Adaptive roll stabilization of ships by U-tanks based on a novel control strategy

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Abstract: The application of passively-controlled U-tanks which have frequency adaptive function was introduced in this paper. The model of air-liquid flow in the U-tanks was established. On the basis of the model, relationship between the sectional area of air valve and fluid flow characteristics was researched. The optimized fluid motion was controlled by continuously turning the section area of air duct, which is calculated in generalized predictive control strategy. In this way, the fluid flow can be tuned to reduce the ship's roll with the wave disturbance frequency varying. Simulation studies showed that the novel strategy can achieve more effective roll reduction compared with traditional intermittent control strategy.

Key Words: ship motion, anti-roll tank, adaptive control, generalized predictive control, passively-control

Introduction

Excessive roll motion not only increases fuel consumption of ships, but also makes working on deck hazardous, affecting the efficiency and safety of the crew [1]. There are several types of roll control devices including fins, gyro stabilizers, free surface tanks and U- tanks. These devices have intrinsic strengths and weaknesses [2, 3]. While active fins are easy to control and give good stabilization in all sea states, they are not effective at low speeds. Anti-roll tanks have the advantage of being able to function when the ship is not underway [4].

Anti-roll tanks generally fall into two categories-free surface tanks and U-tanks [1, 5]. Free surface tanks are simple rectangular containers with a central obstruction such as a baffle, to impede the flow of water. U-tube tanks consist of two separate tanks connected by a water conduit at the base and an optional air conduit in the roof, with valves or baffle plates in the conduits to alter the fluid flow [1].

Anti-roll tanks operates on the following principle: as the ship rolls, the fluid in the tanks moves with it. For a passive tank, the fluid should move with the same frequency as the rolling motion, but having a phase lag. The vessel's kinetic energy is transformed into kinetic and potential energies of the tank fluid [2]. For a passive tank, both the ship and the tank fluid will move with the frequency of the waves [2, 8]. This frequency will depend on the sea state, but the response of the ship will be much more severe if the exciting moment is at the ship's natural roll frequency [2]. So the most common design procedure for passive tanks is to choose the geometry so as to give a natural frequency chose to that of the vessel [1, 2, 5]. While this strategy is effective in reducing the roll at resonance, it can actually increase the roll amplitude at other wave frequency. This is a particular problem at low wave frequencies when the ship is rolling slowly [1]. The other, the vessel's natural roll frequency will depend on loading conditions and other factors, and does not necessarily take the design value [2]. Then the passive tanks will no longer be correctly tuned. Changing the depth of water will alter the tank's natural frequency, but it is more practical to adjust a valve or baffle plates in the conduit [1]. While tanks of this type are already available, the concept is currently only applied to U-tanks. This type of U-tanks are named as passively-controlled anti-roll tanks. A little power is used to adjust the valve or baffles so that the liquid flow can be controlled, and the tank can achieve effective roll reduction in extended frequencies [5,6].

Installing the valve system in the air channel can avoid the high forces and water hammer effects, although the compressibility of the air adds extra complexity to the control system[1]. Attention will be paid to the control problem of tank stabilizers of this type, which is named pneumatical-passively-controlled U-tank in the paper.

The intermittent flow control strategy is usually used in passively-controlled tank stabilizers. The intermittent flow control strategy will attempt to maximize the roll reduction by trapping as much water as possible on the side opposes the ship's roll. Thus, as the ship rolls to port, the majority of the water will be held in the starboard tank with the valves closed and the valves only opened close to the maximum port roll angle, thereby transferring the water into the port roll angle. But U-tanks of this type will have degraded performance if the ship roll frequency is not significantly less than the tank natural frequency. So a prerequisite of this strategy is that there is sufficient time to allow the majority of the water to be transferred in a reasonable fraction of the roll period [1]. In contrast to the intermittent strategy, the continuous flow strategy will involve various opened area of valves in order to modify the flow proportionally. The estimated natural frequency of ship's roll will be available for use as a feedback signal to control the opened area of valves, allowing the tank natural period to be tuned to the current roll period.

The model of air-liquid flow in the U-tanks was established in the paper. On the basis of the model, effect of the opening area of valves in air channel to the complex

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fluid flow was researched. A predictive control strategy was used to estimate optimized parameters of fluid motion in real time. And the opening area of the air valve can be continuously controlled in order to adjust the fluid flow in the tank, so as to achieve more effective roll reduction in extended disturbing frequencies.

