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Experimental evaluation of forward speed effect on maneuvering hydrodynamic derivatives of a planing hull

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Abstract

The calculation of unknown hydrodynamic derivatives of the equations of motion is the first step to estimate ship maneuverability and dynamic stability. These derivatives can be obtained theoretically, experimentally and numerically. Despite the development of the oblique towing model test to measure the hydrodynamic derivatives of displacement ships, limited experimental results are available for hydrodynamic derivatives of high speed crafts and speed dependency of the hydrodynamic derivatives is not understood well. In this paper a systematic series of model tests is described to determine the effect of forward speed on hydrodynamic derivatives of a monohull planing craft and the variations of the hydrodynamic derivatives by forward speed are derived. According to the results, hydrodynamic derivatives of planing hull are dramatically changed by variations of forward speed. Moreover, it is not possible to introduce a constant hydrodynamic derivative in the all the ranges of drift angle. Thus, the method of known constant hydrodynamic derivatives is not applicable to the simulation of planing craft maneuvering and variable hydrodynamic derivatives should be applied.

Introduction

Equations of motion based on captive model tests are currently the most powerful and flexible means to evaluate ship maneuvering. Although these methods are expensive when compared to free running model tests, they allow the simulation of complicated ship maneuvering motions immediately after the hydrodynamic derivatives are evaluated.

The linear ship maneuvering theory is applied to study the effect of motion characteristics in the fixed control condition or the turning capabilities of directional stable ships in linear range. The captive model test is applied to evaluate the hydrodynamic derivatives of these equations. It should be emphasized that the linear theory is not capable to describe the sudden maneuvering acts of most vessels or the maneuvering of ships with no directional stability. At present, there is no comprehensive analytical method to model the nonlinear ship maneuvering. Generally the coefficients of the coupled equations of motion must be determined experimentally and the equations are solved in the time domain.

The earliest attempts of modeling ship maneuvering were focused on the derivation of the mathematical equations for ship displacement and showed the complicated dependence of turning and course-keeping. The formulae deriving from these models are the bass of current maneuvering theories (Davidson & Shiff, 1946). In the subsequent years notable advances were made in numerical and experimental ship maneuvering studies. The methods to solve the ship maneuvering equations were studied and the method proposed in these later papers is almost the same as the earlier one (Eda, 1971; Crane 1973). The introduction of digital computers in the 1970s facilitated for the first time the simulation of ship maneuvering in the time domain and by the 1980s ship designers preferred computer simulations to model tests as they allowed to reduce the costs. This was a great checkpoint in the development of digital computing of ship maneuvering. Many notable publications were written in this period (Doerffer, 1980; Miller et al., 1984; Biancardi, 1988). In all these simulations the hydrodynamic coefficients of ship maneuvering were based on databases gathered from captive model tests, free running radio controlled models and sea trials.

Different mathematical models have been proposed for ship maneuvering. Each of these models is appropriate for certain means of simulation. The mathematical models of ship maneuvering, especially the MMG model, were studied. If the hydrodynamic forces are known in each time interval, no mathematical model is needed and the equations of motion would be sufficient. Computational fluid dynamic methods are based on the determination of hydrodynamic forces in each time step and on the equation of motion. Mathematical models may be applied to determine the hydrodynamic forces but the expressions used are not simple, due to the unsteady effects of the flow. In mathematical models it is assumed that hydrodynamic forces are a function of only velocity and acceleration (Yoshimura, 2005).

Polynomial functions of velocity and acceleration are applied to describe the hydrodynamic forces and moments. The coefficients of these polynomials, called hydrodynamic derivatives, may be obtained as follows:

1. Captive model test such as oblique towing, circulating arm and planar motion mechanism;
2. Computational methods;
3. System identification of free running model test results or full scale trials;
4. Hydrodynamic coefficient databases.

Published research on the measurement of hydrodynamic derivatives of planing hulls in calm water by captive model test is scarcely available, despite the method for vessel displacement being well developed. The most notable research in this field is based on the measurement of the hydrodynamic forces exerted on a small model of planing hull by oblique towing and PMM in a towing tank. According to the results, the measurements are a function of the running attitude of the vessel at high speeds. It is thus essential to introduce other modes of motion in the mathematical model of planing hull maneuvering rather than motions in the horizontal plane (Ikeda, Katayama & Okumura, 2000).

The model test method is applied to study the maneuvering of planing hull and the model is captive in all degrees of the freedom. The forces and moments exerted on the model are measured along three axes. The draught, trim and forward speed are varied systematically during the tests. A regression mathematical model is developed based on the model test results and different planing hull maneuvers are simulated (Plante et al., 1998).

It is assumed that all the added mass coefficients are frequency independent in the evaluation of ship maneuvering in calm water. One of the challenges of experimental model tests of planing hull maneuvering consists in determining whether or not the measured parameters of the planar motion mechanism test are a function of oscillation frequency. According to the results, velocities and accelerations of sway and yaw do not depend on frequency but the added mass terms are dependent on forward speed. Thus the dependency of coefficients on forward speed should be considered in the mathematical models. A mathematical model with six degrees of freedom for planing hull motions was developed in previous works (Toxopeus, Keuning & Hooft, 1997). The authors considered the rudder forces, added mass and damping forces. Generally a six degrees of freedom model would be essential to study the effects of dynamic trim changes, oscillating heel motion, broaching and couple dynamics of the vessel in roll and yaw.

A free running maneuvering model test of planing hull can describe the effect of various parameters that are influential to vessel maneuverability (Deakin, 2008). It is demonstrated that although the ship maneuverability could be directly assessed by a free running model, the scaled model presents problems in relation to the propeller and propulsion arrangements. A captive model test would therefore be the best experimental approach.

The running attitude of planing craft changes rapidly during maneuvering. The running attitude effect on maneuvering is usually neglected for displacement hulls. It should be noted that the changes in running attitude during maneuvering are very small in most displacement hull cases. A mathematical model for high speed craft maneuvering was developed based on experimental results and the maneuvering of a trimaran hull was studied (Katayama et al., 2009).

Different hydrodynamic phenomena appear around planing hulls at high speeds, such as running attitude changes including draught and dynamic trim, and dynamic changes in hull wetted surface. Thus classical ship hydrodynamic theories cannot be applied to planing hulls. Although computational fluid dynamics methods are very promising in this field, there are still concerns about their validity and experimental data on planing hydrodynamics would be crucial. On the other hand, model test methods of planing hull are still not well understood well and it is essential to study the experimental fluid dynamic methods of planing hulls.

The Savitsky method describes the behavior of planing hull with acceptable accuracy; however, it should be emphasized that the forward speed of planing hulls has increased too much when compared to the 1960s. The Savitsky method is therefore unable to describe the transverse stability at high speeds that today is of great concern. Several directional instabilities occur at high speeds due to transverse dynamic stability loss. As an example, a high speed planing boat may capsize due to excessive parametric roll, broaching may take place or its dynamic stability may decrease in oblique waves. Even sudden roll motion of a planing hull in calm water at high speeds is a familiar phenomenon. The transverse stability of planing hulls at high speeds has been studied. In displacement hulls at low speeds the GZ-curve of stability remains constant with speed, but in high speed crafts, draught and trim change significantly as speed increases. At high speeds the GZ-curve is thus completely a function of forward speed (Wang et al., 2014).

The development of modern marine engines and advances in lightweight structures and materials in hull manufacturing have recently allowed the speed of planing hulls to exceed 100 km/h. A major part of the hull would be above the water level at high speeds. Various dynamic instabilities, such as complete transverse stability loss, porpoising, cork-screwing, and Dutch roll occur in planing hulls. It is essential to study these instabilities and present applicable solutions to avoid them in order to guarantee the safety of navigation at sea. The conditions of these instabilities have studied experimentally (Katayama & Habara, 2011).

