Changes in the Hydrodynamic Characteristics of Ships During Port Maneuvers

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KEY WORDS: Port navigation, Shallow water, Computational fluid dynamics, Model test, Autonomous surface ship, Hydrodynamic forces and moments, Maneuverability, Course stability, Collision avoidance

ABSTRACT: To reach a port, a ship must pass through a shallow water zone where seabed effects alter the hydrodynamics acting on the ship. This study examined the maneuvering characteristics of an autonomous surface ship at 3-DOF (Degree of freedom) motion in deep water and shallow water based on the in-port speed of 1.54 m/s. The CFD (Computational fluid dynamics) method was used as a specialized tool in naval hydrodynamics based on the RANS (Reynolds-averaged Navier-Stoke) solver for maneuvering prediction. A virtual captive model test in CFD with various constrained motions, such as static drift, circular motion, and combined circular motion with drift, was performed to determine the hydrodynamic forces and moments of the ship. In addition, a model test was performed in a square tank for a static drift test in deep water to verify the accuracy of the CFD method by comparing the hydrodynamic forces and moments. The results showed changes in hydrodynamic forces and moments in deep and shallow water, with the latter increasing dramatically in very shallow water. The velocity fields demonstrated an increasing change in velocity as water became shallower. The least-squares method was applied to obtain the hydrodynamic coefficients by distinguishing a linear and non-linear model of the hydrodynamic force models. The course stability, maneuverability, and collision avoidance ability were evaluated from the estimated hydrodynamic coefficients. The hydrodynamic characteristics showed that the course stability improved in extremely shallow water. The maneuverability was satisfied with IMO (2002) except for extremely shallow water, and collision avoidance ability was a good performance in deep and shallow water.

Received 13 February 2022, revised 22 March 2022, accepted 8 April 2022
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Nomenclature

| Symbol | Description |
|--------|-------------|
| O| Earth-fixed coordinate |
| o−xy| Body-fixed coordinate |
| ψ| Heading angle |
| U| Ship speed |
| β| Drift angle |
| a| Surge velocity |
| v| Sway velocity |
| r| Yaw rate |
| u| Surge acceleration |
| v| Sway acceleration |
| r| Angular acceleration |
| Iyz| Yaw mass motion of inertia |
| xG| Longitudinal center of gravity |
| m| Ship mass |
| n_p| Propeller revolution |
| t| Thrust deduction factor |
| x_p| Longitudinal coordinate of propeller position |
| y_p| Lateral coordinate of propeller position |
| w_p| Wake coefficient in maneuvering motion |
| K_T| Propeller thrust open water characteristic |
| J_p| Propeller advanced ratio |
| l_R| Steering resistance deduction factor |
| α_H| Rudder increase factor |
| x_R| Longitudinal coordinate of the rudder position |
| F_N| Rudder normal force |
| U_R| Resultant rudder inflow velocity |
| α_R| Effective inflow angle to the rudder |
| w_R| Longitudinal inflow velocity |
| v_R| Lateral inflow velocity |
1. Introduction

When a ship navigates in waters with varying depths, such as in-port, straits, and channel, it encounters danger from sinkage and trim, as well as changes in maneuvering characteristics. The interaction influences the behavior of ships in shallow water, and the flow velocity is increased by the gap between the ship bottom and the seabed. Furthermore, because of Bernoulli’s law, the pressure field in that area decreased, causing fluctuations in the hydrodynamic forces and moments, changing the attitude of ships, potentially resulting in unexpected collisions.

Research on the influence of shallow water has been conducted in recent years. Duarte et al. (2016) classified the water level in ship maneuvering based on the draft to water depth ratio \( \lambda = h/T \) of a ship as deep water: \( \lambda > 3.0 \); medium-deep water: \( 1.5 < \lambda < 3.0 \); shallow water: \( 1.2 < \lambda < 1.5 \); and extremely shallow water: \( \lambda < 1.2 \). In addition, the authors provided some aspects affected by shallow water such as resistance, trim, checking, counter tuning ability, turning diameter, and rate of turn. Jachowski (2008), Yun et al. (2014), and Lee and Hong (2017) examined the ship squat, also known as sinkage and trim in shallow water. It started from horizontal 3-DOF and non-linear unified seakeeping. The maneuvering time-domain using a numerical simulation were then performed using the hydrodynamic coefficients and moments obtained from the model test in the case of static drift test in deep water. The maneuvering characteristics as course stability and maneuvering simulation were then performed using the hydrodynamic coefficients obtained to evaluate the stability and maneuverability of the ship in deep and shallow water. A simple collision avoidance situation was also executed to investigate the effect of shallow water on the collision avoidance ability.