Model of Ship and passively-controlled anti-roll tank system

2.1 Model of Ship-Tank System

Equipped with U-tanks, the ship's roll and the tank fluid flow affect each other. The coupled system can be described by six degrees of freedom equation [5,7]. For simplification, this work treats the ship and tank as a single coupled system ignoring little impact by other degrees of freedom motion. The differential equations of the ship rolling system with tank stabilizer is described as

$$\begin{cases}\ddot{\phi}+2v_s\omega_s\dot{\phi}+\omega_s^2\phi+\alpha\ddot{\delta}+\beta\delta=\dfrac{1}{J_s}K_w \\ \ddot{\delta}+2v_t\omega_t\dot{\delta}+\omega_t^2\delta+\eta\ddot{\phi}+\sigma\phi=\dfrac{1}{J_t}\Delta PS_0R\end{cases} \quad (1)$$

Where, ϕ is ship roll angle. δ is tank fluid angle. J_s is the ship roll inertial moment including the liquid masses. J_t is the liquid motion mass inertial moment. v_s , v_t express the damp coefficients respectively. ω_s , ω_t express the connatural frequencies of ship roll and the liquid vibration respectively. α , η and β , σ represent the coupled inertial moment and the coupled resume moment coefficients of the ship and the tank. K_w is the disturbance moment of the wave roll. R is the horizontal distance from the center of the side tank to the longitudinal center plane of the ship. S_0 is the sectional area of the free liquid. ΔP is the pressure difference of compressible air over both side of the tank.

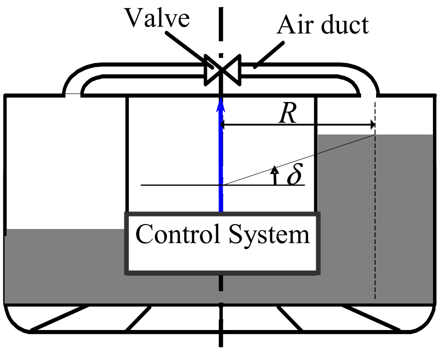

Fig.1 Structure of U-tank

2.2 Model of the Compressible Air Dynamic

When the liquid has a oscillation motion, the air between the liquid surface and the top wall of the tank can be compressed or expanded, which is known as a changeful compressibility process. Subscript number 1 and 2 was used to represent the parameters variates of the left and the right rank respectively, and the air state in both side of the tank can be obtained by the air state equation of a changeful process,

$$P_i v_i^{\gamma}=C_a \quad (i=1,2) \quad (2)$$

Where $v_i=\dfrac{V_i}{m_i}$ is the specific volume of the air. V_i is the air volume above the water and m_i is the mass of the air. C_a is a constant determined by the initial conditions. γ is the adiabatic air constant. In each side of the tank the rate of change in pressure can then be written as

$$\frac{\partial P_i}{\partial t}=\frac{\partial P_i}{\partial V_i}\dot{V}_i+\frac{\partial P_i}{\partial m_i}\dot{m} \quad (i=1,2) \quad (3)$$

According to Eq.(2),

$$\frac{\partial P_i}{\partial V_i}=-\gamma C_a\frac{m_i^{\gamma}}{V_i^{\gamma+1}} \quad (i=1,2) \quad (4)$$

$$\frac{\partial P_i}{\partial m_i}=\gamma C_a\frac{m_i^{\gamma-1}}{V_i^{\gamma}} \quad (i=1,2) \quad (5)$$

and

$$\dot{m}=A\sqrt{\frac{\gamma}{\gamma-1}P_h\rho_h(\frac{P_e}{P_h})^{2/\gamma}[1-(\frac{P_e}{P_h})^{(\gamma-1)/\gamma}]} \quad (6)$$

Where P_h and P_e are the high and low pressure in each side of the valve respectively. $\rho_h=\dfrac{m_h}{V_h}$ is the air density of the higher pressure side; A is the sectional area of the valve. Then the pressure difference between the two sides of the tank is

$$\Delta P=P_1-P_2 \quad (7)$$

As expressed by Eq.(1), the fluid oscillation motion can be assumed to a second order motion when the sectional area of the valve is settled. The ARMA model of fluid oscillation can be described as

$$\delta_k=-a_{21}\delta_{k-1}-a_{22}\delta_{k-2}+b_{21}\phi_{k-1}+b_{22}\phi_{k-2}+b_{20} \quad (8)$$