Scale effects in model tests of planing hulls with small models may be a great concern. Scale effects on wetted area, frictional resistance and pressure forces on a small model have been studied and model tests have been repeated with different prismatic hulls. The analysis of hull resistance shows that transom pressure resistance makes up a great portion of the total planing hull resistance. Moreover, the wetted surface of the hull decreases slightly in small models of large trim angle (Katayama et al., 2002).

Design considerations to prevent transverse dynamic instabilities at high speeds have been established for planing hulls in which the forward speed exceeds 25 knots. The criteria are based on full scale measurements (Blount & Codega, 1992). The dynamic effects are prominent in high speed monohulls; therefore, the velocities and accelerations around different coordinate axes are coupled (Bowles & Blount, 2012).

The added mass, damping and restoring moment coefficients of prismatic planing hulls in planing speeds is determined based on experimental results. The hydrodynamic coefficients are expressed as a function of breadth, deadrise, forward speed, trim, and transom draught. These parameters may be calculated by the Savitsky method. The effect of appendages on hydrodynamic coefficients was also investigated. Various ranges of forward speed, drift angle, roll angle, and trim were tested and hydrodynamic coefficients including Y_v, Y_ψ, K_v, K_ψ, N_v and N_ψ were calculated (Brown & Klosinski, 1990; 1991).

There are few published data on the scale effects of ship hydrodynamic derivatives, which still constitute one of the major challenges in the study of ship hydrodynamics. The worst case would be represented by planing hull hydrodynamics and scale effects. The scale effects of hydrodynamic derivatives for displacement of hulls have been investigated. Basically, the model tests would be influence by the Reynolds number According to results, no considerable scale effects are expected for linear hydrodynamic coefficients. Nonetheless, it should be noted that wave making effects were negligible during the model tests due to the low forward speed (Renilson & Mak, 1998).

Great efforts have been made in order to formulate the damping coefficients of sway and yaw motion for displacement of hulls. According to results, it seems that the hydrodynamic coefficients of displacement hulls do not depend on forward speed. Only a few experiments have been carried out on planing hull hydrodynamic coefficients, but it should be noted that forward speed is the main cause of changes in wetted area, trim, rise, and other hydrodynamic parameters. Therefore the hydrodynamic derivatives would be different for the various regimes. The oblique towing test of planing hull has been therefore performed in different speed ranges.

Mathematical models of ship maneuvering are developed for displacement type hulls but there is no exact mathematical model for planing hull dynamic motions. The running attitude (forward speed, running trim, rise and heel) would affect the planing hull maneuverability and dynamic stability. It should be noted that conventional ship maneuvering methods are developed based on displacement hull where heel, trim and rise changes are not included except for special cases of planing hulls. The planing hull maneuvering model should include the couple equation of motion.

Equations of motion and hydrodynamic coefficients

Ship dynamic motions are usually expressed in a body fixed coordinate system. The range of motions are calculated and then transformed to the earth fixed coordinate system to calculate the maneuvering path. The body fixed coordinate system and ship motions are defined in Figure 1. Surge, sway, roll and yaw motions are the most important in the description of planing hull maneuvering. The four degrees of freedom ship coupled equations of motion are as follow:

$$\begin{aligned} m(\dot{u} - rv) &= X \\ m(\dot{v} - ru) &= Y \\ I_{xx}\ddot{\phi} &= K \\ I_{zz}\dot{r} &= N \end{aligned} \qquad (1)$$

where m is the ship mass, I_{xx} and I_{zz} are the mass moment of inertia around the x and z axis, respectively, u, v and r are velocity components along x and y and the rate of turn around the z axis, respectively and ϕ is the ship's roll angle. X, Y, K and N are surge force, sway force, roll moment and yaw moment respectively.

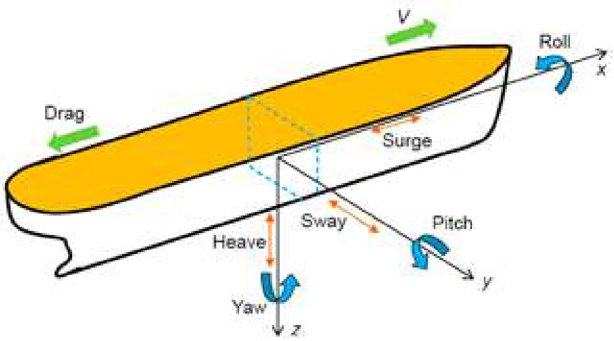


Figure 1. Coordinate system and ship motions

In order to study the dynamic stability of ships in maneuvering, linear equations of motion would be adequate. Forces X and Y and moments N and K in 4 degrees of freedom equations of motion are assumed to be a function of ship motion velocity and accelerations (Lewis, 1987). They can therefore be expressed as:

$$\begin{aligned} X &= F_x(u, v, \dot{u}, \dot{v}, r, \dot{r}, \phi, \dot{\phi}) \\ Y &= F_y(u, v, \dot{u}, \dot{v}, r, \dot{r}, \phi, \dot{\phi}) \\ K &= F_\phi(u, v, \dot{u}, \dot{v}, r, \dot{r}, \phi, \dot{\phi}) \end{aligned} \qquad (2)$$

The above functions should be expressed in algebraic form. The Taylor expansion of a multi variable function provides a good estimation for forces and moments. The Taylor expansion of a single variable function of x around equilibrium point x_1 is:

$$f(x) = f(x_1) + \delta x \frac{\mathrm{d} f(x)}{\mathrm{d} x} + \frac{\delta x^2}{2!} \frac{\mathrm{d}^2 f(x)}{\mathrm{d} x^2} + \cdots \qquad (3)$$

where the derivatives are evaluated at $x = x_1$. The equation can be linearized if the changes in independent variable are small:

$$f(x) = f(x_1) + \delta x \frac{\mathrm{d} f(x)}{\mathrm{d} x} \qquad (4)$$

The concept of small changes in variables is completely consistent with ship dynamic stability, which is defined based on the likelihood that a small perturbation in speed, for a ship in equilibrium conditions, would grow larger or not. For a multi variable function the Taylor expansion is as follows:

$$\begin{aligned} Y = F_y(u_1, v_1, \dot{u}_1, \dot{v}_1, r_1, \dot{r}_1, \phi_1, \dot{\phi}_1) + (u - u_1)\frac{\partial Y}{\partial u} + \\ + (v - v_1)\frac{\partial Y}{\partial v} + \cdots + (\dot{\phi} - \dot{\phi}_1)\frac{\partial Y}{\partial \dot{\phi}} \end{aligned} \qquad (5)$$

where the subscript 1 stands for values of independent variables in initial equilibrium conditions. The initial values are set to $\dot{u}_1 = \dot{v}_1 = r_1 = \dot{r}_1 = \dot{\phi}_1 = \phi_1 = 0$ in order to determine the stability of motion in straight course. Moreover the changes in forward speed induce no forces or moments. Thus $\partial Y / \partial u = \partial Y / \partial \dot{u} = 0$. It may be therefore deduced that $F_y(u_1, v_1, \dot{u}_1, \dot{v}_1, r_1, \dot{r}_1, \phi_1, \dot{\phi}_1) = 0$ and the equations cane be simplified. This procedure is used in the analysis of ship stability of motion in a straight line. The following simplified notation is introduced:

$$X_v = \frac{\partial X}{\partial v}, \quad Y_v = \frac{\partial Y}{\partial v}, \quad K_v = \frac{\partial K}{\partial v}, \quad N_v = \frac{\partial N}{\partial v} \qquad (6)$$

In the traditional ship maneuvering theory for displacement hulls, all the derivatives are evaluated around $v = 0$. In the following, it is shown that this procedure would not be valid for high speed planing hulls.