2. Maneuvering Simulation Model

2.1 Objective

The candidate ship used in this study was an autonomous surface ship equipped with a twin rudder, twin propeller, and a skeg in the center of the stern. Fig. 1 shows a scale model of 1/11 attaching a
twin-rudder and skeg used in the experimental study. Fig. 2 displays the boundary domain of a full-scale ship which consists of a bared hull and skeg, which is applied to numerical study in CFD. Table 1 lists the main particulars of the hull, propeller, and rudder. The propeller and rudder specifications were used in the maneuvering simulation.

2.2 Test Condition

The full-scale ship was assumed to operate at a low speed of 1.54 m/s in shallow water areas. The static drift, circular motion, and combined circular motion were simulated for a full-scale ship in deep and shallow water with water depth ratios of 2.0, 1.5, and 1.2, respectively, using a numerical simulation in CFD. Furthermore, an experiment of scale-model with the length between perpendiculars of 2.0 m was performed in a towing tank in deep water to verify the accuracy of numerical simulation. A corresponding 1.54 m/s in the speed of full-scale ship, the static drift test of model-scale was towed at 0.47 m/s. The numerical simulation in CFD was conducted under the constraints test listed in Table 2. All damping coefficients were obtained for deep and shallow water.

For this experimental study, the ship was towed by a carriage at a given speed. The hydrodynamic forces and moments were measured using two load cells located near the bow (LC#1) and stern (LC#2), where the distance to the midship was \( x_1 \) and \( x_2 \), respectively. Furthermore, all motions were restrained throughout the experimental performance. Fig. 3 presents an experimental installation of the ship.

For a numerical study, the virtual captive model test with the test condition in Table 2 was simulated using the Ansys fluent program to calculate the hydrodynamic forces and moments. In Table 2, and are the drift angle \( \beta \) and \( \dot{\beta} \) dimensionless yaw rate, respectively.

The continuity and momentum equations were applied as governing equations assuming that the flow was incompressible (Mai et al., 2020). As an ITTC (International Towing Tank Conference) recommendation for CFD (ITTC, 2011), the boundary domain size was large enough to avoid backflow excluding the bottom side, and it was generated under shallow water conditions.

A hybrid mesh that includes the tetrahedral, hexahedral, and prism mesh type were adopted for mesh generation, as shown in Fig. 4. Table 3 lists the analytic method for the CFD simulation. The boundary conditions were set for each domain corresponding to its physical characteristics, such as the inlet assigns the pressure-inlet, the outlet sets the pressure-outlet, the top and sides are symmetry, the ship defines the no-slip wall, and the bottom specifies the no-slip wall to consider the influent of shallow water. The \( k-\omega \) SST (Shear stress transport) turbulence model is applied extensively to predict the hydrodynamic forces and moments on a maneuvering ship because of several advantages in terms of its accuracy and time calculation (Quérard et al., 2008). The volume of fluid and open channel flow are...
Table 3 Analytic methods for the CFD simulation

| Item                | Description                                 |
|---------------------|---------------------------------------------|
| Boundary conditions |                                             |
| Inlet               | Pressure-inlet                              |
| Outlet              | Pressure-outlet                             |
| Ship                | No-slip wall                                |
| Bottom              | No-slip wall                                |
| Top, Sides          | Symmetry                                    |
| Two-phase           | Volume of fluid                             |
| Circular motion     | Multiple reference frames                   |
| Turbulence model    | $k-\omega$ Shear stress transport (SST) model|
| Method              | Semi-implicit method for pressure-linked equations (SIMPLE) |
| Algorithm           |                                             |
| Gradient            | Least squares cell based                    |
| Interpolation method for pressure | Second order upwind                      |
| y+                  | 300                                         |
| Meshing             | Number of elements about 4.5 million        |
| Type of mesh        | Tetrahedral, hexahedral, and prism          |