The model coefficients can be found by the recursive least squares algorithm. The damped frequency ($\hat{\omega}_{td}$), non-damped natural frequency ($\hat{\omega}_t$) and the damped coefficient can be calculated from

$$\hat{\omega}_{td}=\frac{1}{\Delta t}\arccos(\frac{a_{21}}{2e^{0.5\ln(-a_{22})}}) \quad (9)$$

$$\hat{\omega}_t^2=\hat{\omega}_{td}^2+\frac{[\ln(-a_{22})]^2}{4(\Delta t)^2} \quad (10)$$

$$\hat{v}_t=\sqrt{1-(\frac{\hat{\omega}_{td}}{\hat{\omega}_t})^2} \quad (11)$$

Where Δt is the sampling time. Using the proposed method, the fluid oscillation parameters can be off-line identified with respect to different sectional area of the valve, and the relationship curve between A and $\hat{\omega}_t$ as well as to $\hat{v}_t$ can be obtained.

3. Control strategy for ship's stabilization by passively-controlled anti-roll tank

The block diagram of the passively-controlled anti-roll tank system is shown in figure 2. The ship-tank dynamics system is included in the dash dot line square. The control signals are produced by the tank controller according to the ship roll motion state as well as the liquid flow.

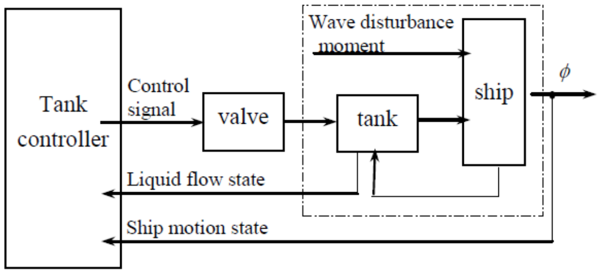

Fig.2 Block diagram of passively-controlled tank stabilizer

3.1 Intermittent flow control strategy

The success of the intermittent flow control strategy is dependent on the correct timing of the opening of the valves, and this will be determined by the flow characteristics of the tank. For an oscillating U-tank with a well-defined natural frequency the timing is relatively straightforward if the next ship roll period, and hence the time of the next peak, can be adequately estimated [1].

In order to ensure the flow of tank 'straddle' the ship's rolling peak giving an effective phase lag of 90 degrees, passively PD control strategy can be used for intermittent flow control strategy of the tank stabilizer system. $\beta = k_1\ddot{\phi} + k_2\dot{\phi}$ is the open signal of the valve, which is consisted of ship's rolling velocity and acceleration. The fluid can move from the left side of tank to the right side when $\beta > 0$ and it is contrary when $\beta < 0$. When the sign of β changed, open the valve and the direction of fluid flow will be reversed. In order to keep the liquid stay in one side of the tank which the ship moves upwards, the valve is closed when the flow-rate drops zero or the column being filled is almost full.

3.2 Continuous flow control strategy

The equation shown in Eq.(1) represents the ship roll motion with a U-tank, which can be known as a second order motion model with two freedom of degree. The wave disturbance K_w is coloured noise which can be obtained by a second order molding filter $G_w(\mathrm{s})$:

$$\frac{K_w(s)}{w(s)} = G_w(\mathrm{s}) = \frac{k_1\mathrm{s}}{\mathrm{s}^2 + k_2\mathrm{s} + k_3} \tag{12}$$

Where, k_1,$k_2$and k_3 are the coefficients being correlative with the height of wave, the heading and velocity of the ship. $w(t)$ is a white noise.

