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Theoretical prediction of running attitude of a semi-displacement round bilge vessel at high speed

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ABSTRACT

Running attitudes of semi-displacement vessels are significantly changed at high speed and thus have an effect on resistance performance and stability of the vessel. There have been many theoretical approaches about the prediction of running attitudes of high-speed vessels in calm water. Most of them proposed theoretical formulations for the prismatic hard-chine planing hull. In this paper, running attitudes of a semi-displacement round bilge vessel are theoretically predicted and verified by high-speed model tests. Previous calculation methods for hard-chine planing vessels are extended to be applied to semi-displacement round bilge vessels. Force and moment components acting on the vessel are estimated in the present iteration program. Hydrodynamic forces are calculated by 'added mass planing theory', and near-transom correction function is modified to be suitable to a semi-displacement vessel. Next, 'plate pressure distribution method' is proposed as a new hydrodynamic force calculation method. Theoretical pressure model of the 2-dimensional flat plate is distributed on the instantaneous waterplane corresponding to the attitude of the vessel, and hydrodynamic force and moment are estimated by integration of those pressures. Calculations by two methods show good agreements with experimental results.

1. Introduction

When a high-speed vessel runs in the high speed region, flow patterns and pressure distributions along the hull surface are dramatically changed, so running attitudes of the vessel are varied with its running speed. Running attitudes of a high-speed vessel have a strong effect on its resistance, stability and manoeuvrability. Therefore, running attitudes of a high-speed vessel should be accurately estimated by model tests or theoretical calculations in its design stage.

There are some researches about vertical plane dynamics and running attitude predictions of high speed vessels. Semi-empirical formulas based on the model tests of prismatic planing hulls are developed by Savitsky [1] and Savitsky and Brown [2]. Martin [3] formulated a mathematical model for the calculation of forces acting on planing crafts, and the model is based on linear strip theory. A planing craft was modeled as a series of strips or impacting wedges. And Zarnick [4] followed and developed Martin's [3] works by using nonlinear strip theory. In Akers's [5] study, Zarnick's [4] formulas were extended to predict the local dynamic pressure for the structural design of planing hulls. Zhao et al. [6] applied the 2D + t theory to solve the steady planing problem of a prismatic hull at high speed. Savander et al. [7] formulated 3D boundary value problem for steady planing surfaces. They mostly treat hard chine planing vessels, so there is little theoretical research for semi-displacement round bilge vessels.

High-speed vessels can be categorized according to the support force or the range of operating speeds. For example, a semi-displacement vessel operates at Froude number 0.5–1.3. The weight of a semi-displacement vessel is mainly supported by buoyancy at low speed, but the hydrodynamic lift becomes larger with increasing speed and supports about 20–30% of the total weight of the vessel at high speed.

In this study, previous running attitude prediction methods for hard chine planing hulls are partially modified and applied to a semi-displacement round bilge vessel. Running attitudes of a semi-displacement vessel are theoretically calculated, and model tests are carried out to verify calculation results. Forces acting on the vessel such as buoyancy force, frictional force and hydrodynamic force are calculated in a present iteration program. Buoyancy and frictional forces are calculated with reference to the change of the wetted shape of the hull in each iteration procedure. Hydrodynamic force is predicted by using 'added mass planing theory' and 'plate pressure distribution method'.

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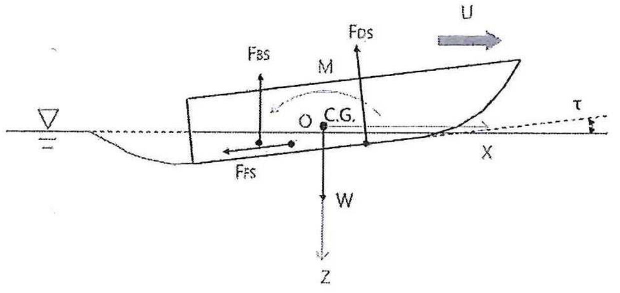

Fig. 1. Coordinate system and force components acting on the vessel.