and body-fixed ($a_{\theta,\phi}$) were set to determine the 3-DOF motion of the horizontal plane, as shown in Fig. 5. The earth-fixed coordinate defined the ship trajectory, orientation angle, and body-fixed ($a_{\theta,\phi}$) defined the equation of motion and external force acting on the ship. The origin of the ship was located at the intersection of the midship, centerline, and draft.

The maneuvering motion of 3-DOF in the horizontal plane was written for surge, sway, and yaw. Eq. (1) expresses the equation of motion of the ship based on the maneuvering modeling group (MMG) model (Yasukawa and Yoshimura, 2015). The external force on the right is the component of hull force, thrust, and rudder force, denoted by the letters $H$, $P$, and $R$.

$$
m(\ddot{u} - \nu r + \omega x_r^2) = X_H + X_P + X_R$$
$$
m(\ddot{v} + \omega r + \nu x_r^2) = Y_H + Y_P + Y_R$$
$$
I_r \ddot{r} + m a_{\theta,\phi} (\ddot{v} + \nu r) = N_H + N_P + N_R
$$

The mathematical model of the hull force, thrust, and rudder force on the right side of Eq. (1) was formulated in Eqs. (2), (3), and (4), respectively, for twin-screw naval ship (Kim et al., 2021; Khanfir et al., 2011). The model of the hull forces Eq. (2) was determined as the regression formula of simulation results. In which the damping coefficients regarding sway velocity $X_{\nu,\nu}$, $Y_{\nu,\nu}$, $N_{\nu,\nu}$, and $N_{\nu,\nu}$ were estimated from the static drift test where the sway velocity was generated, the damping coefficients with respect to the yaw angular velocity $X_{\phi,\phi}$, $Y_{\phi,\phi}$, $N_{\phi,\phi}$, and $N_{\phi,\phi}$ were obtained from the circular motion test, where the yaw angular velocity was given, and the coupling damping coefficients relative to sway velocity and yaw rate $X_{\nu,\phi}$, $Y_{\nu,\phi}$, $N_{\nu,\phi}$, and $N_{\nu,\phi}$ were taken from combined circular motion with drift, where both sway velocity and yaw angular velocity were generated. Furthermore, the added mass coefficients $X_{\theta}$, $Y_{\theta}$, $N_{\theta}$, and $N_{\theta}$ were determined from the pure surge, pure sway, and pure yaw test reported by Kim et al. (2021).

$$
X_H = X_{\nu,\nu} \ddot{u} + X_{\nu,\nu} x_r^2 + X_{\nu,\nu} r^2 - R
$$
$$
Y_H = Y_{\nu,\nu} \ddot{v} + Y_{\nu,\nu} x_r^2 + Y_{\nu,\nu} r^2
+ Y_{\nu,\nu} x_r^2 r + Y_{\nu,\nu} r^2
$$
$$
N_H = N_{\nu,\nu} \ddot{r} + N_{\nu,\nu} x_r^2 + N_{\nu,\nu} r^2
+ N_{\nu,\nu} x_r^2 r + N_{\nu,\nu} r^2
$$

$$
X_{\theta} = (1 - t) p \partial D \left[ (\nu^2)^2 \cdot K_{\theta}^2 (\theta) + (\nu^2)^2 \cdot K_{\phi}^2 (\phi) \right]
$$
$$
Y_{\phi} = 0
$$
$$
N_{\phi} = y_{\phi} (1 - t) p \partial D \left[ (\nu^2)^2 \cdot K_{\theta}^2 (\theta) - (\nu^2)^2 \cdot K_{\phi}^2 (\phi) \right]
$$