Transform the first equation shown of Eq.(1) to a Laplace form:

$$(s^2 + 2\nu_s\omega_s s + \omega_s^2)\phi(s) + \left(\alpha s^2 + \beta\right)\delta(s) = \frac{1}{J_s}K_w(s) \tag{13}$$

Combining Eq.(12) to Eq.(13) ,The CARIMA model of the ship roll motion which controlled by the tank can be obtained by a z-transform operation

$$A(z^{-1})\phi(k) = B(z^{-1})\delta(k) + C(z^{-1})w(k)/(1-z^{-1}) \tag{14}$$

That is

$$\overline{A}(z^{-1})\phi(k) = B(z^{-1})\Delta\delta(k) + C(z^{-1})w(k) \tag{15}$$

where: $A(z^{-1}) = 1 + a_{11}z^{-1} + a_{12}z^{-2} + a_{13}z^{-3} + a_{14}z^{-4}$;

$B(z^{-1}) = b_{10} + b_{11}z^{-1} + b_{12}z^{-2} + b_{13}z^{-3} + b_{14}z^{-4}$;

$C(z^{-1}) = c_{10} + c_{11}z^{-1} + c_{12}z^{-2} + c_{13}z^{-3} + c_{14}z^{-4}$;

$\Delta\delta(k) = \delta(k) - \delta(k-1)$;

$$\begin{aligned}&\overline{A}(z^{-1})\\&= (1-z^{-1})A(z^{-1})\\&= 1 + \overline{a}_{11}z^{-1} + \overline{a}_{12}z^{-2} + \overline{a}_{13}z^{-3} + \overline{a}_{14}z^{-4} + \overline{a}_{15}z^{-5}\end{aligned};$$

$\overline{a}_i = a_i - a_{i-1}$, $(1 \le i \le 4)$; $\overline{a}_{15} = -a_{14}$。

$\{w(k)\}$ is a white noise series, and

$$E\{w(k)\} = 0, \quad E\{w^2(k)\} = 1$$

The Diophantine equation can be introduced after the CARIMA model of the system was built:

$$C(z^{-1}) = \overline{A}(z^{-1})E_j(z^{-1}) + z^{-j}G_j(z^{-1}) \tag{16}$$

Where

$$E_j(z^{-1}) = 1 + e_{j,1}z^{-1} + \cdots + e_{j,j-1}z^{-(j-1)}$$

$$G_j(z^{-1}) = g_{j,0} + g_{j,1}z^{-1} + \cdots + g_{j,4}z^{-4}$$

According to the adaptive control theory, the relationship of j step forward predictive ship roll angle can be expressed as

$$\phi(k+j) = \frac{G_j(z^{-1})}{C(z^{-1})}\phi(k) + \frac{F_j(z^{-1})}{C(z^{-1})}\Delta\delta(k+j-1) + E_j(z^{-1})w(k+j) \tag{17}$$

Eq.(16) can be converted to the other Diophantine equation, shown as follow

$$F_j(z^{-1}) = B(z^{-1})E_j(z^{-1}) = L_j(z^{-1})C(z^{-1}) + z^{-j}H_j(z^{-1}) \tag{18}$$

Where

$$L_j(z^{-1}) = l_{j,0} + l_{j,1}z^{-1} + \cdots + l_{j,j-1}z^{-(j-1)}$$

$$H_j(z^{-1}) = h_{j,0} + h_{j,1}z^{-1} + \cdots + h_{j,3}z^{-3}$$

The polynomial $F_j(z^{-1})$ was divided into two parts in Eq.(18), the frontal part represents the future influence to the ship by the tank, and the latter one is for past.

Set

$$\Delta\delta_f(k-1) = \frac{\Delta\delta(k-1)}{C(z^{-1})}, \quad \phi_f(k) = \frac{\phi(k)}{C(z^{-1})} \tag{19}$$

According to Eq.(17) to Eq.(19), the output as ship rolling of the "ship-tank" system predictive model has a matrix format as follow:

$$Y(k+j) = L\Delta U(k) + H\Delta U_f(k-j) + GY_f(k) + EW(k+j) \tag{20}$$

Where

$Y(k+j) = [\phi(k+1), \phi(k+2), \cdots, \phi(k+N)]^T$ is the future output vector of the ship roll;

$U(k) = [\delta(k), \delta(k+1), \cdots, \delta(k+N-1)]^T$ is the future output vector of the desired liquid flow;

$Y(k) = [\phi(k), \phi(k-1), \cdots, \phi(k-4)]^T$ is the past output vector of the ship roll;

$Y_f(k) = [\phi_f(k), \phi_f(k-1), \cdots, \phi_f(k-4)]^T$;

$\Delta U(k) = [\Delta\delta(k), \Delta\delta(k+1), \cdots, \Delta\delta(k+N-1)]^T$ is the excursion increment of the future liquid flow;

