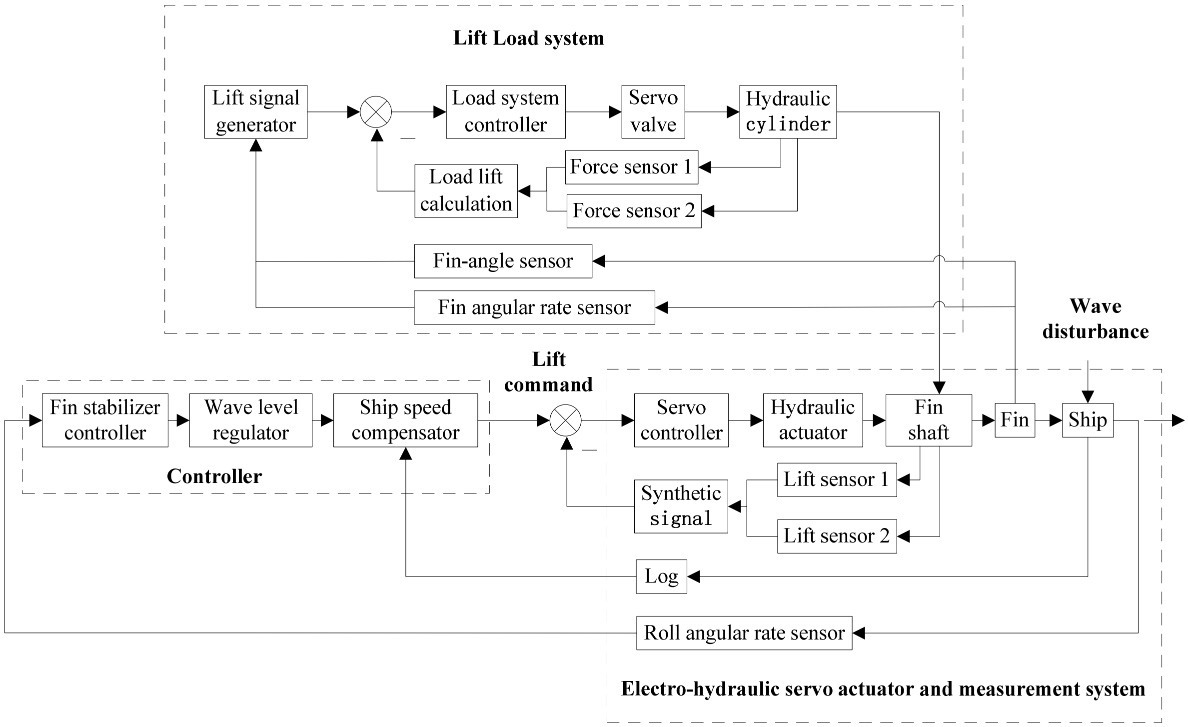

FIGURE 14. Composition of experimental system.

The main experimental devices of the lift feedback control fin stabilizer are shown in Figure 15.

B. MEASUREMENT ANALYSIS OF THE SENSOR

1) ACTUAL MEASUREMENT AND DATA PROCESSING

When the hydraulic servo system is forward-Loading and reverse-Loading, the data between voltage of the sensor and lift are measured. Lift curves are plotted according to the test results. Figure 16(a) shows it can't be used as the calibration data of the lift measurement mechanism. Further processing is required before use. The simplest way is to ignore the nonlinear characteristics of the lift measurement mechanism. The least squares method is used to fit the test curve with the linear scaling characteristics. The fitting linear function is: $y = 1.5402x + 2.4559$. Here, 2.4559 mV is the zero offset voltage reflected by the lift measurement mechanism on the lift sensor when it is not loaded. The fitting results are shown in Figure 16(b).

2) ANALYSIS OF LIFT MEASUREMENT RESULTS

In the measurement curve of lift sensor, hysteresis can be observed. Hysteresis causes a nonlinear interference in the measurement. The output voltage value is different in the process of the same lift value during pressurization and decompression. When the applied force changes near the maximum value, the output signal of the sensor does not change in time. In addition, if the signals of two sensors are synthesized, the output is not zero when the lift force is zero. And the main reasons are as follows.

