Experimental and Numerical Studies on Di-hull Interference

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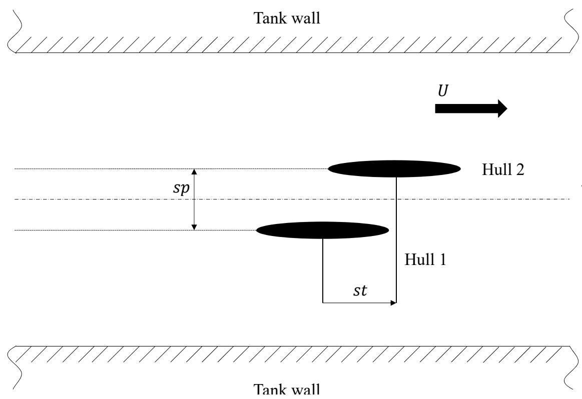
1 INTRODUCTION

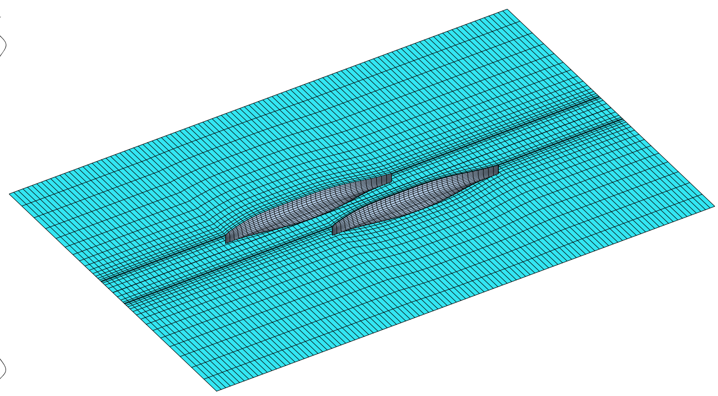
Numerical studies of multi-hull vessels, such as Söding (1997), Tuck and Lazauskas (1998) and Yeung et al. (2004), have shown that the interference between ship-generated waves can significantly reduce the total wave resistance. The two hulls together can experience smaller total wave resistance than the sum of the resistance of each hull moving in isolation. This phenomenon is especially prominent when the hulls have non-zero longitudinal spacing, called stagger *st* (see Fig. 1a). In the present study, experiments are conducted in the towing tank for asymmetric di-hull systems with the stagger being half of a ship length. This "optimal" longitudinal location, in terms of maximum resistance reduction, was discussed in Söding (1997) and confirmed by Yeung et al. (2004) and Faltinsen (2005) as well. Such optimality was based primarily on the superposition of the two sets of waves from the two hulls. In the experiments, the resistance, sinkage, and trim of each hull in the system are measured separately so as to examine the effect of interference on the wave resistance of the entire di-hull system as well as on each individual hull. These individual-hull data were not previously reported in the open literature. Besides, a multi-hull panel method based on the use of "simple source" and Green's Theorem with linearized free-surface boundary condition, as first introduced in Bai and Yeung (1974) and Yeung (1982), is also re-developed in this research. Such a numerical approach takes into account the demi-hull interaction from the body proximity, but requires the solution of the coupled integral equations. A better match to experimental results from these more elaborate modeling than its thin-ship computational counterpart is expected and reported here.

2 EXPERIMENTAL SET-UP

The di-hull towing-tank tests were conducted in the Model-Testing Facility in the Richmond Field Station of the University of California at Berkeley, with the tank measuring 80 m, 2.55 m, and 1.8 m for length, width, and depth, respectively.



(a) Experimental set-up in a tow tank



(b) Mesh on free surface and hulls viewed from below

Figure 1: Schematic for towing-tank test and mesh for MSPM of di-hull systems.

The studied di-hull system consists of two identical Series 60 model hulls at a 1/80 scale ratio. The principal geometric parameters are given in Table. 1. Turbulence stimulator consisting of 1/16

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gauge solder wire were mounted at Station 1. In order to adjust the relative positions between the hulls arbitrarily, a special towing frame with continuously adjustable stagger st, measured midship to midship, and separation sp, measured from centerplane to centerplane, was fabricated. Each hull model was allowed to take independent trim and sinkage, rendering the tested di-hull systems more closely resembling a two-ship formation consisting of independent vessels, rather than a catamaran with two demi-hulls bounded together.