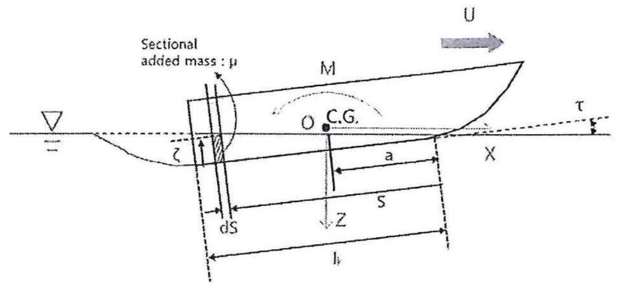

Fig. 2. Sectional added mass for hydrodynamic force calculations.

2. Theoretical approach

2.1. Force and moment equilibriums

The coordinate system and force components acting on the hull are shown in Fig. 1. A high-speed vessel runs at the constant forward speed U, and τ is the trim angle. A body-fixed coordinate O-XYZ is placed at the center of gravity of a vessel. X-axis is parallel to the free surface and positive forward, Z-axis is positive downward, and the pitch angle is positive bow up.

Subscripts DS, BS, and FS respectively indicate hydrodynamic, buoyancy, and frictional forces at the steady state. W is the weight of the vessel. Heave forces and pitch moments should be in equilibrium when the vessel runs at constant forward speed. Force and moment equilibrium equations are shown in Eq. (1).

$$\begin{aligned} W &= F_{DS}\cos\tau + F_{BS} - F_{FS}\sin\tau \\ M_{DS} &+ M_{BS} + M_{FS} = 0 \end{aligned} \tag{1}$$

2.2. Buoyancy force

Buoyancy is obtained by integrating all sectional wetted areas along the keel, and multiplying by the mass density of water and the gravitational acceleration. Buoyancy force and moment are formulated as Eq. (2). A_{sw} is the sectional wetted area, LCG and LCB are respectively the longitudinal center of gravity and buoyancy.

$$\begin{aligned} F_{BS} &= \rho g \int_0^{l_k} A_{sw}\, ds \\ M_{BS} &= F_{BS}\cos\tau \cdot (\mathrm{LCB} - \mathrm{LCG}) \end{aligned} \tag{2}$$

The summation of geometric moments of sectional wetted areas along the keel is divided by the summation of sectional wetted areas in order to obtain the longitudinal center of buoyancy, LCB.

2.3. Frictional force

When sinkage and trim are assumed, wetted surface area under the free surface can be calculated. And the dynamic pressure $1/2\rho U^2$ and the frictional coefficient C_F are multiplied to obtain the total frictional force, as shown in Eq. (3).

$$\begin{aligned} F_{FS} &= \frac{1}{2}\rho U^2 C_F \left[\int_0^{l_k} L_{sw}\, ds\right] \\ M_{FS} &= -F_{FS}\sin\tau\cos\tau \cdot (\mathrm{LCfr} - \mathrm{LCG}) \end{aligned} \tag{3}$$

L_{sw} is the wetted length of each section under the free surface, the longitudinal center of friction LCfr is calculated by dividing the summation of geometric moments of L_{sw} by the summation of L_{sw}.

2.4. Hydrodynamic force – 1st method: added mass planing theory

2.4.1. Added mass planing theory

'Added mass concept' or 'added mass planing theory' is mentioned by some previous researchers such as Wagner [8], Martin [3], Payne [9], and so on. The planing phenomenon of a planing surface was treated to be equivalent to the vertical impact phenomenon of each transverse cross-section in their studies. Under the high frequency free surface boundary condition, damping components vanish, then the lift force on the body is calculated as the time rate of change of momentum due to added mass components.

Hydrodynamic forces acting on a prismatic planing hull are calculated by Martin [3]. Referring to his research, notations are shown in Fig. 2. The value s is the coordinate measured along the keel from foremost immersed point of the keel, and ζ is the component normal to the keel. l_k is the total wetted keel length, a is the value of s at the transverse plane through the center of gravity. And μ is the 2-dimensional added mass of the cross-section at point s.

