Computational Towing Tank Procedures for Single Run Curves of Resistance and Propulsion

A procedure is proposed to perform ship hydrodynamics computations for a wide range of velocities in a single run, herein called the computational towing tank. The method is based on solving the fluid flow equations using an inertial earth-fixed reference frame, and ramping up the ship speed slowly such that the time derivatives become negligible and the local solution corresponds to a quasi steady-state. The procedure is used for the computation of resistance and propulsion curves, in both cases allowing for dynamic calculation of the sinkage and trim. Computational tests are performed for the Athena R/V model DTMB 5365, in both bare hull with skeg and fully appended configurations, including two speed ramps and extensive comparison with experimental data. Comparison is also performed against steady-state points, demonstrating that the quasisteady solutions obtained match well the single-velocity computations. A verification study using seven systematically refined grids was performed for one Froude number, and grid convergence for resistance coefficient, sinkage, and trim were analyzed. The verification study concluded that finer grids are needed to reach the asymptotic range, though validation was achieved for resistance coefficient and sinkage but not for trim. Overall results prove that for medium and high Froude numbers the computational towing tank is an efficient and accurate tool to predict curves of resistance and propulsion for ship flows using a single run. The procedure is not possible or highly difficult using a physical towing tank suggesting a potential of using the computational towing tank to aid the design process. [DOI: 10.1115/1.2969649]

Keywords: computational towing tank, resistance, propulsion, ship flow

1 Introduction

Since William Froude (ca. 1850) ship design is based on towing tank tests, which have been standardized beginning in 1933 by procedures developed under the auspices of the International Towing Tank Conference (ITTC) at its tri-annual meetings, e.g., recent proceeding of the 24th ITTC (2005). The advent of computer technology and computational fluid dynamics (CFD) methods offers an alternative to the traditional build and test design approach, i.e., simulation based design (SBD). It has been conjectured that SBD will offer innovative approaches to design and out-of-the box concepts with improved performance.

CFD for ship hydrodynamics is well developed with advanced capabilities for resistance and propulsion, seakeeping, and maneuvering as evidenced by the ship hydrodynamics CFD workshops, most recently CFD Tokyo (2005), and bi-annual Office of Naval Research (ONR) Symposia on Naval Hydrodynamics (SNH), most recently SNH 2006. One of the leading ship hydrodynamics CFD codes and of present interest is CFDSHIP-IOWA, which has been developed under ONR support at the Iowa Institute of Hydraulics Research (IIHR) over the past 20 years (most recently version 4 [1]). The capabilities of ship hydrodynamics CFD codes have largely been demonstrated mimicking typical towing tank tests using the experimental data for validation of the simulations, including both global and local flow variables (see, e.g., Ref. [2]). Once validated, CFD thus far has largely been used for design analysis, but with its current interdisciplinary capability including global optimization methods [3] and emerging multidisciplinary capability it assures in short time the reality of SBD. Recently, CFD has shown its usefulness for conceptual design for high-speed seafloor concepts [4].

Herein, the concept of a computational towing tank is implemented to show that CFD and SBD have a potential to aid the design process. The idea of a numerical towing tank is not new, as the name has been used for an annual European workshop, most recently Numerical Towing Tank Symposium (NUTTS) (2006); however, herein we demonstrate different possibilities and the full potential of the concept. It will be shown that both resistance and propulsion tests can be conducted in a unique and efficient manner not possible or highly difficult using a physical towing tank whereby the resistance and propulsion curves can be predicted in single computer runs, which cover the entire Froude number (Fr) range of interest. To achieve a computational towing tank for resistance, propulsion, and other applications, absolute inertial earth-fixed coordinates are applied. Most previous CFD simulations for resistance [5], propulsion [6], seakeeping [7], steady maneuvering [8], and dynamic maneuvering [9] applied relative inertial coordinates or noninertial ship-fixed coordinates [10].

2 Computational Method

The general-purpose solver CFDSHIP-IOWA-V.4 [1] solves the unsteady Reynolds averaged Navier–Stokes (RANS) or detached eddy simulation (DES) equations in the liquid phase of a free surface flow. CFDSHIP-IOWA-V.4 is briefly summarized with focus on application of the absolute inertial earth-fixed coordinates for development and implementation of the concept of the computational towing tank.