(1) There is a certain leakage in the hydraulic line of loading system, and then the actual load lift is smaller than the measurement value. But Figure 16(a) does not show this influence factor.

(2) The performance of lift sensor is reduced. Due to the low processing technology and repeated test impact, the variant of the sensor may have undergone some deformation, which can directly reduce its performance and the acquired data is no longer a simple linear relationship.

(3) The lower support point of the fin shaft is a rotary bearing, which has a certain gap. When the loading force changes near the maximum value, the deformation of the supporting point will be released to a certain extent due to the movement of the supporting point. The result is that the sensor can not induce the change of lift in time.

The test data presented here not only reflects the characteristics of the lift sensor, but also gives a comprehensive reflection of the whole lift measurement characteristics. To sum up, it is mainly caused by the following factors.

(1) There is a certain gap between the lift sensor and the fin shaft.

Theoretically, the sensor is close to the fin axis and is integrated with the fin axis. But in fact, the fin shaft itself drives the fin to rotate, while the sensor is fixed. So there is a gap between the sensor and the fin shaft, which causes the nonlinearity of measurement.

(2) Effect of fin axis motion.

The fin shaft is not only affected by the lateral force, but also by self rotation. There is a coupling effect between the

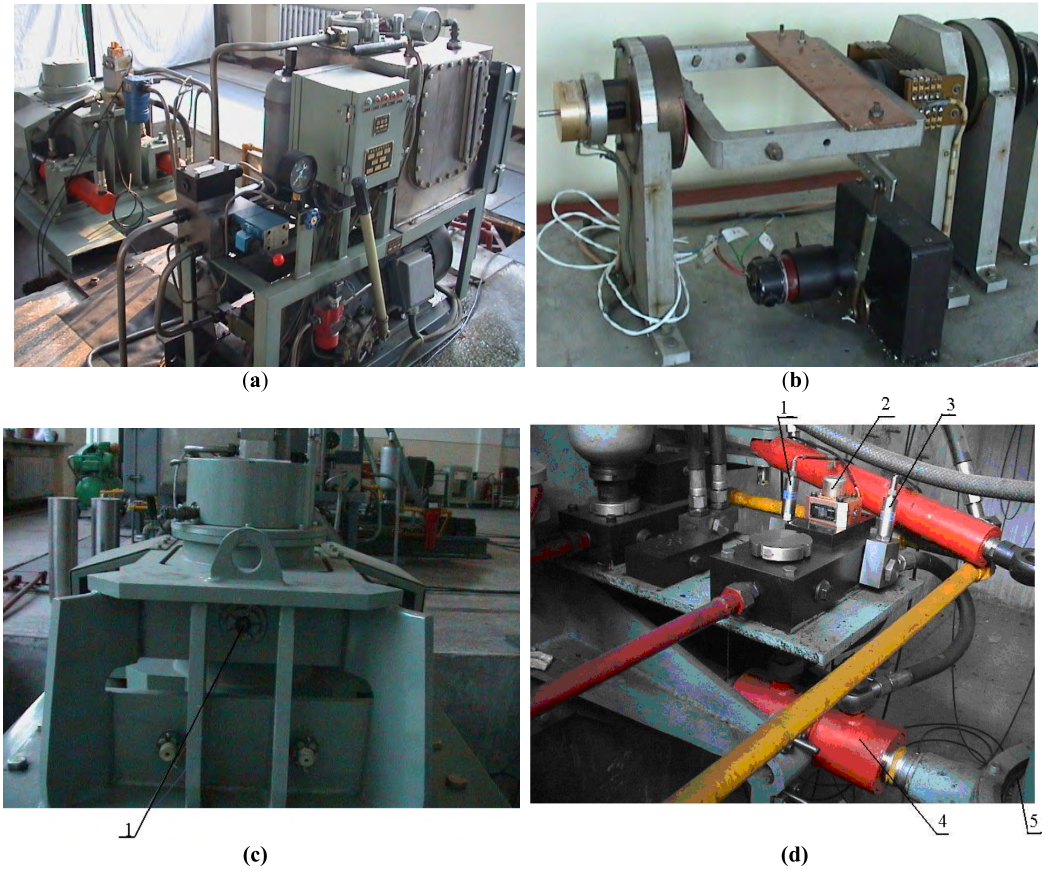

FIGURE 15. Main experimental devices of the lift feedback control fin stabilizer: (a) electro-hydraulic servo actuator and measurement system; (b) ship rolling motion simulator; (c) installation position of the sensor, 1-lift sensor; (d) electro-hydraulic servo actuator and measurement system, 1-sensor 1, 2-electro-hydraulic servo valve, 3-sensor 2, 4-hydraulic cylinder, 5-fin-shaft.

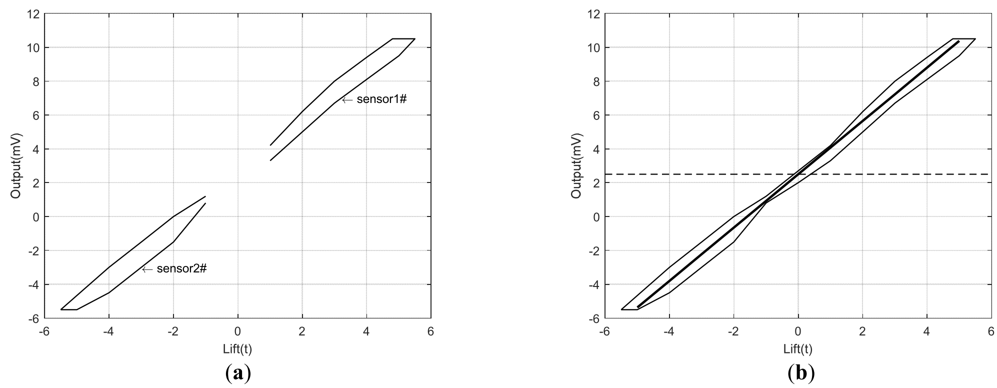

FIGURE 16. Lift data of test results processed: (a) actual measurement results; (b) lift curves obtained by fitting the data.

two directions of the fin shaft movement. While the lift sensor merely measures the lateral force, its measurement is affected by rotation.

(3) There is hysteresis between the measuring mechanisms, and the lift sensor is slow.

The force exerted on the fin shaft is usually variable and varies rapidly. The experimental results show that the pressure sensor can reflect the change of the force in real time. Due to the influence of the drive Mechanism lag and sensor induction time, the lift sensor always lags behind the pressure sensor. This lag affects the output of the lift sensor.

In the practical application, it is necessary to carry out pre correction to eliminate the influence of these factors.

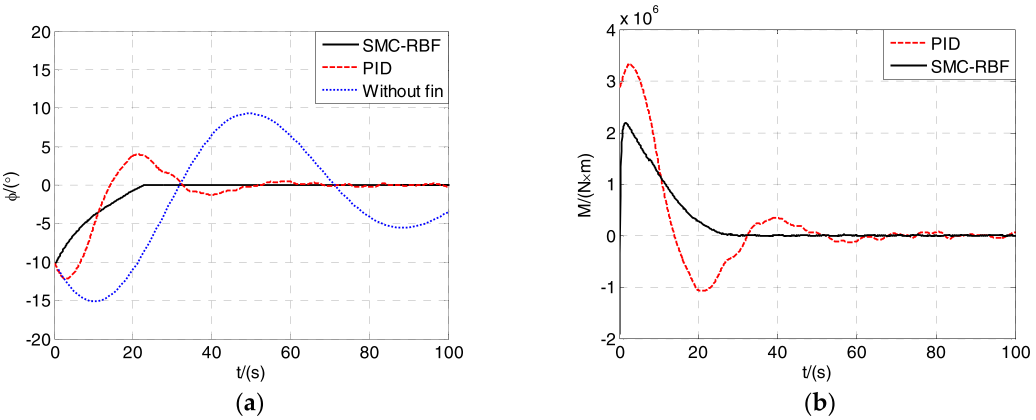

FIGURE 17. Dynamic simulation results: (a) simulation results of the roll angle; (b) simulation results of the control moment.