Table 1: Geometric particulars of studied hulls

Block coefficient	$C_B = 0.6$
Overall length	$L = 1.524$ m
Beam	$B = 0.2032$ m
Draft (at design load waterline)	$D = 0.0813$ m
Displacement	$\Delta = 15.1$ kg
Wetted surface area	$S = 0.396$ m^2
Scale ratio	1/80

Table 2: Hull configurations of towing-tank tests

	Mono-hull test	Di-hull tests	
st (m)	Not applicable. Model towed along the centerline of the tank.	0.762	0.762
sp (m)		0.35	0.45
$\overline{st} = st/L$		0.500	0.500
$\overline{sp} = sp/L$		0.230	0.295
U (m/s)	min: 1.00; max: 1.66		
F_n	min: 0.26; max: 0.43		

The test matrix is shown in Table 2. Twelve forward speeds U between 1.00 m/s and 1.66 m/s were selected, with the corresponding Froude number $F_n = U/\sqrt{gL}$ between 0.26 and 0.43. Before conducting the more complicated di-hull towing-tank tests, conventional mono-hull towing-tank tests were performed first to acquire the wave resistance of a mono-hull in isolation. In the di-hull tests, two beam-type load cells mounted on the towing frame were used to measure the total resistance of each hull separately. The wave resistance was extracted by subtracting from the total resistance the frictional drag, which is given by the standard the ITTC line correlation formula (Hughes, 1954), while the wetted surface areas used are based on the hulls with zero sinkage and trim. In addition to total resistance, the sinkage h of each hull was measured by a precision potentiometer, while the trim φ was measured by the MPU-6050 micro-mechanical gyroscope. The sinkage and trim data were collected at 30 Hz frequency by Arduino Nano controllers.

To represent the effects from wave interference on the total (wave) resistance of the di-hull system, the interference parameter α, as defined in Yeung et al. (2004), was computed from the experimental results for each test case, namely

$$\alpha = \frac{C_{w_1} + C_{w_2} - 2C_{w_m}}{4C_{w_m}}, \quad C_w = \frac{R_w}{\frac{1}{2}\rho U^2 S}. \tag{1}$$

where C_{w_1} and C_{w_2} are the coefficients of wave resistance of each individual hull in the di-hull system, and C_{w_m} is the wave resistance coefficient of each constituent hull in isolation (i.e., a mono-hull). The wetted surface area with zero trim and sinkage is given by S. α has the physical meaning of the % change in resistance, averaged over the two hulls, relative to a mono-hull of the same displacement. Thus, a negative α value indicates the existence of beneficial wave interference for the given hull configuration and forward speed, signifying the total wave resistance of the di-hull system being smaller than the sum of the wave resistances of the component mono-hulls moving in isolation. Conversely, a positive α value indicates detrimental wave interference which increases the total wave resistance of the di-hull system.

3 SOLUTION BY MULTI-HULL SIMPLE-SOURCE PANEL METHOD (MSPM)

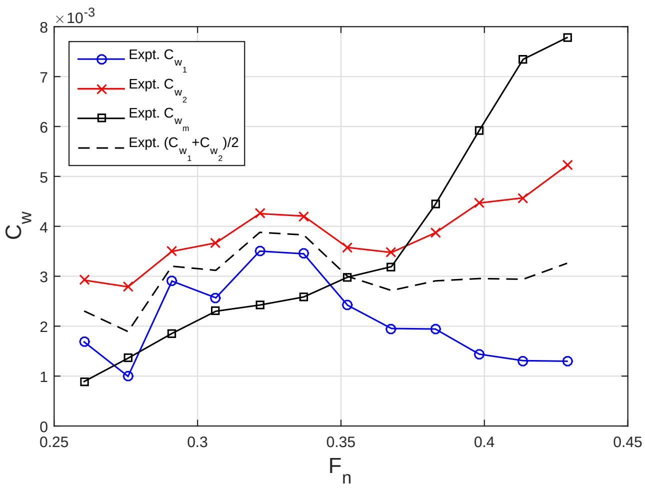
Apart from the experimental investigation, a numerical scheme based on the “simple-source formulation” using Green’s Theorem and constant-strength boundary elements as in Hess and Smith (1964) is used. The term “simple-source” was coined in (Yeung, 1982) with the purpose to encompass both 2D and 3D formulations in general. The linearized free-surface boundary condition $U^2\phi_{xx} + g\phi_z = 0$ (see Wehausen & Laitone, 1960) is used with a non-linear pressure expression on the hull surfaces. Boundary conditions on the hull surfaces are of the Neumann type: $\phi_n = Un_x$. Typically, each of hull consists of 648 quadrilateral panels (36×9) and 4104 panels are used to construct the free-surface