Determination of hydrodynamic derivatives by oblique towing

Velocity dependent hydrodynamic derivatives of Y_v, K_v and N_v in pre-defined draught and running trim may be determined by scaled model testing in towing tank for speed corresponding to the Froude number and various drift angles, β. Figure 2 shows

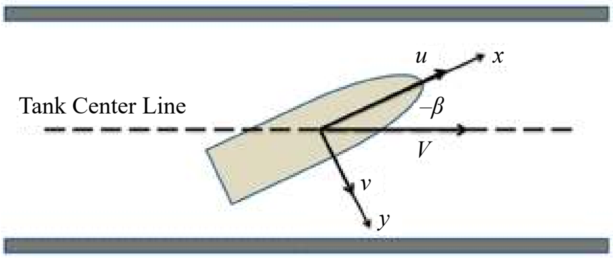


Figure 2. Orientation of model for oblique towing test

the orientation of the model in the towing tank. The transverse velocity component v would be generated along the y-axis:

$$v = -V \sin \beta \tag{7}$$

A dynamometer is located in the origin and measures the force Y, moment K, and moment N for different values of β. Next, the force or moment is plotted versus v and the tangent to the curve in $v = 0$ yields the value of the hydrodynamic derivative. There is no need to install the dynamometer in the center of gravity of the model but it should be reported that hydrodynamic coefficients due to the moments are measured with respect to that point. Consider Figure 3 in order to clarify the nature of derivatives Y_v and N_v.

In Figure 3 the directions of forces exerted on a body with forward speed u and lateral velocity v is shown. A lift force, in the opposite direction of v, is exerted on the body due to the flow angle of attack, $\beta = -v/V$, at bow and stern. Thus the coefficient Y_v would be negative. Moreover, the effect of the total force in bow, $Y_v v$, is larger than stern because the force effect point is closer to the ship bow. In addition, by considering the origin in the amidship, N_v would be usually negative for ships without rudders or fins. In most cases the addition of a rudder to the stern would increase $(Y_v v)_{stern}$, thus the negative N_v would be lower. If the ship rudder is large enough it is possible to experience a positive N_v; however, this is not the dominant situation for most hull forms and N_v would remain negative.

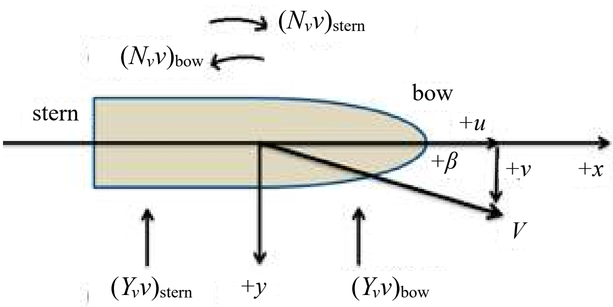


Figure 3. Ship with forward speed *u* and lateral velocity *v*

Planing hull model

Model tests were performed in the towing tank of Sharif University of Technology. A digital ruler was used to record the draught changes of the model in the center of gravity. Trim was measured by a digital angle meter. A dynamometer was used to measure resistance, sway force, yawing moment and rolling moment. All the measurements, recording rates, and data analysis were handled according to ITTC guidelines and procedures (ITTC, 2002).

Model selection for test depends on several parameters. First, the bodylines should be such that different flow phenomena around the hull could be observed. In this paper a high speed small pleasure craft has been studied. Figure 4 shows the bodylines.

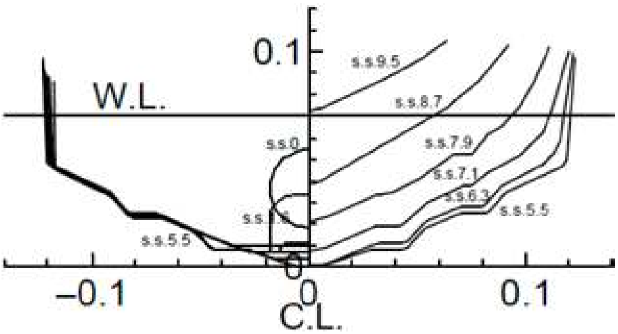


Figure 4. Model bodylines

Dynamic and geometric similarities should be provided during planing hull model tests. In most cases the flow around ship hull is turbulent. An important parameter, which determines whether the flow is laminar or turbulent better than the amplitude of turbulence, is the Reynolds number. The Reynolds number is the ratio between inertial forces and viscous forces. As the Reynolds number increases, viscosity breaks the flow instabilities and its damping effects decrease. These considerations should be taken into account for the selection of a suitable model size. The model size should be chosen according to towing tank dimensions. The most important factors influencing the model dimensions are:

1. Maximum model weight should be below the limit that would damage the measuring units at maximum test speed.
2. The maximum Froude number during the test should be within the limit in which the planing regime is expected.
3. The flow Reynolds number should be as large as possible to assure sufficient turbulence around the model hull.

4. The maximum hull length should satisfy the condition that the flow blockage factor is less than 7.5% at maximum drift angle, according ITTC guidelines.

The maximum forward carriage speed is considered to be 6 m/s, maximum drift angle is 30 degrees, water depth 1.2 m, and towing tank beam is 2.5 m. Thus the main dimensions of the model are as in Table 1.

Table 1. Planing hull model main dimensions

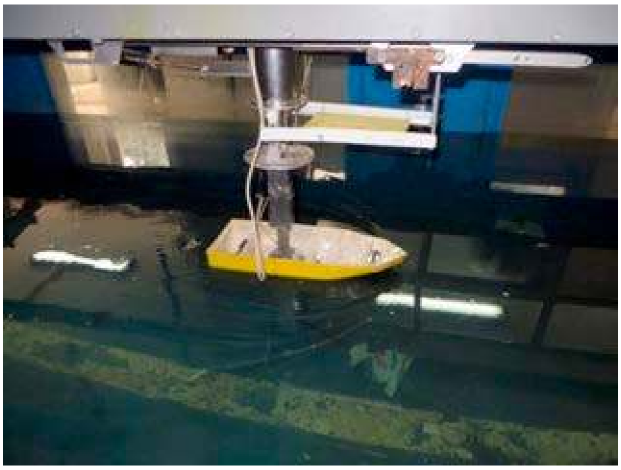
Length Between Perpendiculars (m)	0.50
Wetted Breadth (m)	0.19
Draught (m)	0.05
Weight (kg)	3.000
Deadrise (deg)	22

Figure 5. Manufactured model of planing hull

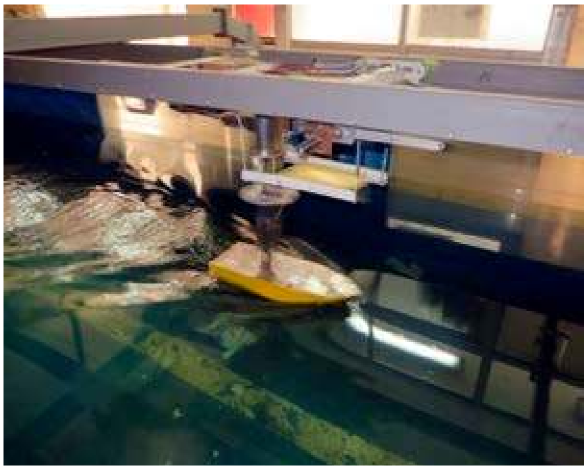
The model (Figure 5) is fabricated using fiberglass in accordance to ITTC tolerances. Next the model displacement weight is adjusted. Rigid weights are arranged in the model so that the specific pitch and yaw gyration radius is obtained. It is necessary to satisfy the turbulent boundary layer around the hull to reduce the model scale effects. The model size adopted in this paper is small. The Reynolds number range selected is large enough, but it is necessary to generate sufficient turbulence even at low speeds. Therefore a steel rod is mounted on the model perpendicularly to the direction of fluid flow. The steel rod thickness is 3 mm and it is located at 5% of the ship's length according to ITTC2002 recommendations. Certainly the rod would be out of water as the forward speed increases and dynamic lift and trim occurs but at these speeds the Reynolds number would be sufficiently high.