The normal hydrodynamic force over the entire hull is obtained by integrating the time rate of the momentum of each cross-section along the wetted length of the hull and multiplying by a 3-dimensional effect coefficient $\varphi(\lambda)$. Hydrodynamic force and moment are formulated as Eq. (4).

$$\begin{aligned} F_D &= \varphi(\lambda)\int_0^{l_k} \frac{d}{dt}(\mu\dot{\zeta})\, ds \\ M_D &= \varphi(\lambda)\int_0^{l_k} (a-s)\frac{d}{dt}(\mu\dot{\zeta})\, ds \end{aligned} \tag{4}$$

In Eq. (4), an integral term of hydrodynamic force formulation is the sum of the time rate of change of momentum of all 2-dimensional cross-sections. Therefore, the integral term implies that the flows are generated in the vertical direction of each cross-section, and there are no interactions between adjacent cross-sections. But the real flow is generated in the longitudinal direction of the 3-dimensional body, so the additional corrections are required. In other words, 3-dimensional effect coefficient, $\varphi(\lambda)$ is the correction factor to account for the three-dimensionality of the flow. According to the previous research such as Shuford's [10] report, 3-dimensional effect coefficient is the function of the aspect ratio λ.

Expanding Eq. (4) in the same way as Martin's [3] report, and dropping the second order perturbation terms, the constant hydrodynamic force and moment in the steady state are obtained as Eq. (5).

$$\begin{aligned} F_{DS} &= \varphi(\lambda)U^2 \sin\tau\cos\tau \int_0^{l_k} \frac{\partial\mu}{\partial s}\, ds \\ M_{DS} &= \varphi(\lambda)U^2 \sin\tau\cos\tau \int_0^{l_k} (a-s)\frac{\partial\mu}{\partial s}\, ds \end{aligned} \tag{5}$$

When the sinkage and the trim of a vessel are assumed during the iteration procedure, added masses of each sectional wetted area are estimated by Lewis's [11] method. 3-Dimensional effect coefficient $\varphi(\lambda)$ is calculated as $\lambda/(1+\lambda)$, which is applied to prismatic planing

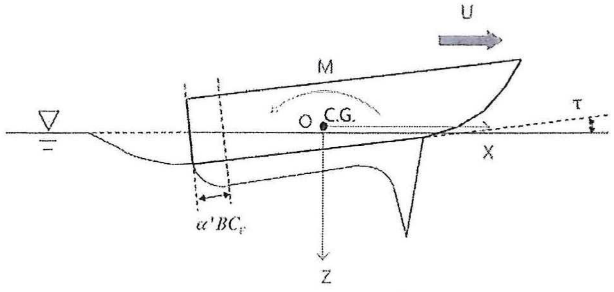

Fig. 3. Pressure distribution on the bottom and transom at high speed.

hull, suggested by Shuford [10]. In the case of the calculation of the planing hull, λ is the non-dimensionalized value of the mean wetted keel length l_k and wetted chine length l_c with the breadth of the hull. But in the present calculation for a semi-displacement vessel, λ is non-dimensionalized value of wetted keel length l_k with the breadth of the hull.

2.4.2. Near-transom correction function

The transom of the high speed vessel is dry at high speed, so the pressure on the transom becomes atmospheric pressure. Garme [12] proposes a near-transom pressure correction function C_{tr} for planing hulls, as shown in Eq. (6). It is a reduction function that has a large gradient aft which decreases towards 0, and approaches 1 at a distance forward.

$$C_{tr} = \tanh\left[\frac{2.5}{\alpha' B C_V}(l_k - s)\right] \quad (6)$$

B is the breadth of the hull, C_V is the Froude number over the breadth of the hull, $U/\sqrt{gB}$. α' is non-dimensional reduction length as shown in Fig. 3.

After the near-transom correction function is applied to buoyancy and hydrodynamic forces, force formulations are replaced with Eqs. (7) and (8). Subscript '*tr,cor*' means the corrected terms by the near-transom correction function.

$$\begin{aligned} F_{BS\_tr,cor} &= \rho g \int_0^{l_k} C_{tr}(s) \cdot A_{sw}\, ds \\ M_{BS\_tr,cor} &= F_{BS\_tr,cor} \cos\tau \cdot (LCB_{\_tr,cor} - LCG) \end{aligned} \quad (7)$$

$$\begin{aligned} F_{DS\_tr,cor} &= \varphi(\lambda) U^2 \sin\tau \cos\tau \int_0^{l_k} C_{tr}(s) \frac{\partial \mu}{\partial s}\, ds \\ M_{DS\_tr,cor} &= \varphi(\lambda) U^2 \sin\tau \cos\tau \int_0^{l_k} C_{tr}(s)(a - s) \frac{\partial \mu}{\partial s}\, ds \end{aligned} \quad (8)$$

Garme [12] proposes the value α' in high speed region ($C_V \geq 2$) as 0.34, which is based on his model tests and published model test data.