$\Delta U(k-j) = [\Delta\delta(k-1), \Delta\delta(k-2), \cdots, \Delta\delta(k-4)]^T$ is the excursion increment of the past liquid flow;

$\Delta U_f(k-j) = [\Delta\delta_f(k-1), \Delta\delta_f(k-2), \cdots, \Delta\delta_f(k-4)]^T$;

$W(k+j)=[w(k+1),w(k+2),\cdots,w(k+N)]^T$ is random white noise vector;

$$L=\begin{bmatrix} l_{1,0} & 0 & \cdots & 0 \\ l_{2,1} & l_{2,0} & \cdots & 0 \\ \vdots & \vdots & & \vdots \\ l_{N,N-1} & l_{N,N-2} & \cdots & l_{N,0} \end{bmatrix}, \quad H=\begin{bmatrix} h_{1,0} & h_{1,1} & h_{1,2} & h_{1,3} \\ h_{2,0} & h_{2,1} & h_{2,2} & h_{2,3} \\ \vdots & \vdots & \vdots & \vdots \\ h_{N,0} & h_{N,1} & h_{N,2} & h_{N,3} \end{bmatrix},$$

$$G=\begin{bmatrix} g_{1,0} & g_{1,1} & \cdots & g_{1,4} \\ g_{2,0} & g_{2,1} & \cdots & g_{2,4} \\ \vdots & \vdots & & \vdots \\ g_{N,0} & g_{N,1} & \cdots & g_{N,4} \end{bmatrix}, \quad E=\begin{bmatrix} 1 & 0 & \cdots & 0 \\ e_{2,1} & 1 & \cdots & 0 \\ \vdots & \vdots & & \vdots \\ e_{N,N-1} & e_{N,N-2} & \cdots & 1 \end{bmatrix}$$

Where, $L\in R^{N\times N}$, $H\in R^{N\times 4}$, $G\in R^{N\times 5}$, $E\in R^{N\times N}$;

To optimize the control performance, the objective function was chosen as follow

$$J=E\left\{\sum_{j=1}^{N}\left\{\phi^2(k+j)+[r\delta(k+j-1)]^2\right\}\right\} \tag{21}$$

Where N is the predictive length. If the sampling period is T_0, the predictive time is NT_0. r is the weight coefficient for limiting the excursion of the liquid flow considering the volume is limited in both side of the tank.

Transform Eq.(21) to a matrix form:

$$J=E\left\{\left[Y^T(k+j)QY(k+j)\right]+U^T(k)RU(k)\right\} \tag{22}$$

Where

$Q=\mathrm{diag}[1,1,\cdots,1]$ is the output weight matrix of the ship roll;

$R=\mathrm{diag}[r,r,\cdots r]$ is the output weight matrix of the liquid flow excursion;

According to Eq.(8):

$$\delta(k)=\frac{B_2(z^{-1})}{A_2(z^{-1})}\phi(k) \tag{23}$$

Where $A_2(z^{-1})=1+a_{21}z^{-1}+a_{22}z^{-2}$, $B_2(z^{-1})=b_{20}+b_{21}z^{-1}+b_{22}z^{-2}$

So the control problem of the "ship-tank" system can be converted to multivariable optimization problem as following:

$$\hat{\eta}=\arg\min_{\eta\in\Omega} J \tag{24}$$

$$\eta=[a_{21},a_{22},b_{20},b_{21},b_{22}]$$

Restrictions are:

$$\eta\in\Omega$$

Ω is a sequence space composed of vary parameters in Eq.8 corresponding to different valve opening, which can be obtained from the U-tank vibration test. Each array of parameters can make sure that the system is stable. The optimal parameters can be evaluated by using an augmented lagrangian multiplier method from Eq.24. The natural frequency $\hat{\omega}_t$ and damp coefficient $\hat{v}_t$ can be calculated according to Eq.(9) to Eq.(11), which then can be used as a set-point in a position controller to adjust the opening of valve.