C. COMPARISON OF ANTI-ROLLING EFFECTS

Taking a fin stabilizer system of a certain ship as an example, the accuracy of the sensor for measuring lift is verified using the experimental platform of the fin stabilizer.

The internationally accepted ocean wave spectrum is the wave spectrum with double parameters proposed by the 12th International Towing Tank Conference. It is also referred to as ITTC double parameters spectrum [20]. Here, the wave is simulated using ITTC double parameters spectrum, which is adopted to recreate the fully developed sea environment. It is presented as following.

$$S(\omega_i) = \frac{173H_{1/3}}{T^4\omega_i^5}\exp(-\frac{691}{T^4\omega_i^4}) \tag{26}$$

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

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 Eq. (27) and (28).

$$\zeta_i = \sqrt{2S(\omega_i)\Delta\omega} \tag{27}$$

$$\zeta = \sum_{i=1}^{60}\zeta_i\cos(\omega_i t + \varepsilon_i) \tag{28}$$

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

The model is Conolly nonlinear model [20] as Equation (1). The main parameters are shown in Table 7.

The nominal model of fin stabilizer is obtained from Equation (1), and the parameters are set according to Table 7. The nominal model is shown in Equation (29):

$$\begin{aligned} d^2\theta/dt^2&+0.25174d\theta/dt+0.7056\,|d\theta/dt|\,d\theta/dt+0.648360\\ &-15.76960^3+20.65620^5 = -0.00973u-0.39479e-7K_D \end{aligned} \tag{29}$$

where u is the control variable of lift.

TABLE 7. The main parameters of simulation system.

NO.	Type	Symbol	Value	Unit
1	displacement	D	1457.26	t
2	ship length	L	98.0	m
3	ship beam	B	10.2	m
4	draught	T	3.1	m
5	metacentric height	h	1.15	m
6	resonant period	T_ϕ	7.8	s
7	speed	V_{ship}	18	Kn
8	significant wave	$H_{1/3}$	3.8	m

This paper mainly studies the design and measurement form of lift sensor and the structural characteristics of lift control system. The research of control strategy is mainly focused on [2], [5], [18], [21], and [22]. The design and comparison of different control strategies are introduced in these references. As a comparison, the improved PID control strategy is proposed in the simulation, which is usually used in practical engineering.

Under the same conditions, the simulation results of fin angle feedback control system and lift feedback control system are studied.

Here, the PID controller of fin angle feedback system is:

$$\alpha_{pid}(s) = (k_I\frac{1}{T_I s+1} + k_D\frac{T_{D1}s}{(T_{D1}s+1)(T_{D2}s+1)} + k_p)\phi(s) \tag{30}$$

According to the fin angle feedback system, the PID controller of the lift feedback system is shown as follows.

$$\begin{aligned} L_{pid}(s) \\ = (k_{IL}\frac{1}{T_I s+1} + k_{DL}\frac{T_{D1}s}{(T_{D1}s+1)(T_{D2}s+1)} + k_{pL})\phi(s) \end{aligned} \tag{31}$$

where, k_p, k_I and k_D are the adjustment coefficients of proportion, integral and differential in the controller respectively.