(108×38). The computational domain is four ship-lengths in length and two ship-lengths in width. Expressions for higher-order panels of triangular patches were available in Bai and Yeung (1974) and can speed up solution convergence though not pursued here.

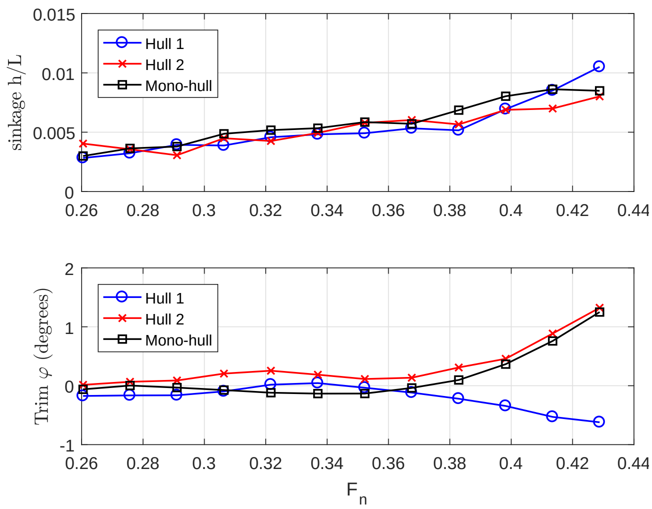
Compared with the thin-ship computation such as the computer program CATRES provided in Yeung et al. (2004), the present numerical solution essentially solves the di-hull equivalent of the so-called Neumann-Kelvin problem, with an iterative pressure Kutta condition applied at the stern of each hull to ensure the uniqueness of the solution. Fig. 1b displays some details of the panellization system. In this way, the proximity effects between hulls can be better accounted for both near-field and far-field, and the solution is free from the influence of thin-ship approximation. Besides, compared with a Computational Fluid Dynamics (CFD) approach such as the Reynold-Averaged Navier-Stokes (RANS) Simulation which takes the viscosity of the fluid into account, the present method requires significantly less computational resources to qualitatively capture the basic wave-making properties and overall wave interference effects of the di-hull system.

4 COMPARISON OF EXPERIMENTAL AND NUMERICAL RESULTS

Experimental results of C_{w_1}, C_{w_2}, and C_{w_m} as well as the sinkage h and trim φ of the individual hulls in the di-hull system with $(\overline{st}, \overline{sp}) = (0.5, 0.230)$ and of an isolated mono-hull are given in Fig. 2. In computing C_w's from experimental measurements using the ITTC line (Hughes, 1954), the form factor is taken to be zero as there are no appendages.