Running attitude determination in straight line

The resistance, rise and dynamic trim versus Froude number of the planing hull is determined by model test in displacement, semi-planing and



a) Fn = 0.1



b) Fn = 0.6, significant wave making and sinkage

c) Fn = 1.4, the hull separates water spray

d) Fn = 2.4, Fully planing condition

Figure 6. Towing test in calm water

planing modes. The maximum test speed considered was 5.99 m/s, equivalent to Fn = 2.58. Higher speeds were not achievable due to towing system limitations. Despite this matter, the hull lift was considerable and planing conditions were achieved.

Figure 6a shows the model test in Fn = 0.1. It is obvious that there is little turbulence around the hull and not much wave making. Rise and trim would also be negligible due to the low speed. Figure 6b depicts the model test in Fn = 0.6, corresponding to a forward speed of 1.39 m/s. In this range of speeds the vessel has no lift and is completely in semi-planing mode. The wave making effects are dominant and waves surrounded the hull completely. The trim angle (by aft) is significantly increasing up to this speed. There is no lift and limited deep water squatting is observed, just as in displacement hulls.

Figure 6c refers to the model test at Fn = 1.4 corresponding to a forward speed of 3.35 m/s in the model scale. The flow separates significantly from the chines. The dynamic trim is decreased by increasing the forward speed that is expected in planing hulls. Hull rise is significant and draught is reduced to 30% of static draught. The hull shape has great ability to separate the water spray. Figure 6d refers to Fn = 2.4, corresponding to a forward speed of 5.57 m/s in model scale. The hull is fully in planing conditions. The dynamic trim is decreased and vessel draught is at the minimum value.

The planing hull resistance is usually made dimensionless by dividing it by vessel displacement. This parameter is very useful because it can be interpreted as the ratio of hull drag to lift. Figure 7, shows the ratio of hull resistance to weight versus Froude number measured from the model test. One of the main challenges in this paper is to show that reliable results can be obtained from tests carried out with a small-sized model of planing hull. In this view, in Figure 7 the results are compared to various references. In the displacement region up to Fn = 0.4 the resistance is compared to the Holtrop method, suitable for displacement hulls (Holtrop & Mennen, 1978). In the semi-planing ranges of the Froude number, 0.6 to 0.8, the results are compared to the Savitsky semi planing resistance estimation method (Savitsky & Brown, 1976). Finally, in the planing regime, corresponding to Froude numbers 0.9 to 2.4, Savitsky's method is applied for validation (Savitsky, 1964).

According to Figure 7, the hull resistance in Fn = 2.4 reaches 31.9% of craft weight, which is expected for small sized planing crafts. The transition to planing condition occurs for Froude numbers in the range of 0.9 to 1.1. The changes in the resistance with respect to forward speed are not sharp in this range due to the significant increase of hull rise. The Holtrop method is a good approximation of hull resistance in displacement mode of motion and the

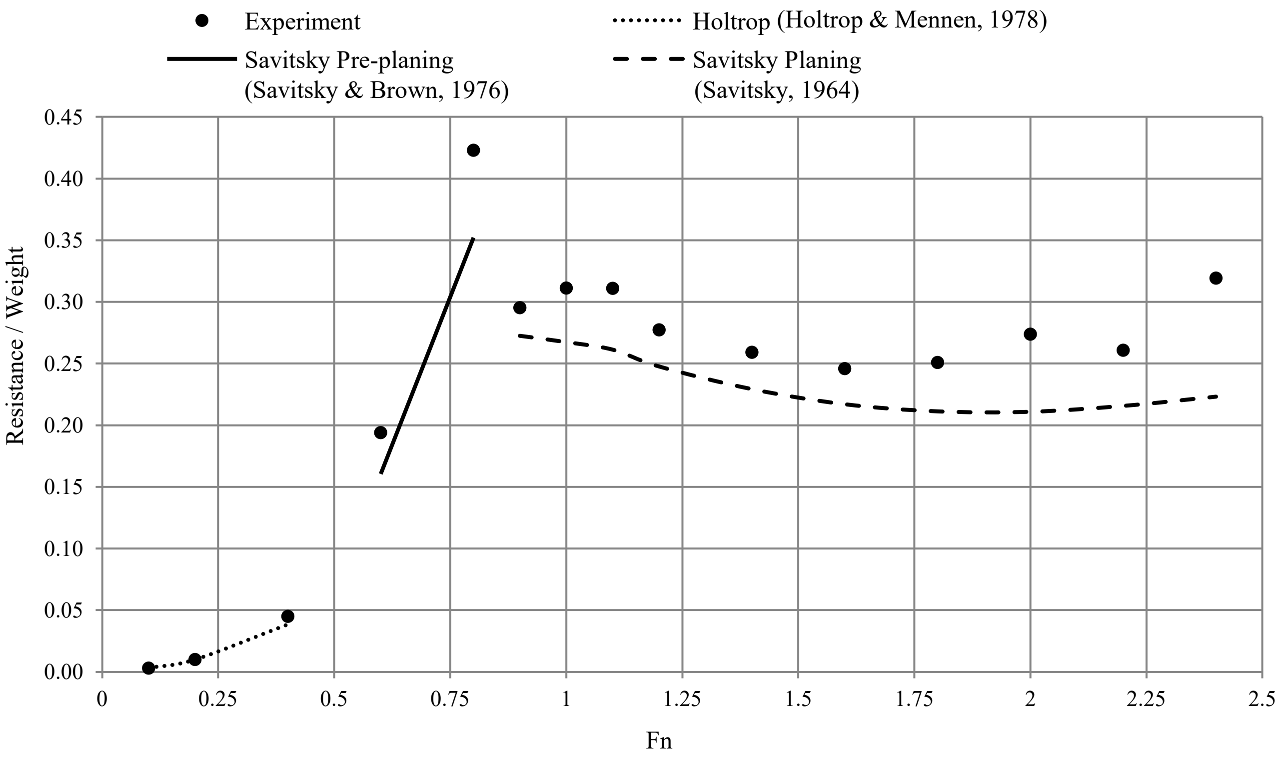


Figure 7. Resistance curve versus Froude number, calm water towing test

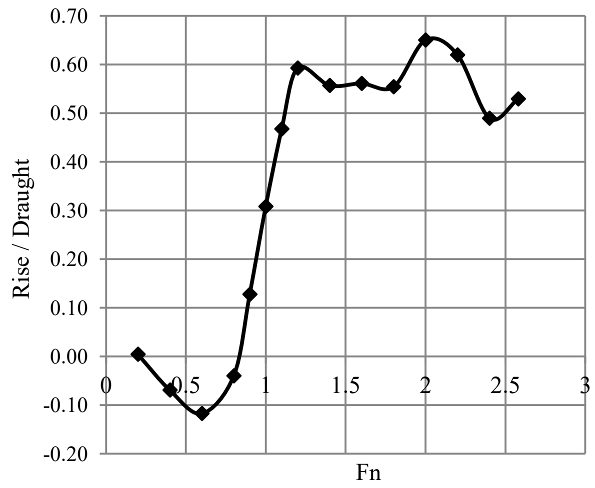


Figure 8. Rise versus Froude number in calm water and straight course

average error of test results using this method has been 4%. The results have a 17% deviation from the Savitsky semi-planing method, meaning that the error in the Savitsky method is 15% for the planing region. The error increased at higher forward speeds, mainly due to assumptions of Savitsky's method concerning fully planing condition and prismatic hull form of constant deadrise. It should be noted that the planing hull studied here is not prismatic. Despite this fact, the Savitsky's formulae would be a good estimation of hull resistance at the early design stages. In Figure 8, rise is made dimensionless by dividing it by the static draught and plotted against Fn.

By analyzing Figure 8, it seems that not only does the rise not increase up to Fn = 0.8, the deep water squat is also limited. A 10% static draught is recorded, in agreement with the physical behavior of planing hulls. After that, for Fn = 0.9, the hydrodynamic characteristics of planing surfaces of the hull begin to affect the running attitude. The rise is positive and begins to increase considerably, proving the planing behavior. When Fn = 1.2, the hull rise is 59% of the static draught. For larger Froude numbers, the rise remains approximately constant and it reaches its maximum value for Fn = 2.0.