2.1 Modeling. Governing differential equations. The governing differential equations (GDEs) of motion are derived and solved in absolute inertial earth-fixed coordinates \((X,Y,Z)\) for an arbitrary moving but nondeforming control volume and solution domain, respectively. As shown in Fig. 1, the control volume is
The black control volume performs up to six degrees of freedom (6DOF) motions (surge, sway, heave, roll, pitch, and yaw). The gray control volume performs up to 3DOF (surge, sway, and yaw) motions copied from the corresponding degrees of freedom of the ship’s motions. Any number of degrees of freedom can be imposed and the rest is predicted by the 6DOF solvers, which results in captive, free, or semicaptive motions. 

The Reynolds transport theorem [11] for incompressible flow with an arbitrary nondeforming control volume moving at \( V_G \) (shown in Fig. 1) is applied with the velocity relative to the control volume defined by

\[
\overrightarrow{V}_r = \overrightarrow{V} - \overrightarrow{V}_G
\]  

where \( \overrightarrow{V} \) is the absolute velocity in \((X,Y,Z)\). The flux term is transformed to a volume integral using the Gauss divergence theorem. Taking the limit for elemental control volume for which the integrand is independent of the volume and dividing by the volume, the differential form per unit volume is obtained. Conservation of mass gives the continuity equation for \( \overrightarrow{V}_r \) as follows:

\[
\nabla \cdot \overrightarrow{V}_r = 0
\]  

Substitution of Eq. (2) into Eq. (3) with \( \nabla \cdot \overrightarrow{V}_G = 0 \) for a nondeforming control volume results in

\[
\nabla \cdot \overrightarrow{V} = 0
\]  

Conservation of momentum using the divergence operator expansion and the continuity equation, and expressing the body and surface forces per unit volume give the momentum equation

\[
\rho \left[ \frac{\partial \overrightarrow{V}}{\partial t} + (\overrightarrow{V} - \overrightarrow{V}_G) \cdot \nabla \overrightarrow{V} \right] = -\nabla (p + \gamma Z) + \frac{1}{Re} \nabla^2 \overrightarrow{V}
\]  

which is nondimensionalized using a reference velocity \( U_r^* \) (generally the maximum speed of the ship), the ship length \( L^* \), the water density \( \rho^* \), and viscosity \( \mu^* \). When the control volume is limited to a point, the physical meaning of \( \overrightarrow{V}_G \) is the local grid velocity.

Equation (5) can be transformed into the relative inertial coordinates \((X', Y', Z')\) translating at a constant velocity \( \overrightarrow{V}_C \) relative to \((X, Y, Z)\) by replacing \( \overrightarrow{V} \) with \( \overrightarrow{V} + \overrightarrow{V}_C \) and \( \overrightarrow{V}_G \) with \( \overrightarrow{V}_G + \overrightarrow{V}_C \), where \( \overrightarrow{V}_C \) and \( \overrightarrow{V}_G \) are the fluid and control volume velocities in \((X', Y', Z')\), respectively. The time derivatives in the two inertial coordinates are the same. Since the gradient, divergence, and Laplacian operators in Eq. (5) are frame invariant, the governing equations in terms of \( \overrightarrow{V}' \) in \((X', Y', Z')\) are obtained as follows:

\[
\rho \left[ \frac{\partial \overrightarrow{V}_r'}{\partial t} + (\overrightarrow{V}_r' - \overrightarrow{V}_G) \cdot \nabla \overrightarrow{V}_r' \right] = -\nabla (p + \gamma Z) + \frac{1}{Re} \nabla^2 \overrightarrow{V}_r'
\]  

Equation (5) can also be transformed into the noninertial ship-fixed coordinates by replacing \( \overrightarrow{V} \) with \( \overrightarrow{V} + \overrightarrow{V}_C \) and \( \overrightarrow{V}_G \) with \( \overrightarrow{R} + \overrightarrow{\Omega} \times \overrightarrow{r} \). For any vector, the time derivative \( \partial \overrightarrow{r} / \partial t \) in \((X, Y, Z)\) is related to its time derivative \( \overrightarrow{\tilde{r}} / \partial t \) in \((x, y, z)\) by [12]

\[
\frac{\partial \overrightarrow{r}}{\partial t} = \overrightarrow{\tilde{r}} + \overrightarrow{\Omega} \times \overrightarrow{r}
\]  