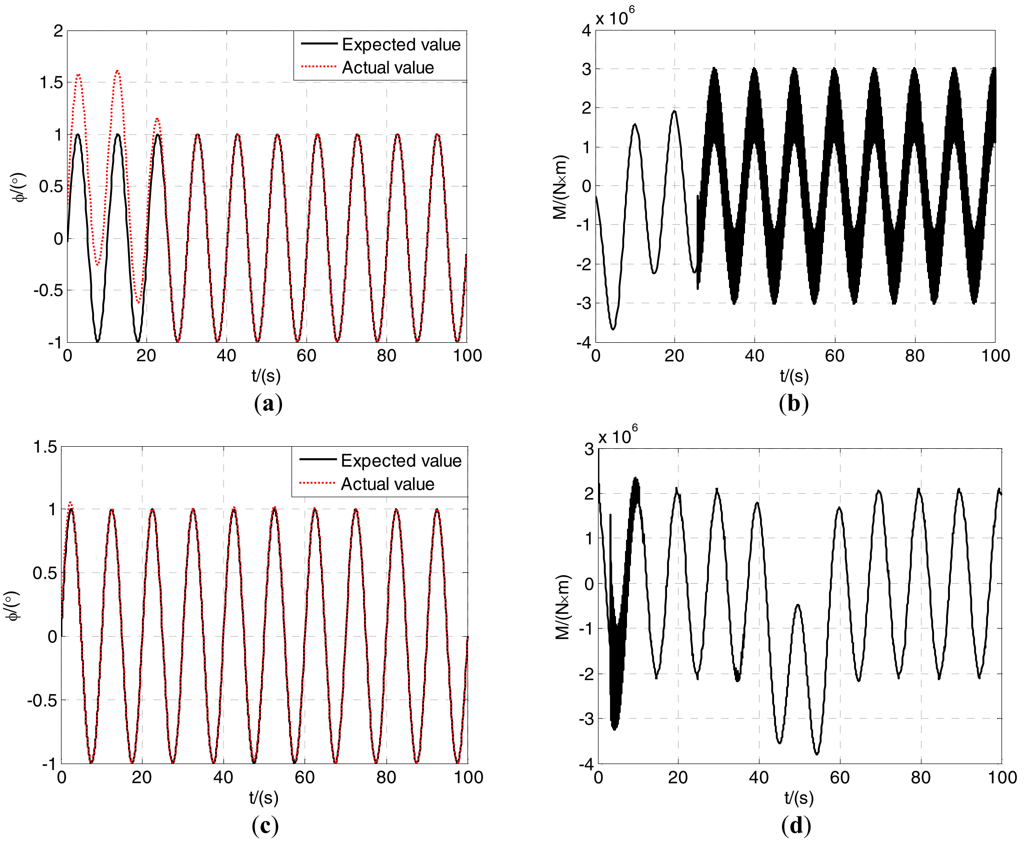

FIGURE 18. Comparison of simulation results: (a) tracking of fin angle by using the conventional sliding mode control; (b) adjustment of control moment by using the conventional sliding mode control; (c) tracking of fin angle by using the designed adaptive sliding mode control; (d) adjustment of control moment by using the designed adaptive sliding mode control.

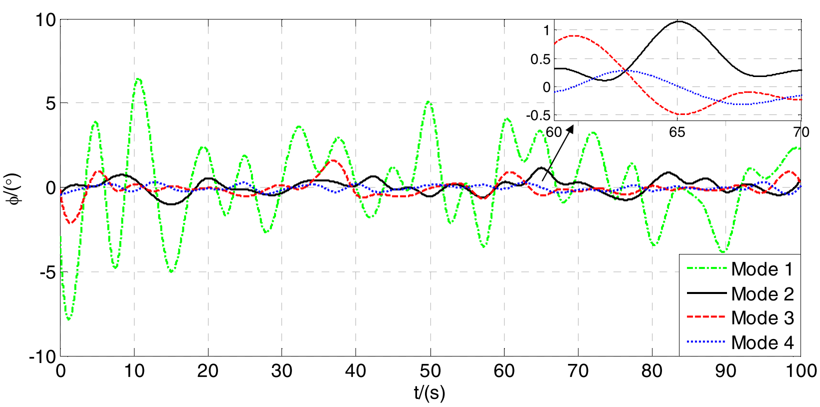

FIGURE 19. Comparison of simulation results for different modes.

T_I is time constant. T_{D1} and T_{D2} are the corresponding time constants.

Here $k_p = 6.89$, $k_I = 38.68$, $k_D = 2.05$, $T_I = 24.61$, $T_{D1} = 0.065$, $T_{D2} = 0.178$.

Simulations of roll motion for the ship by the experimental devices are carried out in two steps:

(1) Comparison of control system.