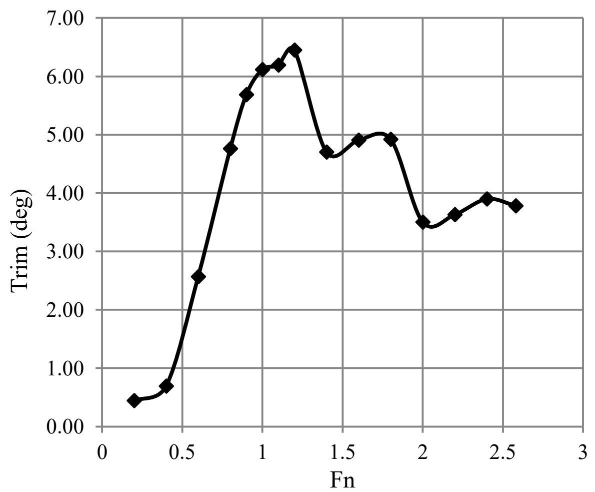


Figure 9. Dynamic trim in calm water

The dynamic trim of the hull is presented in Figure 9. Conventional displacement hulls are not able to produce a sufficient lift force. An increase in speed would lead to an excessive increase of dynamic trim by aft, but as shown in Figure 9, for Fn = 1.0 and above the dynamic trim remains approximately constant and significant lift is generated thanks to the action of planing surfaces.

Static drift model test

The main objective of the static drift model test was the determination of hydrodynamic derivatives X_v, Y_v, K_v in different forward speeds for a domain of drift angles. The test procedure was in accordance with ITTC2002. The carriage settings were such that the model reached the desired speed in the minimum time and continued the course at constant speed. Thus the measurement of flow parameters would be in steady state flow condition. The values of surge force, sway force and rolling moment were measured in each run. Each run was repeated three times to check the repeatability of test results. If the differences of the results were within the acceptable limit the three values obtained were averaged, else the test was repeated for a fourth time. A time interval of 15 minutes was considered between each run in order to be sure that the tank water was calm and no surface wave or internal vorticity would affect the test results.

Tests were repeated for forward speeds corresponding to Froude numbers of 0.4, 0.8, 1.2, 1.6 and 2.0. The characteristic length for the evaluation of Fn was considered to be the hull length on waterline in stationary condition. The range of drift angles was taken to be from –18 to +18 degrees. For simplicity, the forces and moments were made dimensionless by reference to the length on waterline, L, and forward speed, V, as follow:

$$v' = \frac{v}{V} \qquad X' = \frac{X}{0.5\rho V^2 L^2}$$
$$Y' = \frac{Y}{0.5\rho V^2 L^2} \qquad K' = \frac{K}{0.5\rho V^2 L^2} \tag{8}$$

Figure 10, represents the surge force versus sway velocity. The surge force in zero drift angle is equal

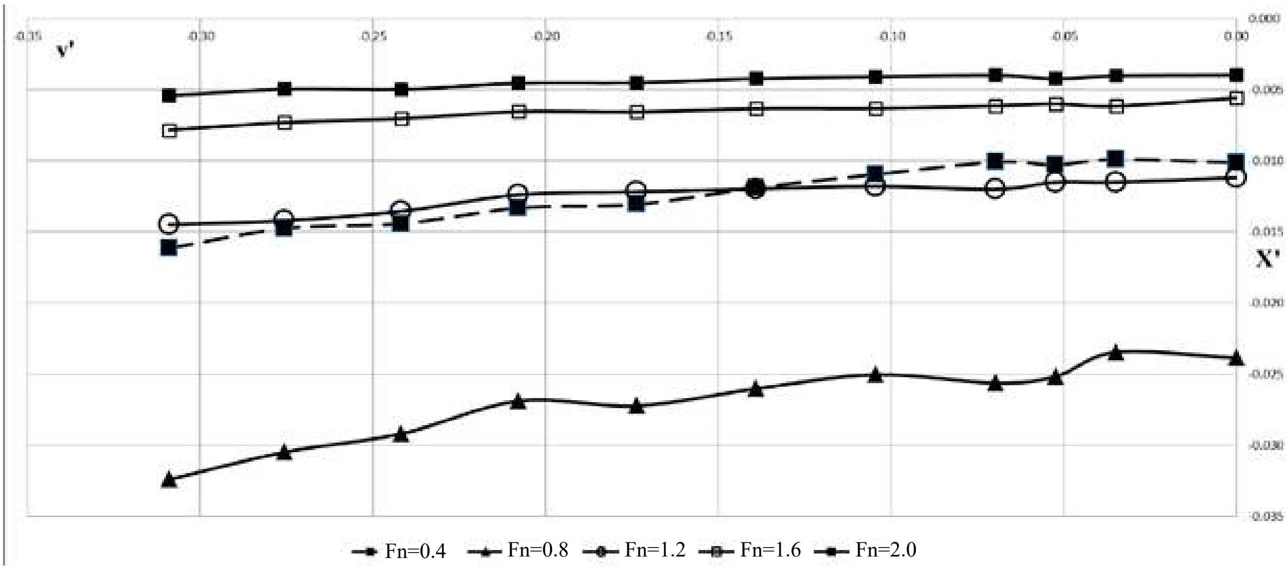


Figure 10. Surge force versus sway velocity

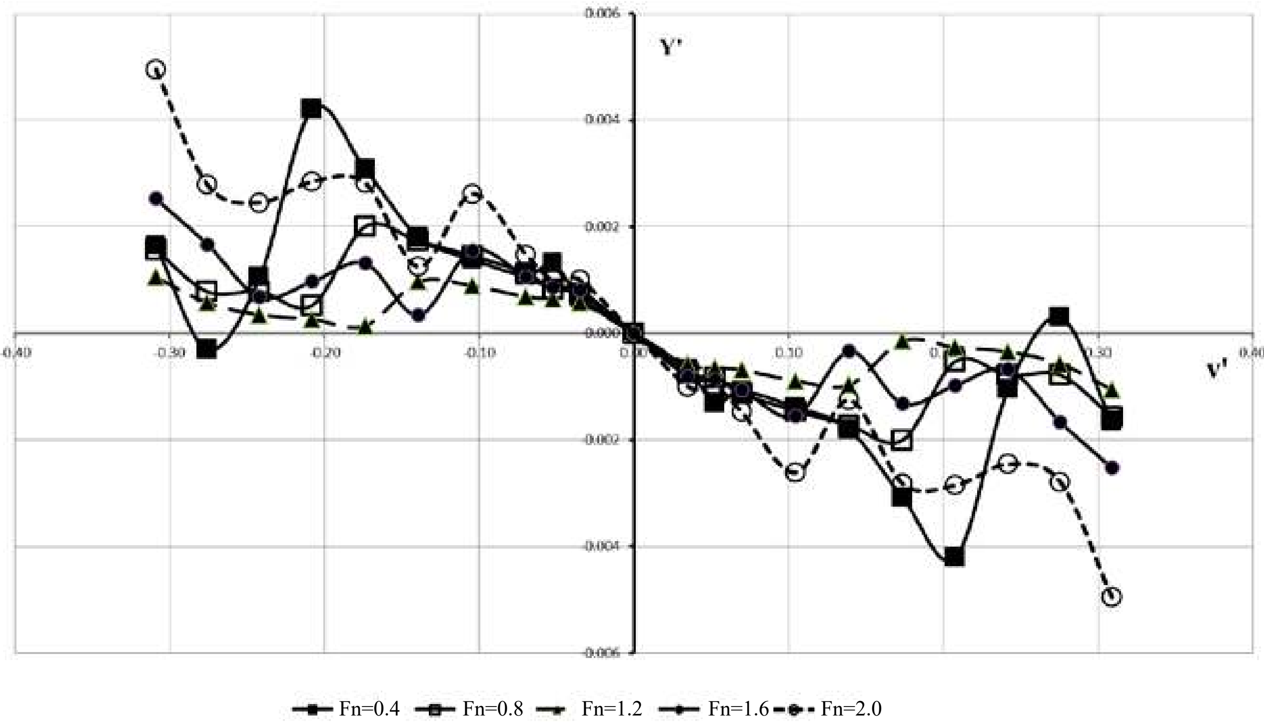


Figure 11. Plot of sway force against sway velocity in static drift test

to hull resistance at that speed. Generally all the surge forces are measured with a negative sign, consistently with the orientation of the system of coordinates selected. According to Figure 11, one of the influencing parameters in a specific forward speed is the increase in drift angle that leads to a larger surge force, although the change of surge force by drift angle is very slight for displacement and planing conditions. However, in the range of moderate Froude numbers, corresponding to the semi-planing regime, the surge force changes significantly as the drift angle increases. Moreover, it should be noted that the surge force increases by forward speed in the same way as the hull resistance in zero drift angle.