2.6. Hydrodynamic force – 2nd method: plate pressure distribution method

If the total pressure distributions on the hull bottom are given, the hydrodynamic force and moment can be directly obtained. There are no experimental or numerical bottom pressure data for semi-displacement vessels, so theoretical pressure distribution formula of 2D flat plate is used for the estimation of hydrodynamic force and moment acting on the vessel.

Pierson and Leshnover [13] used potential theory and conformal transformations to formulate pressure distribution on 2-D flat plate as Eq. (9). P is the pressure, ρ is the mass density of fluid, U is the horizontal velocity of the flat plate, and τ is the trim angle of a flat plate.

$$C_P = \frac{P}{1/2\rho U^2} = 1 - \left[\frac{\xi - \cos\tau}{1 - \xi\cos\tau + \sin\tau\sqrt{1-\xi^2}}\right]^2 \quad (9)$$

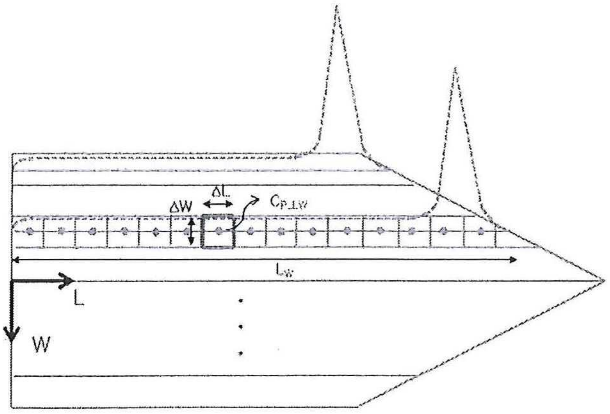

Fig. 4. Longitudinal strips distribution on the waterplane.

ξ is the real part of the conformal transformation function, and is known as the longitudinal position which is non-dimensionalized by the distance between the leading edge and the spray root of the flat plate, have the limits $-1 \leq \xi \leq 1$. Both -1 and 1 respectively correspond to the trailing edge and the spray root of the flat plate.

Longitudinal strips are distributed on the instantaneous waterplane corresponding to the vessel's attitude assumed, and Eq. (9) are applied to each strip as shown in Fig. 4. Then, the stagnation line at the bow and the pressure drop near the transom can be realistically considered.

As shown in Fig. 4, longitudinal strips are divided into many segments in the longitudinal direction for the discretized computation. L_W is the length of each longitudinal strip, ΔL and ΔW is the length and the width of each segment, and C_{P_LW} is the pressure coefficient at each segment.

When the equivalent trim angle of a flat plate is τ_{eq}, the pressure coefficient at one segment, the pressure coefficient at each segment and the total lift are formulated as Eqs. (10) and (11).

$$C_{P\_LW} = 1 - \left[\frac{\xi - \cos\tau_{eq}}{1 - \xi\cos\tau_{eq} + \sin\tau_{eq}\sqrt{1-\xi^2}}\right]^2 \quad (10)$$

$$F_{waterplane} = 1/2\rho U^2 \int_{-B/2}^{B/2}\int_0^{L_W} (C_{P\_LW})\Delta L\, \Delta W \quad (11)$$

Shuford [10] suggested a lift coefficient formula of V-bottom surfaces as Eq. (12). β is the deadrise angle, and Λ is the aspect ratio (beam over wetted length). S is the wetted area bounded by trailing edge, chines, and spray line.

$$\begin{aligned} C_L &= \frac{Lift}{1/2\rho S U^2} = \frac{0.5\pi\Lambda\tau}{\Lambda + 1}\cos^2\tau(1 - \sin\beta) \\ &+ \frac{4}{3}\sin^2\tau\cos^3\tau\cos\beta \end{aligned} \quad (12)$$

When β_m is a mean deadrise angle of the total wetted volume of the vessel, Λ_m is the beam over the instantaneous wetted keel length, the total lift of the vessel can be predicted by Eq. (13). κ is a shape parameter, and is determined as 0.5 by present calculations.

$$\begin{aligned} F_{vessel} &= 1/2\rho S U^2 \cdot \kappa \left[\frac{0.5\pi\Lambda_m\tau}{\Lambda_m + 1}\cos^2\tau(1 - \sin\beta_m)\right. \\ &\left. + \frac{4}{3}\sin^2\tau\cos^3\tau\cos\beta_m\right] \end{aligned} \quad (13)$$

Equivalent trim angle of a flat plate is τ_{eq} is obtained by the equation $F_{vessel} = F_{waterplane}$. The overall concept of plate pressure distribution method is shown in Fig. 5.