Again, since the gradient, divergence, and Laplacian operators in Eqs. (5) are frame invariant, the governing equations in terms of \( \overrightarrow{V}_r \) in \((x, y, z)\) are obtained, which have a similar form to the Navier–Stokes equations for a fixed control volume in absolute inertial coordinates (standard NS equations hereinafter), i.e., Eq. (5) with \( \overrightarrow{V}_G = 0 \), except with an additional body-force term [11]

\[
\rho \left[ \frac{\partial \overrightarrow{V}_r}{\partial t} + \overrightarrow{V}_r \cdot \nabla \overrightarrow{V}_r \right] = -\rho \overrightarrow{g}_r - \nabla (p + \gamma Z) + \frac{1}{Re} \nabla^2 \overrightarrow{V}_r
\]  

\[
\overrightarrow{g}_r = \overrightarrow{g} + 2 \overrightarrow{\Omega} \times \overrightarrow{V}_r + \overrightarrow{\Omega} \times (\overrightarrow{\Omega} \times \overrightarrow{r}) + \overrightarrow{\Omega} \times \overrightarrow{r}
\]  

The GDEs for continuity and momentum in \((X, Y, Z)\) and \((X', Y', Z')\) are transformed from the physical domain in Cartesian coordinates \((X, Y, Z, t)\) to the computational domain in nonorthogonal curvilinear coordinates \((\xi, \eta, \zeta, \tau)\) using the chain rule without involving grid velocity for the time derivative transformation [5]

\[
\frac{1}{J} \frac{\partial}{\partial \xi} (b_i \overrightarrow{U}_i) = 0
\]  

\[
\frac{\partial \overrightarrow{U}_i}{\partial \tau} + \frac{1}{J} b_i (\overrightarrow{U}_i - \overrightarrow{U}_{\Omega i}) \frac{\partial \overrightarrow{U}_i}{\partial \xi} = \frac{1}{J} \frac{\partial}{\partial \xi} \left( \frac{b_i b_j}{J Re_{\Omega i}} \frac{\partial \overrightarrow{U}_i}{\partial \xi} \right)
\]

\[
+ \frac{b_i \overrightarrow{\Omega}_j \overrightarrow{U}_j}{J \partial \xi} + \frac{b_i \overrightarrow{\Omega}_j \overrightarrow{U}_j}{J \partial \xi} + S_i
\]  

Equations (10) and (11) are identical to Warsi [13], who transformed the standard NS equations in \((X, Y, Z, t)\) to a fixed computational domain \((\xi, \eta, \zeta, \tau)\) with the inclusion of the grid velocity \( \partial \xi / \partial \tau \) in the time derivative transformation. In the present derivation of Eqs. (5) and (6) the solution domain and control volume are coincident with each other, which conceptually better shows the relationship between the moving but nondeforming control volume and solution domain and additionally provides the continuous \( \overrightarrow{V}_G / \overrightarrow{V}_r \) form of the NS equations. Compared with Eq. (6), Eq. (5) allows nonconstant \( \overrightarrow{V}_C \). When \( \overrightarrow{V}_C = 0 \), Eq. (6) reduces to Eq. (5). Transformation of the continuous \( \overrightarrow{V}_G \) form of the NS equations in \((X, Y, Z)\) into \((x, y, z)\) clearly shows the difference of
GDEs using the absolute inertial earth-fixed or noninertial ship-fixed coordinate systems. Compared with Eq. (8), application of Eq. (5) simplifies the specification of boundary conditions, saves computational cost by reducing the solution domain size, and can be easily applied to simulate multi-objects such as ship-ship interactions. In general, implementation of Eq. (5) to simulate captive, semicaptive, or full 6DOF ship motions is straightforward.

Turbulence and free surface modeling. The turbulent kinetic energy $k$ is computed using a blended $k-\omega/k-\varepsilon$ model [14]. A single-phase level set method is used [1]. The location of the free surface is given by the zero level set of the level set function $\Phi$, set to the positive distance to the free surface in water and negative in air. Since the free surface is a material surface, the equation for the level set function is

$$\frac{\partial \Phi}{\partial t} + (U_j - U_{(j)}) \frac{\partial \Phi}{\partial x_j} = 0$$

(12)

The jump conditions for velocity and the dimensionless piezometric pressure at the free surface are given by Eqs. (15) and (18) in Ref. [1], respectively. Transport of the level set function with Eq. (12) does not guarantee that $\Phi$ remains a function as the computation evolves. To resolve this difficulty, an implicit extension is performed every time $\Phi$ is transported. The first neighbors to the free surface are reinitialized geometrically. The rest of the domain is reinitialized using an implicit transport of the level set function.