In most conventional hull forms, the slope of sway force curve against lateral speed in zero transverse speed is negative and the sway force becomes larger as the drift angle increases. This behavior is what is expected from conventional displacement hulls but in planing hulls the situation may be slightly different due to dynamic lift effects. Figure 11 shows the

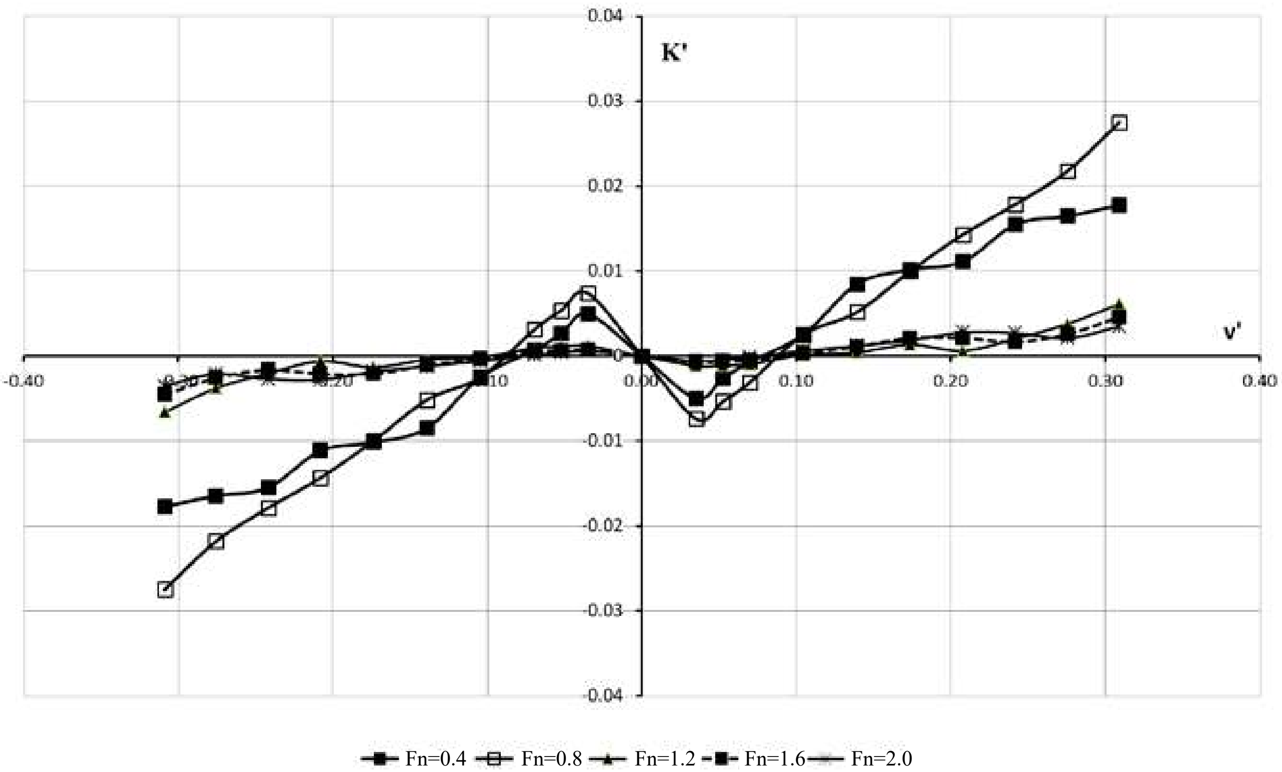


Figure 12. Rolling moment against sway velocity from static drift test

sway force against sway velocity of a planing hull measured from static drift tests.

The behavior around $v' = 0$ is the same as the one observed for displacement hulls. The only difference is that the slope of the curve in zero sway velocity would be the same for various forward speeds in the case of displacement hulls. Therefore the hydrodynamic derivatives are independent of speed for displacement hulls. In the interval $-0.17 \leq v' \leq +0.17$ the behavior is very similar, but outside this interval the sway force decreases unexpectedly as the drift angle increases. Obviously the definition of hydrodynamic derivatives should be modified in these regions in the case of a planing hull. According to the test results a small perturbation of drift angle about the equilibrium point $v' = 0$ would increase the sway force exerted on the hull. But in $v' = \pm 0.17$ a small perturbation of drift angle would decrease the sway force. Thus it can be concluded that the hydrodynamic derivative definition around the equilibrium condition $v' = 0$ would not be sufficient to describe the planing hull behavior. The hull dynamic in other equilibrium conditions may be different. According to Figure 11, the sudden changes in hydrodynamic coefficients in a specific running attitude of planing hulls could be considered as one of the main reasons of planing hull dynamic transverse instabilities.

Figure 12 describes the test results of rolling moment versus sway velocity: the rolling moment is significant in planing hulls at high forward speeds and the hydrodynamic coefficient, K_v, takes on two different values. When the drift angle exceeds a certain limit the value and sign of K_v change dramatically. Moreover, the forward speed (in terms of Fn) has a great effect on K_v. The hydrodynamic coefficient is smaller in the dimensionless form in the planing conditions. It should be noted that for a specific sway velocity, approximately $v' = \pm 0.07$, the induced rolling moment is zero and its sign changes.

Validation

There is no theoretical method to calculate the hydrodynamic derivatives of planing hulls. Lewandowski 1997 proposed a semi-empirical method to estimate the linear sway, roll and yaw coefficients of hard-chine hulls in planing regime under the assumption that water is completely separated from chine and transom. Model tests were performed within the ranges of speed, drift angle, roll angle, trim angle and turning radius and formulae for $Y_v, Y_{\dot{\psi}}, K_v, K_{\dot{\psi}}, N_v, N_{\dot{\psi}}$ were proposed (Lewandowski, 1997).