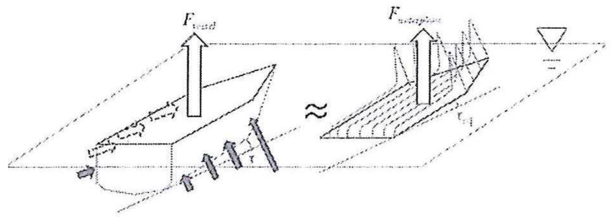

Fig. 5. Equivalent trim angle and pressure distributions on the waterplane.

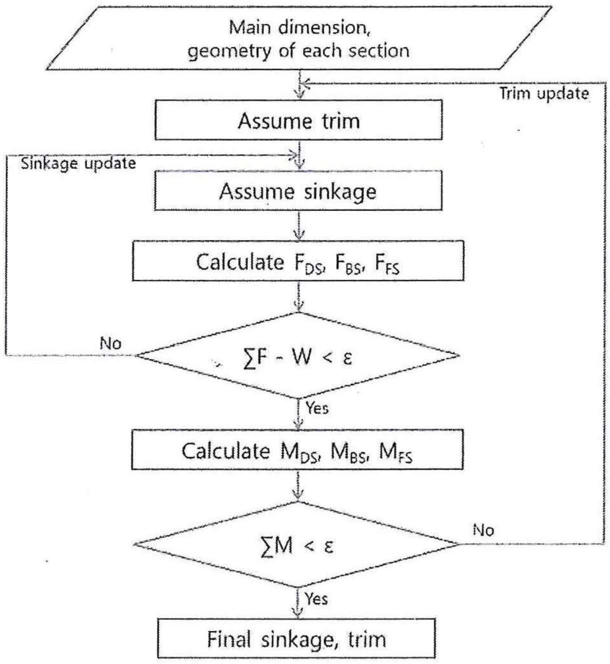

Fig. 6. Flow chart of an iteration program.

2.7. Computational procedure

Running attitudes of a semi-displacement vessel are calculated by a present iteration program. The flow diagram of the present program is shown in Fig. 6.

Sinkage and trim are updated iteratively so that the residues of the force and moment equilibrium equations converge to zero. For example, the trim angle is assumed in the range of 0–5° at intervals of 0.1°, the sinkage is calculated from the force equilibrium equation in each case, and assumed trim and calculated sinkage are substituted into the moment equilibrium equation to check the residue of the moment equation.

Figs. 7 and 8 show one example of calculation results of force and moment by the present program. Fig. 7 shows the vertical force components with assumed trim angles. Pitching moment components and the residues of the moment equations are plotted according to the assumed trim angles in Fig. 8.

In Fig. 7, the weight of the hull is always constant regardless of the attitude of the vessel. Vertical hydrodynamic force gets larger as the trim angle increases. And the hull rises with the increase of trim, the buoyancy force decreases. Therefore the total vertical forces are in equilibrium. The frictional force is relatively so small.

In Fig. 8, when the trim angle increases, the hydrodynamic moment that raises the bow up becomes larger. And in that case, buoyancy moment lets the bow down, because the center of buoyancy moves toward the stern.

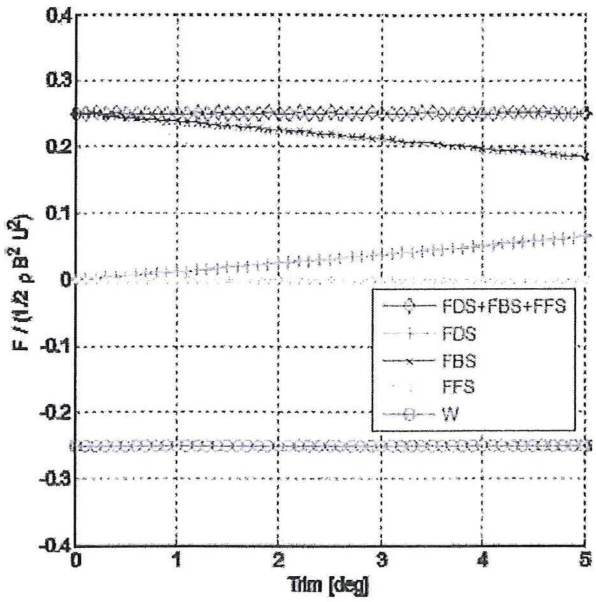

Fig. 7. Force components with assumed trims.