Propeller model. For propelled simulations, a simplified body-force model is used to prescribe axisymmetric body force with axial and tangential components [15]. The radial distribution of forces is based on the Hough and Ordway circulation distribution, which has zero loading at the root and tip. A vertex-based search algorithm is used to determine which grid-point control volumes are within the actuator cylinder. The force and torque of each propeller are projected into the noninertial ship-fixed coordinates and used to compute an effective force and torque about the center of rotation, which is coincident to the center of gravity in this study.

Calculations and transformations of forces and moments. The dynamic pressure force $F_{pi}$ and the hydrostatic pressure (buoyancy) force $B_i$ in the absolute inertial earth-fixed coordinates for the ship are computed from

$$F_{pi} = - \int S_w p d\zeta$$

(13)

$$B_i = \int_S Z \frac{\dot{V}}{Fr^2} d\zeta$$

(14)

where $q$ is the outward pointing area vector and $S_w$ is the wetted surface area. The friction forces are computed using the velocity in the absolute inertial earth-fixed coordinates

$$F_{fi} = \frac{1}{2} \text{Re} \int_{S_w} (\nabla V + \nabla V^\circ) \cdot d\zeta$$

(15)

Thus the total force is

$$E_i = E_{fi} + F_{pi} + B_i$$

(16)

The total moments are found from integrating the elemental forces with the distance to the center of gravity $E_{CG}$ as

$$M_i = \int_{E_{CG}} \left\{ \left( \frac{\nabla V^\circ + \nabla V}{2 \text{Re}} \right) - \left( \frac{p}{\text{Fr}} \right) I \right\} \cdot d\zeta$$

(17)

$F_i$ and $M_i$ are then projected into the noninertial ship-fixed coordinates $(x, y, z)$ using

$$E_b = J_1(E_i)$$

$$M_b = J_1(M_i)$$

(18)

The matrix $J_1$ transforms any vector in $(X, Y, Z)$ to a vector in $(x, y, z)$ as follows:

$$J_1 = \begin{bmatrix}
\cos \psi \cos \theta & \sin \psi \cos \phi + \sin \phi \sin \theta \cos \psi & \cos \psi \sin \phi \sin \theta - \sin \theta \\
\sin \psi \cos \phi - \sin \phi \sin \theta \cos \psi & \cos \psi \cos \phi + \sin \phi \sin \theta \sin \psi & \sin \phi \cos \theta \\
\sin \phi \cos \theta & - \sin \phi \cos \phi \sin \psi & \cos \phi \cos \psi
\end{bmatrix}$$

(19)

where $\phi$, $\theta$, and $\psi$ are the Euler angles for roll, pitch, and yaw, respectively. The three components of $E_b = (E_{b1}, E_{b2}, E_{b3})$ are the surge, sway, and heave forces, respectively. The three components of $M_b = (M_{b1}, M_{b2}, M_{b3})$ are the roll, pitch, and yaw moments, respectively.

6DOF module. The evolution of the location and the attitude of the ship are computed solving the rigid body equations in $(x, y, z)$. The translation of the ship is described by $V_b = (u_b, v_b, w_b)$, where $u_b$, $v_b$, and $w_b$ are the surge, sway, and heave velocities of the ship with respect to $(x, y, z)$. The ship rotation is described using the Euler angles $\sigma = (\phi, \theta, \psi)$, whose rates of change are computed using $\dot{\Omega}_b = (\dot{\Omega}_b, \dot{\Omega}_b, \dot{\Omega}_b)$, where $\dot{\Omega}_x$, $\dot{\Omega}_y$, and $\dot{\Omega}_z$ are the three angular velocity components with respect to $(x, y, z)$ due to roll, pitch, and yaw, respectively.

$$\dot{\sigma} = J_1 \dot{\Omega}_b$$

(20)

The rigid body equations are

$$m\ddot{u}_b = \ddot{u}_b + u_b \dot{\Omega}_b^2 = F_{bfx} + F_{bpx} + B_{bx} + W_{bx} + P_{bx} - \text{SFC}_{bx}$$

(21)

$$m\ddot{v}_b = \ddot{v}_b + v_b \dot{\Omega}_b^2 = F_{bfx} + F_{bpx} + B_{bx} + W_{bx} + P_{bx} - \text{SFC}_{bx}$$

(22)

$$m\ddot{w}_b = \ddot{w}_b + w_b \dot{\Omega}_b^2 = F_{bfx} + F_{bpx} + B_{bx} + W_{bx} + P_{bx} - \text{SFC}_{bx}$$