Recursive least square method was used to estimate the parameters in Eq.(15) for achieving an adaptive adjustment to the control system when the ship's sailing state changed.

$$\begin{cases} \hat{\theta}(k)=\hat{\theta}(k-1)+K(k)\left[\Delta\phi(k)-\hat{\varphi}(k)\hat{\theta}(k-1)\right] \\ K(k)=P(k-1)\hat{\varphi}(k)\left[\lambda+\hat{\varphi}^{\mathrm{T}}(k)P(k-1)\hat{\varphi}(k)\right]^{-1} \\ P(k)=\dfrac{1}{\lambda}\left[I-K(k)\hat{\varphi}(k)\right]P(k-1) \end{cases} \tag{25}$$

Where

$$\hat{\varphi}^{\mathrm{T}}(k)=[-\Delta\phi(k-1),-\Delta\phi(k-2),\cdots,-\Delta\phi(k-4),\Delta\delta(k-1),\Delta\delta(k-2),\cdots,\Delta\delta(k-5),\hat{w}(k-1),\hat{w}(k-2),\cdots,\hat{w}(k-4)]$$

$$\hat{\theta}=[\hat{a}_{11},\hat{a}_{12},\hat{a}_{13},\hat{a}_{14},\hat{b}_{10},\hat{b}_{11},\hat{b}_{12},\hat{b}_{13},\hat{b}_{14},\hat{c}_{11},\hat{c}_{12},\hat{c}_{13},\hat{c}_{14}]^{\mathrm{T}}$$

$\hat{w}(k)$ can be shown as a filter as follow:

$$\hat{w}(k)=\phi(k)-\hat{\phi}(k|k)=\phi(k)-\hat{\varphi}^{\mathrm{T}}(k)\hat{\theta}(k)$$

4. Simulations and Analysis

A Ro-Ro ship has been procured to assist in the development of the roll stabilization system. The parameters of the ship is shown as following

Length: L=195.30m.

Width: B=25.6m.

Height: 8.70m(To the main board)

Tonnage: D=19655.9t.

Initial metacentric height: GM=2.282m.

Natural roll period: T_s=14.1s.

Aiming at this ship, the coefficients of the tank stabilizer is optimized according to the method mentioned in reference [9]. The geometry of U-tank which decide the tank coefficients are introduced in reference [1]. In order to insure the passively-controlled tank has a more extended adjustable range, the tank's natural frequency should be higher than the ship's natural frequency, close to the largest wave disturbance frequency. The total mass of the tank is 707t, which is 3.6 percents of the tonnage of the ship. When the valve is totally open, the natural frequency $\omega_t=0.64\mathrm{rad/s}$, and the damp coefficient v_t =0.3.

The sectional area of wave A is limited at $0\sim 0.5\mathrm{m}^2$. Using Eq.(12)- Eq.(15) to identify the natural frequency and damp parameters, and the fluid flow parameters varied with the open size of the valve were shown in figure.3 and figure.4. The natural frequency of tank increases with the increase of valve opening. And the damping coefficient of tank decreases with the increase of valve opening.

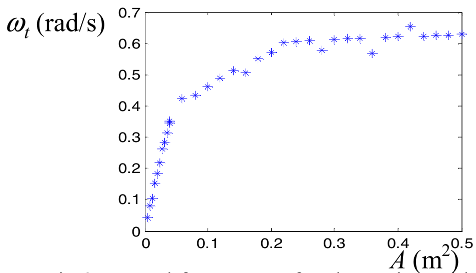

Fig.3 Natural frequency of tank varying with opening area of valve

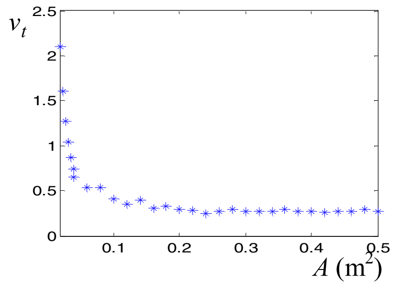

Fig.4 Damping coefficient of tank varying with opening area of valve

In order to test the roll reduction performance of the tank stabilizer with different wave disturbance frequency, choosing the ship's heading angle to wave as 30°, 90° and 145° respectively. The simulation was carried out with irregular wave height of 6m and 3m. And the velocity of the ship is 18kn. The wave disturbance signal were made according to P-M single parameters spectrum. Figure 5 to figure 10 show the simulation results. And the roll reduction performance of the tank with all kinds of waves were shown in table 1and table 2.