$$
X_H = - (1 - t_H) (F_N^N + F_N^S) \sin \delta
$$
$$
Y_H = (1 + a_H) (F_N^N + F_N^S) \cos \delta
$$
$$
N_H = (x_H + a_H \sin \delta) (F_N^N + F_N^S) \cos \delta - y_H (1 - t_H) (F_N^N - F_N^S) \sin \delta
$$
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The parameters mentioned in the thrust model (3) and rudder model (4) are expressed below. The parameter in the formulae was written for the rudder and the propeller on the starboard side (subscript \( R \)), so the formulae of the port side could be obtained by replacing them with subscript \( P \), and the formulae are the same except for the flow straightening coefficient \( (C_p) \) and wake correction coefficient \( (C_w) \). Considering the symmetry of twin-propeller and twin-rudder, the same value of \( C_p \) was used as \( \beta_p \geq 0 \) in the starboard side and \( \beta_p < 0 \) in the port side, and vice versa.

\[
D_p = \frac{1 - w_p}{n^2 D} \tag{5}
\]

\[
F_p = 0.5 \rho A_p \left( U_{in}^2 \right) f_p \sin^2 \gamma_p \tag{6}
\]

\[
U_{in} = \sqrt{w_p^2 + r_p^2}, \quad \alpha_p = \frac{1}{2} \left[ 1 + \left( \frac{8 \rho S c^2}{\pi n^2} - 1 \right)^2 + 1 \right],
\]

\[
\gamma_p = \gamma_p (v + l_p r)
\]

\[
(\beta_p = -(v' + l_p r')).
\]

\[
\gamma_p = \gamma_p \quad \text{when } \beta_p \geq 0, \quad \gamma_p = \gamma_p \quad \text{when } \beta_p < 0
\]

\[
w_p = (1 - w_p) u \quad \text{with } w_p = w_p \exp(-C_p \beta_p)
\]

\[
(\beta_p = -(v' + x_p r')).
\]

\[
C_p = C_p \quad \text{when } \beta_p \geq 0, \quad C_p = C_p \quad \text{when } \beta_p < 0
\]

3. Analysis results

3.1 Verification

The modeling and simulation method in CFD was verified by comparing the results of the sway force and yaw moment of static drift test in the deepwater with the experimental results, as shown in Fig. 6. The simulation results matched well with the experimental method, particularly at slight drift angles (<±12°). The yaw moment differed slightly as the drift angle increased, and it was asymmetrical between the negative and positive drift angles. On the other hand, this did not affect maneuvering analysis because of the required linear coefficients. The comparison demonstrated the accuracy of the simulation method in estimating the hydrodynamic forces and moments on a maneuvering ship.

### Table 4 Static hydrodynamic force model for the hull motion variables

| Test type                  | Model       | Formula                                                                 |
|---------------------------|-------------|-------------------------------------------------------------------------|
| **Static drift**          | Linear      | \[ X_{BD} = X_0 + Y_0 r + Z_0 r', -9 < \beta < 9 \]                     |
|                           | Non-linear  | \[ X_{BD} = X_0 + Y_0 r + Z_0 r', \beta < 0 \]                         |
|                           |             | \[ X_{BD} = X_0 + N_r \]                                                |
| **Circum motion**         | Linear      | \[ X_{BD} = X_0 + Y_0 r, \quad \beta < 0 \]                           |
|                           | Non-linear  | \[ X_{BD} = X_0 + Y_0 r + N_z \]                                       |
| **Combined circum motion**| Non-linear  | \[ X_{BD} = X_0 + Y_0 r + X_z r' + N_z \]                              |
non-linear model, where the non-linearity of the force was predicted at greater motion variables. Non-dimensionalization complies with the prime system of the Society of Naval Architects and Marine Engineers (SNAME), in which non-dimensional forces and moments were written as $F' = F/(0.5pL^2)$ and $M' = M/(0.5pL^3)$, respectively. This model was applied to the ship under deep and shallow water conditions. Fig. 7 presents the results of the static drift test at various water depths. The results were obtained by adjusting the drift angle by $\pm 18^\circ$. As the water depth became shallower, the hydrodynamic forces and moments were greater, particularly in extremely shallow water ($h/T = 1.2$). It increased two times compared to $h/T= 1.5$.