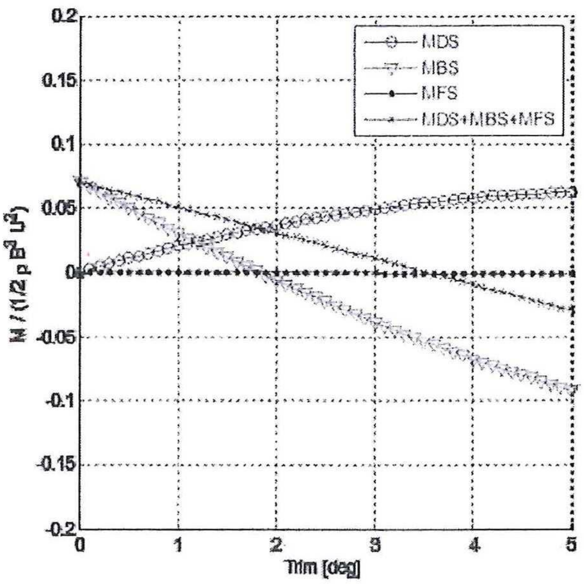

Fig. 8. Moment components with assumed trims.

Although it is true that the buoyancy moment becomes zero when the trim angle is zero, calculated buoyancy moment is positive when the trim angle is zero as shown in Fig. 8. Because the near-transom pressure correction function is always included in the calculation.

In Fig. 8, the trim angle is 3.5° and the sinkage is 0.29% of the length between perpendiculars of the vessel, when the residues of force and moment equilibrium equations converge to zero.

3. Model tests

High speed model tests are carried out to verify the calculation results. Main particulars of the model ship and the high speed towing carriage are shown in this chapter.

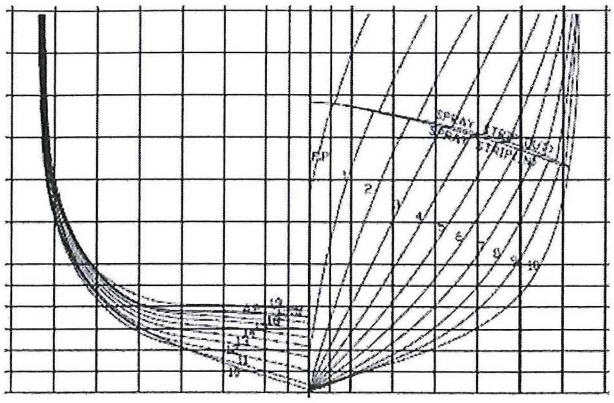

Fig. 9. Body plan of a model ship.

Table 1
Main particulars of a model.

Particulars	Non-dimensional value
L/B	6.21
B/T	2.70
Block coefficient	0.47
Design Froude number	1.19

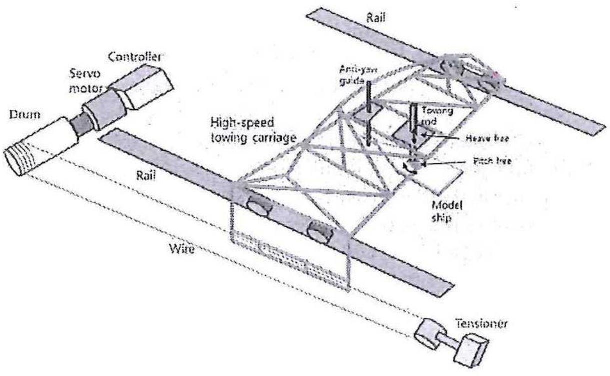

Fig. 10. High speed towing system.

3.1. Model ship

The length between perpendiculars of the model ship is 2 m, and the displacement of it is 36.975 kg. A body plan and main particulars of the model are shown in Fig. 9 and Table 1. The model is a water-jet propelled high-speed vessel, and it is a semi-displacement round bilge type hull.