In order to study the valve's modification in real time under different control strategy, variable V is employed as following:

$$V = (A_0 - A) / A_0 \tag{26}$$

Where A_0 is the sectional area of the valve which is totally opened. Then V represents the relative opening area for the valve. The valve is totally closed when $V = 1$ and the valve is totally open when $V = 0$.

The results for the intermittent flow control strategy are shown in figure 5 to figure 7, table 1 and table 2. The valve is opened and closed twice per roll period, and consisting with only two states which are closed and open. The strategy achieved good roll reduction performance for a wave height of 6m. But the performance of the intermittent flow control strategy deteriorated markedly for the wave height of 3m. The results illuminate that the intermittent flow tank can't achieve satisfying performance in all rolling conditions, especially in high frequency.

The results for the continuous flow control strategy are shown in figure 8 to figure 10, table 1 and table 2. The predictive length N=5 and weight coefficient r=0.5. It shows that the opening size of the valve varying continuously with the fluid flow in the tank. Comparing with intermittent flow control strategy, the operational frequency and amplitude of the valve was decreased obviously, which has reduced the wear and tear of the valve in great degree. The continuous control strategy achieves more effective roll reduction under different ship sailing state compared with intermittent control strategy. In high frequency, the continuous flow tank achieved fair effectiveness.

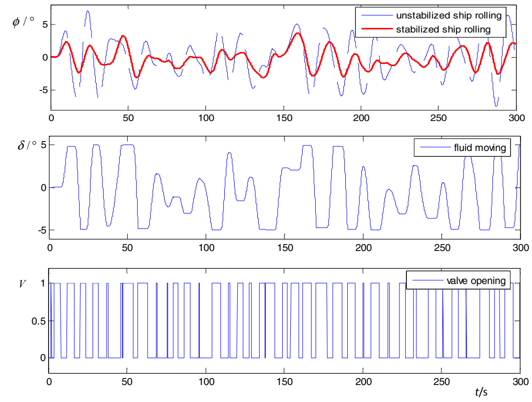

Fig.5 Control characteristics of intermittent flow control strategy （wave height=6m，encounter angle=45°，ship velocity=18kn）

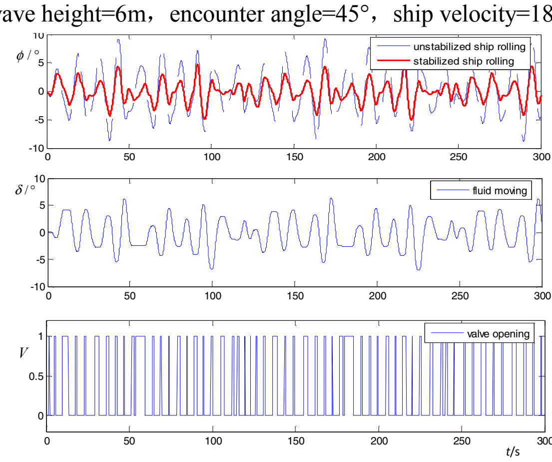

Fig.6 Control characteristics of intermittent flow control strategy （wave height=6m，encounter angle=90°，ship velocity=18kn）

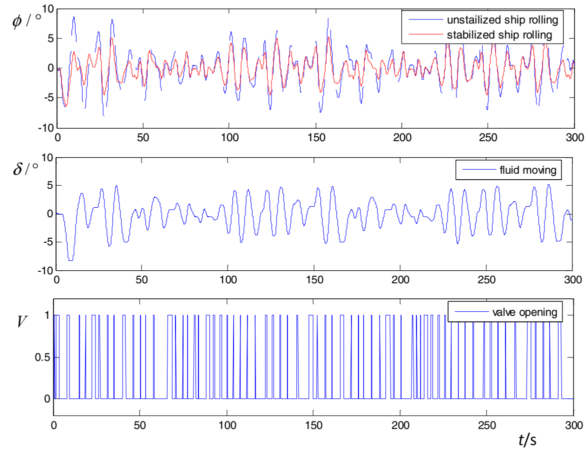

Fig.7 Control characteristics of intermittent flow control strategy （wave height=6m，encounter angle=135°，ship velocity=18kn ）