Similar to the static drift test, the hydrodynamic forces and moments in the circular motion test were obtained by changing the yaw angular velocities, as shown in Fig. 8. Fig. 9 shows the results of the combined circular motion with drift by varying drift angles and yaw angular velocities. The hydrodynamic forces and moments increased with decreasing water depth, and the drift angle and yaw angular velocity increased. The hydrodynamic forces and moments increased slightly from deep to medium shallow water but significantly increased in extremely shallow water. These results proved the influence of shallow water on the sway velocity and yaw rate. The increase in hydrodynamic forces and moments in shallow water was caused by an increase in flow velocity and a decrease in pressure through the gap between the ship bottom and the seabed. Fig. 10 shows the increase in flow velocity through the gap between the ship bottom and seabed as water depth becomes shallower. Table 5 lists the damping coefficients in deep and shallow water.
4. Dynamic Simulation

4.1 Course Stability

Stability analysis was performed to evaluate the course stability relative to the course of a ship. It was analyzed based on the linear hydrodynamic coefficients. As mentioned above, the linear coefficients were obtained by analyzing the hydrodynamic forces and moments at a small motion variable as the drift angle was smaller than 6° in the static drift test and the non-dimensional yaw angular velocity was less than 0.3 in the circular motion test. The course stability was examined as a function of the water depth using a sway-damping lever ($C_{s} = N_s / (Y_s - m')$, yaw damping lever ($C_{y} = N_y / (Y_y - m')$), and course stability index ($C = N_s / (Y_s - m') - N_y / (Y_y - m')$). The course stability index showed that it was unstable from deep water to $h/T = 1.5$ and improved in extremely shallow water. The yaw damping lever affected the course stability significantly.

4.2 Maneuverability Simulation

The maneuvering performance of the ship in shallow water was evaluated by simulating the turning circle 35° to the starboard side and zigzag 10°/10° test under an initial speed of 1.54 m/s. It was obtained by solving the 3-DOF equation of motion with the maneuvering coefficients, in which the damping coefficients were determined by the virtual captive model test, as shown in Table 5. The added mass

Table 5 Damping coefficients in deep and shallow water

| HD Coef. | Deep water | $h/T = 2.0$ | $h/T = 1.5$ | $h/T = 1.2$ |
|----------|------------|-------------|-------------|-------------|
| $X_{sv}$ | -1.84E-02  | -3.84E-02  | -6.32E-02  | -1.17E-01  |
| $X_{sv}$ | 2.26E-03   | -1.14E-03  | -1.19E-03  | -2.94E-03  |
| $X_{sv}$ | 6.47E-03   | 1.12E-03   | 1.61E-02   | 2.84E-03   |
| $Y_{s}$  | -7.56E-03  | -1.18E-02  | -1.76E-02  | -3.17E-02  |
| $Y_{s}$  | -4.21E-03  | -9.71E-02  | 1.64E-01   | -1.85E-01  |
| $Y_{s}$  | 9.35E-03   | 1.05E-02   | 1.06E-02   | 1.25E-02   |
| $Y_{s}$  | 3.64E-03   | 5.80E-04   | 1.62E-03   | 1.80E-02   |
| $Y_{s}$  | -2.72E-02  | -3.45E-02  | 7.91E-02   | -7.55E-01  |
| $Y_{s}$  | -2.47E-02  | -5.84E-02  | -1.45E-01  | -3.12E-01  |
| $Y_{s}$  | -3.10E-03  | -4.16E-03  | -6.00E-03  | -1.03E-02  |
| $Y_{s}$  | -1.54E-03  | -1.37E-04  | -3.32E-03  | -1.05E-02  |
| $Y_{s}$  | -1.35E-03  | -1.43E-03  | -1.60E-03  | -2.53E-03  |
| $Y_{s}$  | -2.93E-03  | -3.55E-03  | -4.87E-03  | -9.75E-03  |
| $Y_{s}$  | -3.96E02   | -7.08E-02  | -1.70E-01  | -1.51E-01  |
| $Y_{s}$  | -1.65E-03  | -1.14E-02  | -1.63E-02  | -4.10E-02  |