3.2. High speed towing system

Model tests are performed by a high-speed towing carriage in a towing tank at Seoul National University. The length of the towing tank is 117 m, the beam is 8 m, and the depth is 3.5 m. The mass of the carriage is around 600 kg, the maximum towing speed is 10 m/s, and towing is accomplished by wires drawn by a servo motor. Fig. 10 shows a schematic view of the high speed towing system.

Trim angles and thrust angles of high-speed vessels are changed considerably at high speed, so towing devices that can tow the model in the thrust direction are developed and used in high speed model tests.

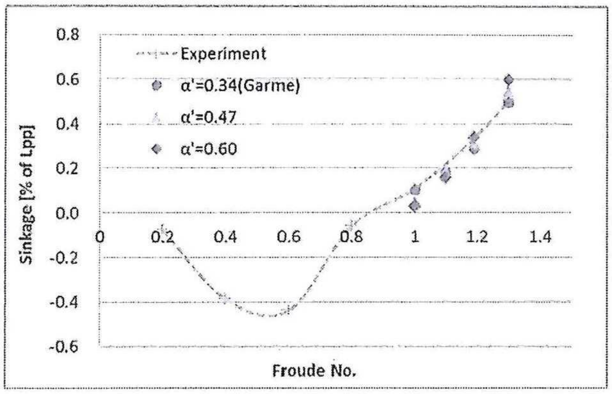

Fig. 11. Measured and calculated sinkages according to reduction length α'.

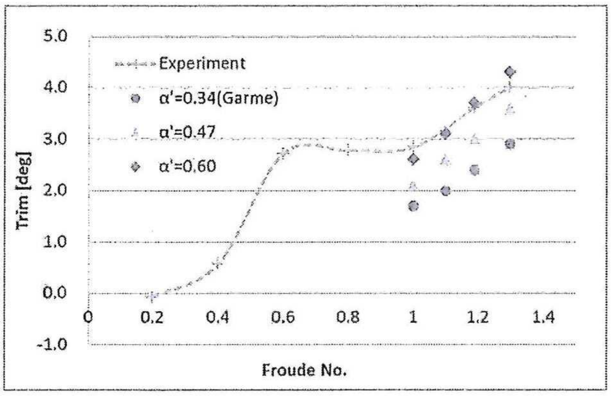

Fig. 12. Measured and calculated trims according to reduction length α'.

4. Comparison of calculations with model test results

4.1. Comparison of measured data with calculations by added mass planing theory

At first, running attitudes of a semi-displacement round bilge vessel at Froude number 1.0–1.3 are calculated when the reduction length α' in the near-transom pressure correction function in Eq. (6), is 0.34, based on Garme's [12] hard-chine planing model test data.

At low speed, water may not be perfectly separated on the transom, so the different pressure correction function is required to be applied to low speed cases. Actually, α' for the vessel with low speed is more than 0.34, and is about 0.70–0.90 in Garme's [12] hard-chine vessel data. The operating speed for semi-displacement vessels is between low speed and high planing speed of hard-chine planing vessels. So running attitudes are calculated in cases that a' is changed up to 0.60. Figs. 11 and 12 show that 0.60 is most suitable value as the reduction length α' of a present semi-displacement vessel.

4.2. Comparison of measured data with calculations by plate pressure distribution method

Sinkages and trims are estimated again by plate pressure distribution method. Figs. 13 and 14 show comparison of calculations by two methods with experimental results. Both calculations are in good agreement with measured data, even though sinkages by plate pressure distribution method are a little overestimated.

An example of calculated force and moment components by two methods is shown in Figs. 15 and 16. Those components are calculated when the sinkage and the trim are fixed as measured values at design

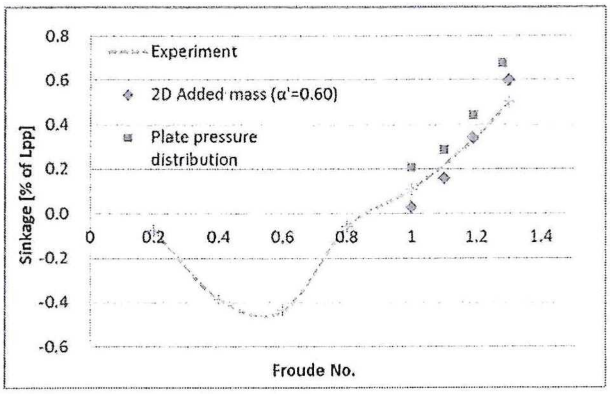

Fig. 13. Measured and calculated sinkages by added mass planing theory and plate pressure distribution method.