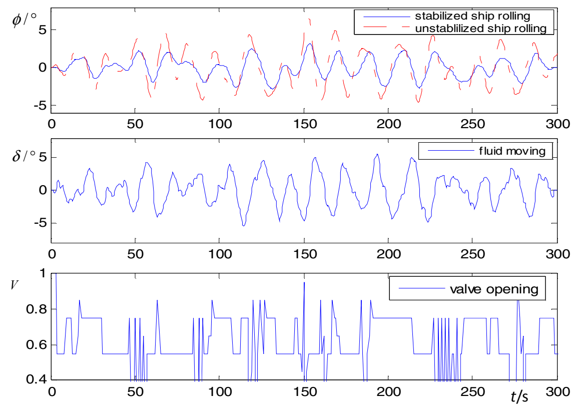

Fig.8 Control characteristics of continuous flow control strategy

（wave height=6m， encounter angle=45°, ship velocity=18kn）

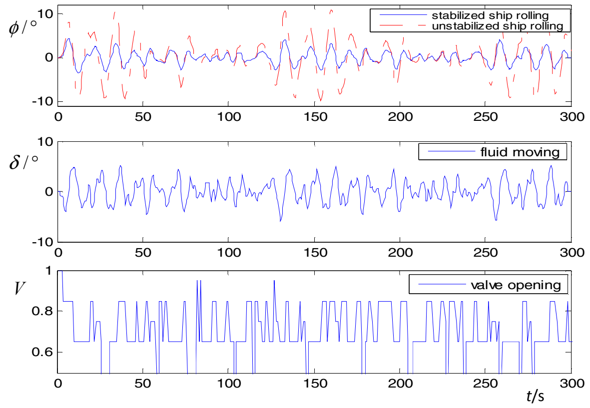

Fig.9 Control characteristics of continuous flow control strategy （wave height=6m， encounter angle=90°， ship velocity=18kn）

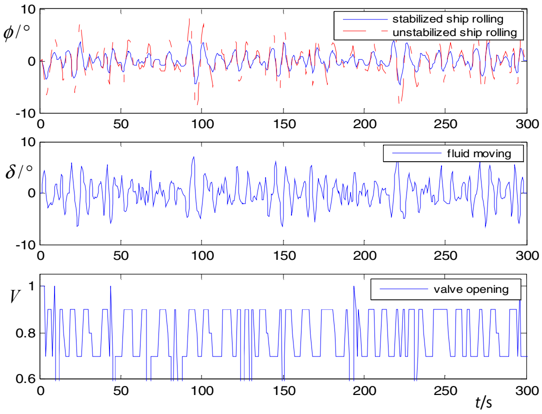

Fig.10 Control characteristics of continuous flow control strategy

（wave height=6m， encounter angle=135°, ship velocity=18kn）

Table 1 Simulation Results under wave height of 6m

Wave height (m)	Encounter angle(°)	No roll reduction	Intermittent flow strategy		Continuous flow strategy	
		Standard deviation (°)	Standard deviation (°)	Anti-rolling performance	Standard deviation (°)	Anti-rolling performance
6	45	2.8075	1.6814	42.07%	1.4970	46.68%
	90	3.3713	1.4063	58.29%	1.2285	63.55%
	135	2.3775	1.0016	57.54%	1.0825	54.47%

Table2 Simulation Results under wave height of 3m

wave height (m)	Encounter angle(°)	No roll reduction	Intermittent flow strategy		Continuous flow strategy	
		Standard deviation (°)	Standard deviation (°)	Anti-rolling performance	Standard deviation (°)	Anti-rolling performance
3	45	1.0893	0.5779	46.25%	0.5369	50.71%
	90	1.0725	0.7624	27.74%	0.6401	40.24%
	135	0.2341	0.1940	17.55%	0.1578	32.59%

Conclusion

The model of air-liquid flow in the U-tanks was established in the paper. The coefficient such as damp and natural frequency of the tank is affected by the open size of the air valve. The relationship between the open size of the valve and the characteristics of fluid flow in tank is obtained based on the model established. The fluid flow parameters of the tank can be controlled by this relationship, which can be utilized to reduce the wave disturbance for ship.

A continuous flow control strategy to the valve of the air duct was achieved. The optimized parameters of fluid flow in tank were calculated by utilizing GPC strategy in real time. Simulation results show that the proposed continuous control strategy achieve more effective roll reduction in extended frequencies compared with intermittent flow control strategy. And the operational frequency of the valve, which is adjustable to the wave disturbance, is reduced obviously compared with intermittent control strategy.

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