Based on the experimental results of the planing hull model test, the tangent of sway force (in *N*) and roll moment (in *N–m*) curve against lateral speed was calculated in the origin and the hydrodynamic coefficient was estimated. The hydrodynamic coefficients

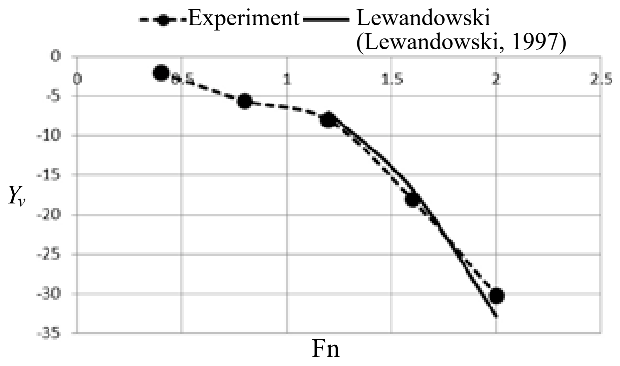


Figure 13. Comparison of Y_v from model test reults and Lewandowski formulae

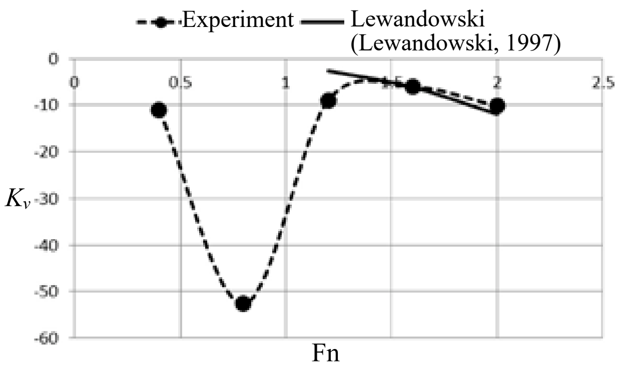


Figure 14. Comparison of K_v from model test results and Lewandowski formulae

were then calculated through the Lewandowski formulae and compared to experimental results. When calculating the coefficients by the Lewandowski method, the hull was assumed to be completely prismatic. The mean deadrise of midship sections is used as characteristic deadrise in the formulae. Figure 13 shows the linear damping coefficients of sway and Figure 14 shows the roll moment. The coefficients are plotted against Fn and forward speed effects on hydrodynamic coefficients have been the main concern.

For the fully planing conditions realized for Fn values greater than 1.2, there is a good agreement between the results of the experiment and of the Lewandowski formulae. The fully planing condition is, in effect, a basic assumption of the Lewandowski method, thus the differences observed in the other ranges of Froude number are reasonable.

It is emphasized that the Lewandowski formula is applicable just in the fully planing regime. The results are therefore comparable for Fn ≥ 1.0. Another point is that a great increase of forces is observed at Fn = 0.8, as shown in Figure 14. At this speed the wave making is dominant and the forces exerted on the hull increase due to wave making; the hull lift is small and the generated waves surround the hull completely.

Determination of hydrodynamic coefficients

The relation of Y to v can be assumed to be linear in displacement hull types. As a consequence, the constant hydrodynamic derivatives can be applied to the mathematical model of displacement hull maneuvering. By analyzing Figures 10, 11 and 12 three principle facts can be deduced for planing hulls:

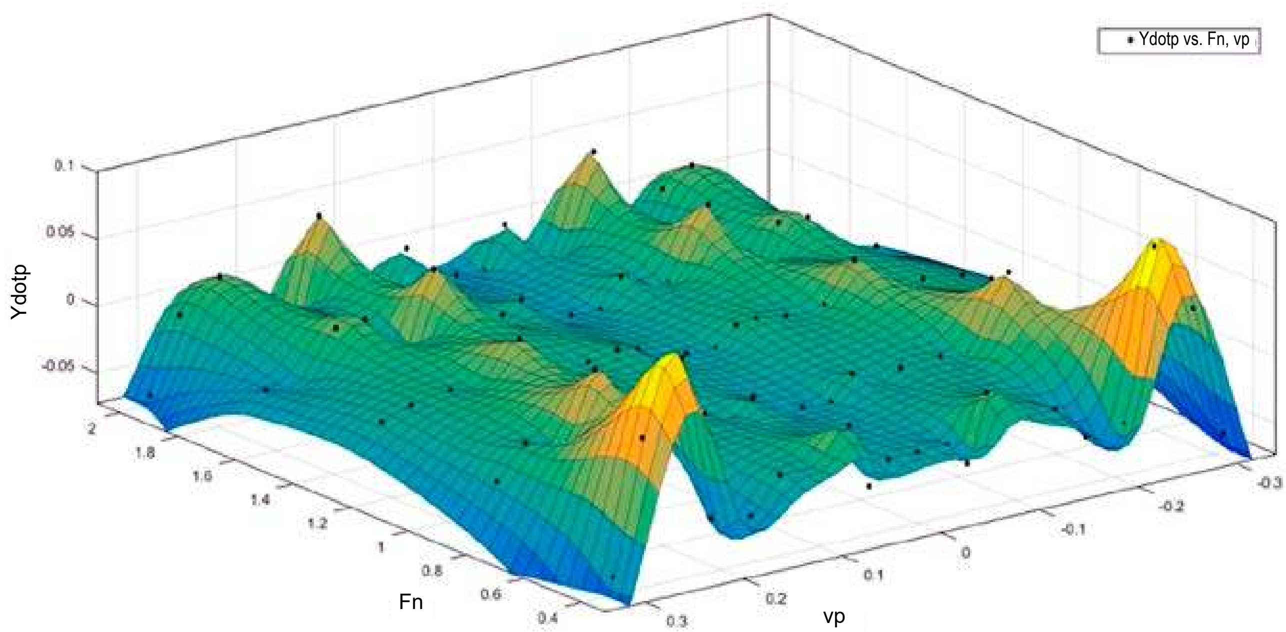


Figure 15. Y'_v versus sway velocity and Fn

1. In the mathematical model of planing hull maneuvering the slope of curves X, Y, and K versus v cannot be assumed constant and, depending on the value of the lateral speed and equilibrium point at which the simulation is executed, variable hydrodynamic derivatives should be applied to the mathematical model.
2. Actual hydrodynamic coefficients are a function of forward speed and flow regime around the hull.
3. The initial value of simulation
$$F_y(u_1, v_1, \dot{u}_1, \dot{v}_1, r_1, \dot{r}_1, \phi_1, \dot{\phi}_1) = 0$$
in Eq. 5 should be determined based on the instant value of drift angle and forward speed.

The above three facts are discussed in detail in the following paragraphs.

According to the model test results, described in Figures 11 and 12, the tangents to the Y and K curves at different drift angles are not the same. Therefore, the method based on constant hydrodynamic coefficients cannot be applied to planing hull dynamics. A computer code was developed here that calculates the tangent to the curves in Figures 11 and 12 for