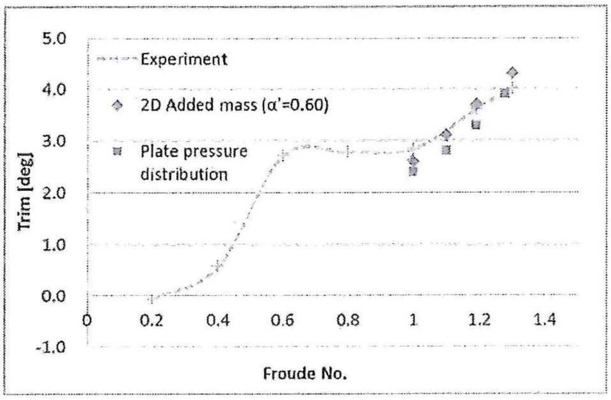

Fig. 14. Measured and calculated trims by added mass planing theory and plate pressure distribution method.

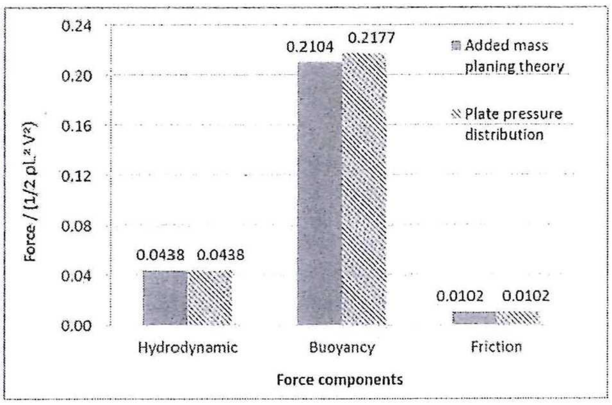

Fig. 15. Calculated force components by added mass planing theory and plate pressure distribution method.

Froude number 1.19. Force and moment components obtained by two methods are similar.

5. Conclusions

Running attitudes of a semi-displacement round bilge vessel in calm water are calculated and compared with experimental results in this study.

Previous research for prismatic planing vessels is extended to be applied to semi-displacement round bilge vessels. Forces acting on

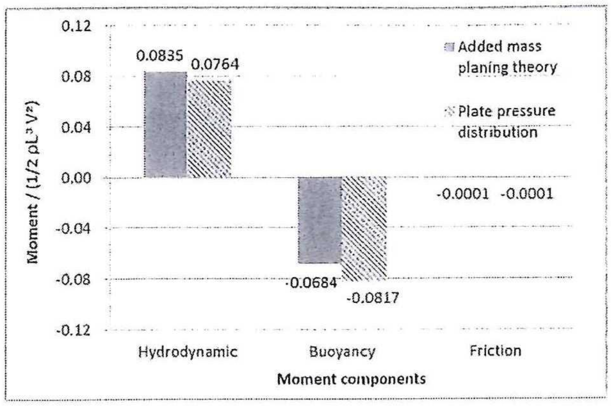

Fig. 16. Calculated moment components by added mass planing theory and plate pressure distribution method.

the vessel such as the buoyancy force, the frictional force and the hydrodynamic force are theoretically estimated. The buoyancy and the frictional force are calculated by using the information of instantaneous wetted hull shapes. The hydrodynamic force is calculated by added mass planing theory and plate pressure distribution method. Sinkages and trims are calculated by the present iteration program.

In calculations by added mass planing theory, there are some differences between calculated trims and measured trims when the near-transom correction function for hard-chine planing vessels is used. Reduction length in near-transom correction function is modified in order that calculated sinkages and trims agree with measured data.

Theoretical plate pressure distributions are used to estimate the hydrodynamic force and moment acting on a semi-displacement vessel. 2-Dimensional plate pressure formulations are distributed on longitudinal strips of the instantaneous waterplane corresponding to the attitude of the vessel. Hydrodynamic force and moment by using plate pressure distribution method are similar with those by using added mass planing theory. And calculated sinkages and trims are in good agreement with experimental results.

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