Comparison of fin angle feedback control system and lift feedback control system, the improved PID controller is adopted in practical engineering.

(2) Comparison of controller.

For lift feedback control system, control strategies are compared and analyzed. The sliding mode variable structure controller based on RBF neural network and PID controller is adopted to carry out simulation.

Taking the initial roll angle of $-10°$ as an example, the comparison of the two control systems is made. Under the action of wave disturbances, the dynamic simulation results of the roll angle and the control moment are shown in Figure 17.

TABLE 8. Statistics of simulation results of different modes.

Encounter angle (°)	30	45	60	90	120	135	150
Mode 1/ mean value (°)	0.879	1.565	2.549	3.012	1.750	1.161	0.816
Mode 1/ variance (°)	0.530	1.400	3.121	5.579	1.805	0.756	0.278
Mode 2/ mean value (°)	0.326	0.417	0.510	0.673	0.502	0.309	0.198
Mode 2/ variance (°)	0.100	0.175	0.234	0.312	0.217	0.186	0.009
Mode 2/ effect (%)	62.912	73.355	79.992	77.656	71.314	73.385	75.735
Mode 3/ mean value (°)	0.125	0.147	0.205	0.278	0.202	0.106	0.089
Mode 3/ variance (°)	0.050	0.129	0.140	0.238	0.099	0.005	0.002
Mode 3/ effect (%)	85.779	90.607	91.958	90.770	88.457	90.870	89.093
Mode 4/ mean value (°)	0.116	0.109	0.186	0.185	0.188	0.098	0.075
Mode 4/ variance (°)	0.003	0.007	0.013	0.029	0.009	0.006	0.001
Mode 4/ effect (%)	86.803	93.035	92.703	93.858	89.257	91.559	90.809

The results of 100 s in Figure 17 show that the PID controller has a stable anti-rolling effect at about 45 s, and the designed controller in this paper has a stable anti-rolling effect at about 22 s. The latter has faster dynamic response and smaller overshoot. So the latter is better than the former in the dynamic response. For the dynamic response of the control moment, PID controller is less than that of the controller designed in this paper, and it is constantly adjusted in a certain range.

Considering the chattering problem of sliding mode control for the lift feedback control system, the control strategies are simulated by using the conventional sliding mode control and the adaptive sliding mode control in this paper. A comparison of the two states in the tracking of fin angle and adjustment of control moment was made with the simulation results shown in Figure 18.

In Figure 18, for conventional sliding mode control on tracking of fin angle, in the first 23 s, the actual value lags behind the expected value and has a deviation at the peak value. After that, the tracking effect is good, but the corresponding parameters of control moment need to be adjusted in a certain range. Adjustment of parameters is large between the larger of in the 23 s $\sim$ 100 s, and there is an obvious chattering phenomenon. The adaptive sliding mode control designed in this paper has better overall performance. At the beginning, the adjustment of control moment changes in a small range. After 10 s, the RBF neural network is used to make it continuous, which can suppress chattering effectively.

The ship's navigation is simulated at the encounter angles 30°, 45°, 60°, 90°, 135° and 150° respectively. Due to limited space, at the encounter angle 90°, tracking results of fin angle are given as shown in Figure 19. In other cases, the simulation results are shown in Table 8 and Figure 20.

Here, Mode 1 is without fin. Mode 2 is the conventional fin stabilizer by using PID controller in practical engineering. Mode 3 is the designed lift feedback fin stabilizer by using PID controller. Mode 4 is the designed lift feedback fin stabilizer by using the improved sliding mode control based on RBF neural network controller (SMC-RBF).

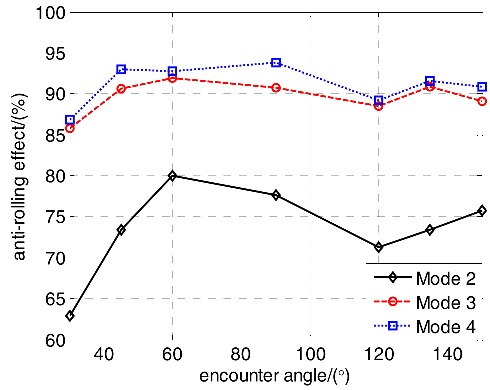

FIGURE 20. Comparison of anti-rolling effects for different modes.