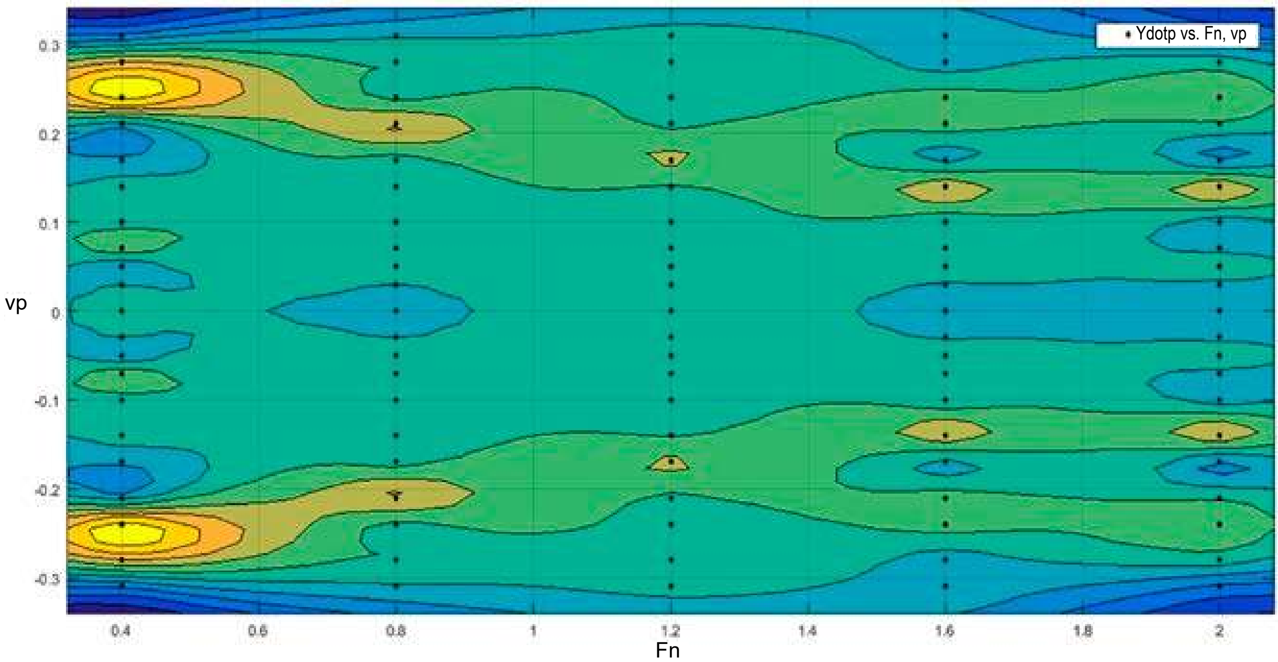


Figure 16. Contours of Y'_v versus sway velocity and Fn

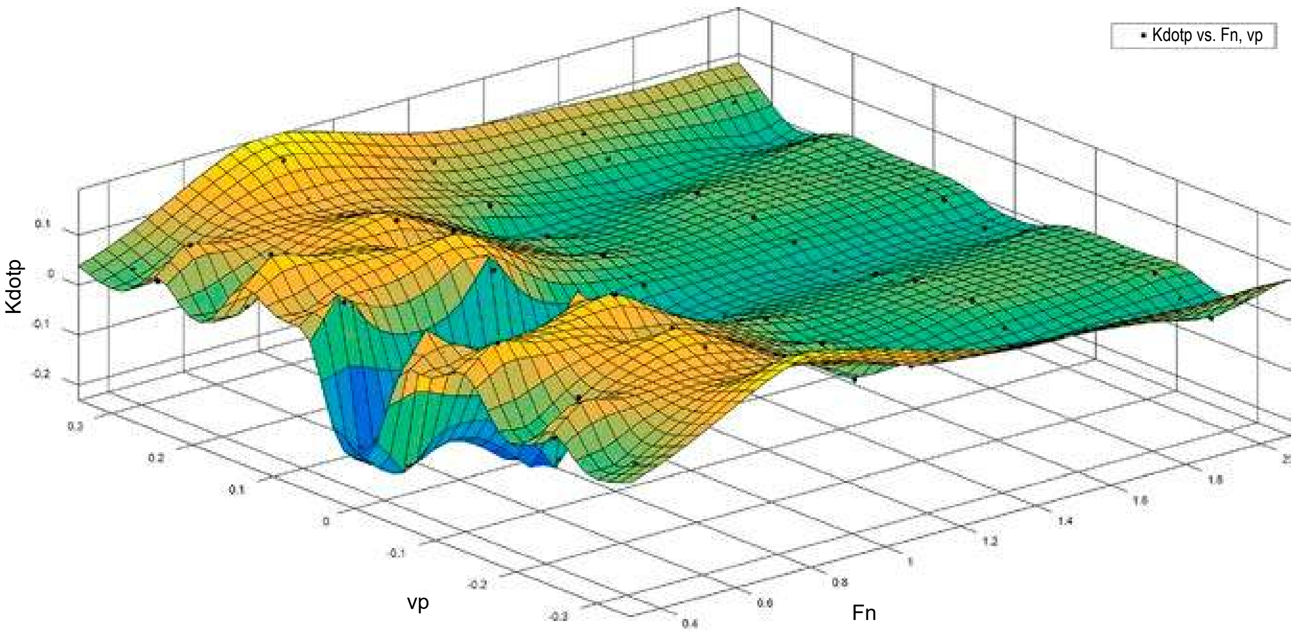


Figure 17. K'_v versus sway velocity and Fn

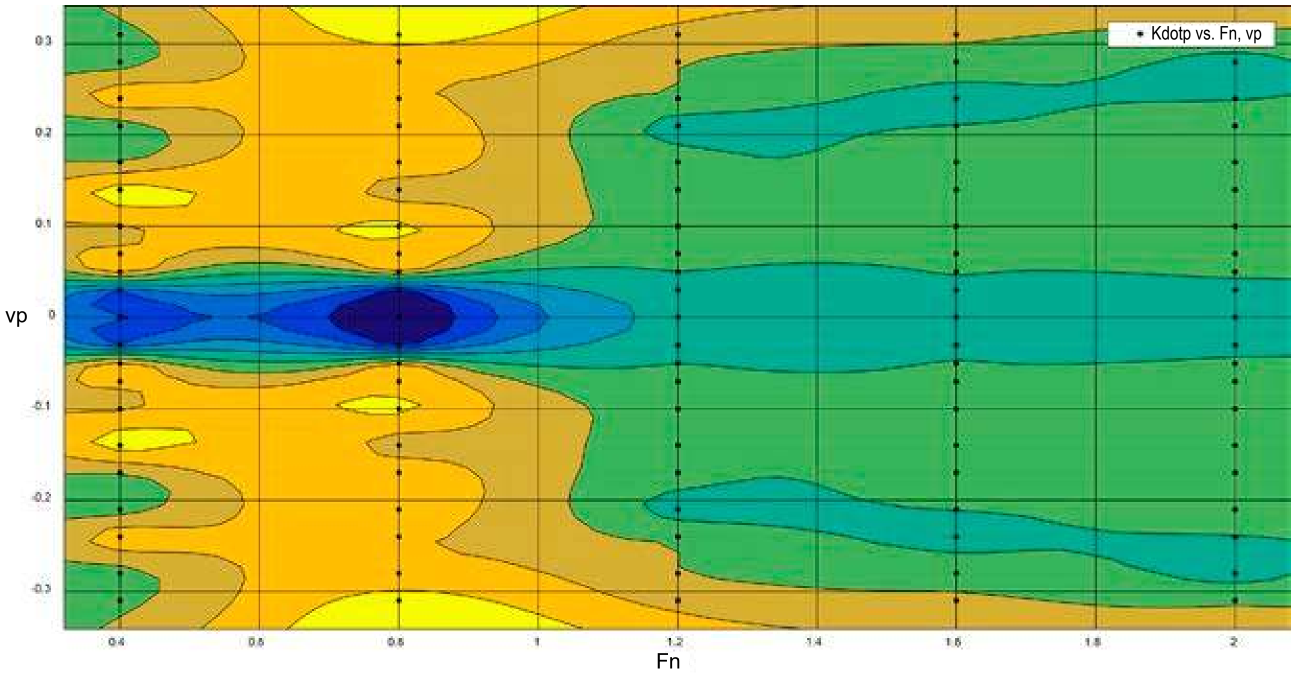


Figure 18. Contours of K'_v versus sway velocity and Fn

various sway velocities and Froude numbers. The calculated points and interpolated surface are shown in Figure 15 and contours of Y'_v are presented in Figure 16.

According to Figure 15, while the lateral velocity is small up to $v' = 0.1$, the hydrodynamic derivative Y'_v remains approximately constant for all the Fn ranges. Meanwhile, when the lateral velocity (drift angle) increases, significant changes in the hydrodynamic coefficient is recorded and may also become negative.

The calculated points and interpolated surface are shown in Figure 17 and contours of K'_v are shown in Figure 18.

According to Figures 17 and 18, changes in hydrodynamic roll coefficient of the vessel in planing condition (Fn above 1.2) are very smooth; however, K'_v is very sensitive to Fn and v in the semi-planing regime.

Conclusions

Systematic model tests to describe the effect of forward speed and drift angle on hydrodynamic coefficients of planing hulls have been the main concern of this paper. It is demonstrated that it is possible to achieve reliable results even with small models of a planing hull. According to the results, the hydrodynamic derivatives of planing hull are a function of the actual drift angle more than forward speed. A unique hydrodynamic derivative cannot be applied to a planing hull when the drift angle exceeds a certain limit and nonlinear variations of the hydrodynamic derivative with running attitude and forward speed should be considered. The effect of dynamic trim and rise on hydrodynamic derivatives were neglected and should be studied further. Although the previously proposed methods, such as Lewandowski's semi-empirical model, describe the effect of forward speed on the hydrodynamic derivatives, it is shown that the effect of actual drift angle in severe conditions should be taken into account.

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