Fig. 11 Shallow water effects on course stability
coefficients were obtained from Kim et al. (2021), and the interaction coefficients of the rudder model and thrust model were taken from Kim et al. (2021). Figs. 12 and 13 present the results of turning circle and zigzag $10^\circ/10^\circ$ in the deep and shallow water. The turning circle was similar to the previous studies. The turning rate and drift angle were reduced, leading to a larger trajectory than that in deep water. This suggests that the influence of shallow water increased with decreasing water depth to $h/T = 1.2$. It was caused by the increasing hydrodynamic forces acting on the hull as the depth decreased. Therefore, the turning parameters in $h/T = 1.2$ did not satisfy the International Maritime Organization (IMO) (IMO, 2002). For the zigzag $10^\circ/10^\circ$ test, the first overshoot angle was more dominant at $h/T = 1.5, 2.0$, and smaller at $h/T = 1.2$ compared with those of deep water. By contrast, the second overshoot angle decreased gradually with increasing water depth to $h/T = 1.2$. Furthermore, the zigzag maneuver satisfied the IMO (2002) for deep and shallow waters.

The ship collision avoidance was conducted to examine the effect of shallow water on the collision avoidance ability of the ship. The dynamic model was applied to the equation of motion for both the own ship (OS) and target ship (TS) to simulate the ship collision avoidance. A simple scenario based on the encounter situation of the cross-right according to the International Regulations for Preventing Collisions at Sea (COLREGs), was undertaken in this study, as shown in Fig. 14. Collision avoidance was implemented when the owner detected the trajectory of the target ship at a distance of $6L$, as suggested by Fuji’s ellipse model based on the collision risk (Fuji and Tanaka., 1971). The rudder or propeller was then commanded to prevent the collision. In this case, the rudder of the own ship was turned $10^\circ$ when it was a distance of $6L$ from the target ship. Fig. 15 shows the collision avoidance simulation in deep and shallow water at different times. The collision avoidance ability of the ship was expressed well in both deep and shallow water because the target ship could reach a safe area when the own ship touched the trajectory of the target ship. On the other hand, the ability to avoid collisions in shallow water appeared to be superior.
5. Concluding Remarks

This study examined the hydrodynamic characteristics of an autonomous surface ship throughout the port maneuvers. A numerical study in CFD was used to perform the virtual captive model test to measure the hydrodynamic forces and moments in deep and shallow water. In addition, a model test for static drift test in deep water was also carried out to demonstrate the accuracy of the numerical study, and a good agreement was observed between the experimental and numerical study. The numerical results in shallow water confirmed that the hydrodynamic forces and moments increased as water depth became shallower because of the accelerating flow velocity through the gap between ship bottom and seabed. The maneuvering coefficients were determined by distinguishing a linear model from the non-linear model, in which the linear coefficients were achieved from small motion variables.

The hydrodynamic characteristics in deep and shallow water were analyzed by assessing the course stability, maneuvering simulation, and collision avoidance. The course stability was conducted using the linear coefficients. The sway-damping lever decreased gradually while the yaw-damping lever increased as the deep water became shallower, and the course stability improved in extremely shallow water. A maneuvering simulation was obtained by solving the 3-DOF equation of motion with the mathematical models of twin-propeller and twin rudder. This denoted a larger turning circle in shallow water due to increased hydrodynamic forces and moments. Moreover, the parameters of the turning test and zigzag 10°/10° in deep and shallow water were satisfied with the IMO (2002) except for the parameters of the turning test at $h/T = 1.2$. Furthermore, a simple collision avoidance of cross-right situation was executed to investigate the effect of shallow water on the collision avoidance ability. Shallow water looked better than deep water in the collision avoidance ability of the ship, even though the turning ability was better in deep water.

Conflict of Interest

Hyeon Kyu Yoon serves as an editor of the Journal of Ocean Engineering and Technology, but has no role in the decision to publish this article. No potential conflict of interest relevant to this article was reported.

Acknowledgments

This research was supported by Changwon National University in 2021-2022.

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