The anti-rolling effect is calculated by (1- Encounter angle with fin/ Encounter angle without fin)$*$100 %. From the above results, the anti-rolling effect of Mode 2 is 62.912 % - 79.992 %. The anti-rolling effect of Mode 3 is 85.779 % - 91.958 %. And the anti-rolling effect of Mode 4 is 86.803 % - 93.858 %. Anti-rolling effect comparison of Mode 2 and Mode 3 demonstrates the lift feedback system is better than the conventional fin angle feedback system. Anti-rolling effect comparison of Mode 3 and Mode 4 shows the improved controller is more suitable for lift feedback system.

In short, the designed control system of the fin stabilizer in this paper can reduce the influence of nonlinearity and uncertainty on the system, which can improve roll stability. It is further proved that the designed sensor can measure the dynamic lift effectively and accurately.

VI. CONCLUSIONS AND FUTURE WORKS

Based on the analysis of the problems existing in the traditional fin stabilizer control system, a novel method is proposed to solve the deviation of conventional fin angle feedback by using lift as feedback in this paper. However, the difficulty of lift feedback control is lift measurement technology. According to the comparison and analysis of lift measurement methods in different research institutions, this paper presents a novel lift measurement scheme. Taking the precondition that the direction of fin lift and the direction of fluid motion are mutually perpendicular, the mechanical decoupling method of fin hydrodynamic force and the lift measurement method based on double bearing load are proposed. And the lift measurement position and the installation of lift measurement sensors are given, which fundamentally solves the problem of installation and maintenance for lift sensors. Then lift sensors are designed using Spoke-type force structure and Wheatstone bridge-type strain measurement conversion circuit. Compensation resistors are used in the circuit, which effectively reduce the effect of temperature on zero drift and sensitivity of lift sensors. The control system and control strategy of the conventional fin stabilizers are improved in order to solve the main technical problems such as simulation load and control in the laboratory simulation. Finally, the experimental verification is carried out. The results show that the designed sensors meet the requirements of lift measurement. And the simulation results of different situations show that the effect of roll stabilization is improved effectively.

Although the designed sensors are suitable for measuring lift, there are deficiencies:

(1) The voltage value of strain gauges output is in the milivolt level, but the measurement signal is generally in the volt level. So the output voltage signal amplitude is small. In practical application, it is necessary to design an amplifier circuit to meet the engineering requirements. In addition, the anti-interference ability is poor, and the signal wire needs to take measures for shield.

(2) Under the condition of large strain, the nonlinearity of the variant is obvious. Therefore, in the sensor applications, the maximum value should be between 1/2 and 2/3 of the measurement range. In this paper, the design range of lift sensor is a little small. In the actual lift measurement, the maximum range should be taken as the upper limit, so as to avoid the nonlinear influence of the sensor.

For the lift feedback control system of the fin stabilizer, the following aspects need to be further studied:

(1) One of the focuses of future work will be on the design and application of full bridge circuit sensor ship fin stabilizer to measure dynamic lift. And the linear output, sensitivity and temperature compensation of the sensor will also be studied. On the basis of satisfying the dynamic lift detection of fin stabilizer, the signal to noise ratio should be improved. In addition, one of the future work is that suitable metal materials gauges will be selected to improve the temperature compensation.

(2) Furthermore, the influence of the lower support point on the lift measurement will be analyzed, and the quantity correction method of lift measurement will be provided, which can further improve the accuracy of lift measurement. A sensor with high reliability and long life will be developed by improving the lift sensor.

(3) A large number of simulation experiments and tank experiments should be studied, and the experimental data should be further analyzed. For the nonlinear and uncertain relationship between the measured lift by the support point and the actual lift of the fin, the effective solution should be proposed. In order to overcome the disadvantages of lift feedback and fin angle feedback, a fusion technique which can give full play to both advantages will be proposed.

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