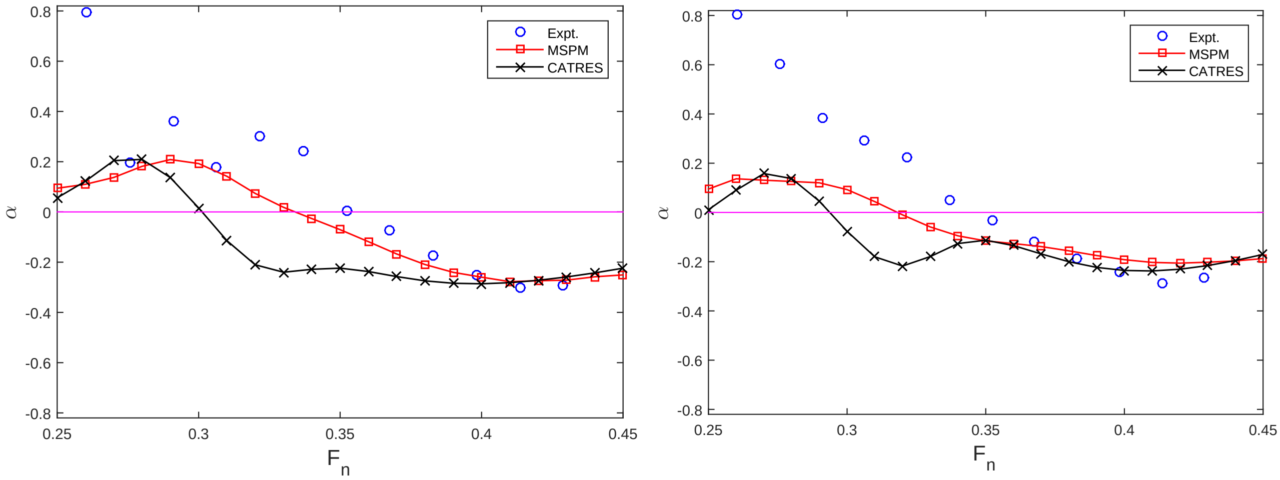
(a) Resistance coefficient C_w



(b) Sinkage and trim of individual hulls

Figure 2: Experimental results of C_w, sinkage h, and trim φ for each hull in the di-hull system and for an isolated mono-hull. Positive sinkage and trim correspond to increased draft and trim by the stern, respectively. $(\overline{st}, \overline{sp}) = (0.5, 0.230)$.

Experimental results of C_w show that both hulls in the di-hull system tend to experience larger wave resistance than their mono-hull counterpart in isolation for $F_n < 0.35$. The increase in drag is more prominent for the hull in the front (Hull 2 in Fig. 1a). However, the opposite is observed when $F_n > 0.37$ with both hulls experiencing reduced drag compared to the mono-hull value. The drag reduction is most significant for the hull in the rear (Hull 1). The measured sinkage increases monotonically with increasing F_n for the mono-hull case and the di-hull case. Trivial differences exist between the hulls for the speed range mentioned. The magnitudes of trim measurements are observed to be generally insignificant for both hulls in the di-hull system and for the isolated mono-hull until $F_n \approx 0.38$, at which point the three sets of measurements start to diverge: the trim of Hull 1 at the rear becomes increasingly negative (trim by the bow), while the front hull, as well as the mono-hull case, shows increasing trim by the stern with increasing Froude number. In short, for an asymmetric di-hull system, the effect of di-hull wave interference is more significant for the hull in the rear, in part because it is in the wake region of the front hull. Furthermore, the non-zero stagger also leads to a distinct distribution of vertical force on Hull 1, making it trim by the bow. In Fig. 2a, the dashed line represents the average of the resistance of the component hulls, which can be directly compared with the curve of the mono-hull of the same displacement to provide a sense of α.



(a) $(\overline{st}, \overline{sp}) = (0.5, 0.230)$ (b) $(\overline{st}, \overline{sp}) = (0.5, 0.295)$

Figure 3: Comparison between experimental and numerical results of α for different hull configurations.

The experimental results of α, compared with those from the MSPM computation and the thin-ship computation of CATRES, are given in Fig. 3. The values of α from tow-tank tests turn negative at $F_n \approx 0.35$ and reach the lowest value of -0.3 at $F_n = 0.414$, which agrees with the MSPM computational results of α. MSPM captures the variation and magnitude of their experimental counterparts quite well, particularly the predicted zero-crossing at $F_n \approx 0.34$ for Fig. 3(a), and 0.32 for Fig. 3(b). The thin-ship computation of CATRES, on the other hand, provides less satisfactory predictions, depicting an unrealistic "hollow" at $F_n \approx 0.32$. Apart from using the thin-body approximation, CATRES computes the wave-making properties of the di-hull system by directly superposing two sets of non-interacting mono-hull wave patterns, as if the wave field from each hull in the system is not affected by the presence of the other. It was analytically simple to model. The interaction of near-field waves between the hulls was also neglected. Thus, in order to better capture the effect of such interference on the wave resistance of the di-hull system, the interacting wave-making property of each hull caused by the presence of the nearby hull, and the contribution from near-field disturbances should be accounted for. In the Workshop, more extensive results will be presented to elucidate how MSPM improves the sinkage and trim predictions of individual hulls as well as their wave resistance.

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