(23)

$$I_x \ddot{\Omega}_x + (I_x - I_z) \dot{\Omega}_z \dot{\Omega}_x = M_{bfx} + M_{bpx} + M_{bPTx} + M_{bPQx}$$

(24)

$$I_y \ddot{\Omega}_y + (I_y - I_z) \dot{\Omega}_z \dot{\Omega}_y = M_{bfy} + M_{bpy} + M_{bPTy} + M_{bPQy}$$

(25)
\[ I_d \ddot{\Omega}_c / \partial t + (I_y - I_z) \ddot{\Omega}_c \dot{\Omega}_x = M_{bF_c} + M_{bpc} + M_{bh} + M_{bPT} + M_{bPQ} \tag{26} \]

where \( m \) is the mass of the ship, \( B_b = (B_{bx}, B_{by}, B_{bz}) \), \( \bar{W}_b = (\bar{W}_{bx}, \bar{W}_{by}, \bar{W}_{bz}) \), and \( P_{b} = (P_{bx}, P_{by}, P_{bz}) \) are the buoyancy, gravity, and propeller thrust forces in \((x, y, z)\), respectively. \( SFC_b = (SFC_{bx}, SFC_{by}, SFC_{bz}) \) is the skin friction correction in self-propulsion simulations. The moments and motions due to \( SFC_b \) are considered by subtracting \( SFC_{c} \) from the propeller thrust. \( I_x, I_y, \) and \( I_z \) are the moments of inertia with respect to the principal axes of rotation, which are assumed to be coincident to the \( x, y, \) and \( z \) coordinates. \( M_{bF_c} = (M_{bh}, M_{bpc}, M_{bh}) \), \( M_{bPT} = (M_{bPT}, M_{PQ}, M_{PQ}) \), and \( M_{bPQ} = (M_{bPQ}, M_{bPQ}, M_{bPQ}) \) are the moments caused by the buoyancy, the propeller thrust force, and the propeller torque in \((x, y, z)\), respectively. It is assumed that the center of rotation and the towing point coincide with the center of gravity. Thus the moments caused by the towing force and gravity are zero. Any number of degrees of freedom can be imposed and the rest is predicted, which results in captive, free, or semicaptive motions. \textsc{cfdsihip-iowa-v4.4} only solves rigid body equations for the predicted degrees of freedom using a predictor/corrector implicit approach. The prescribed motions for position, translation velocities, and Euler angles are specified as functions of time in the overset information is read from a binary file produced by \textsc{suggar}, the grids are moved according to the motions resulting from the \textsc{6dof} predictor/corrector steps, and the transformation metrics and grid velocity are computed. Then the \( k = o \) equations are solved implicitly followed by the level set function transport and reinitialization. With the new location of the free surface the pressure gradient is computed and the pressure-implicit split-operator (PISO) [17] method is used to enforce a divergence free velocity field where the pressure Poisson equation (Eq. (35)) is solved using the \textsc{petsc} toolkit [18]. Once the velocity field is obtained, the forces and moments are weighted with coefficients provided as a preprocessing step by \textsc{usurp} [19] used to properly compute area and forces on overlap regions for a ship hull with appendages.

Then the residuals of the nonlinear iterations are evaluated. If the residuals of all variables drop to \( 10^{-5} \) indicating converged for that time step, the motions are predicted for the next time step using Eq. (23) in Ref. [7]. \textsc{suggar} is called to compute the interpolation given the new location of the moving grids. If the nonlinear iteration is not converged, then the motions are corrected using Eq. (22) in Ref. [7]. \textsc{suggar} is called, and a new nonlinear iteration starts.

A message passing interface (MPI)-based domain decomposition approach is used. Two to five nonlinear iterations are required for convergence of the flow field equations within each time step. Convergence of the pressure equation solver is reached when the residual imbalance of the Poisson equation drops six orders of magnitude. All other variables are assumed converged when the linear iteration residuals drop to \( 10^{-5} \).

3 Implementation of Absolute Inertial Coordinates for Resistance and Propulsion

3.1 Single Fr Resistance and Propulsion. For a single Fr resistance simulation, the ship is towed at a speed based on the Fr, i.e., \( u_t = V_b = Fr \sqrt{gL} \). The rigid body equations (21)–(26) are simplified to

\[ u_b = (v_b \sin \theta + V_c) \cos \theta \tag{36} \]

\[ v_b = 0 \tag{37} \]

\[ m(\ddot{u}_b / \partial t + u_b \dot{\Omega}_x) = F_{bF_c} + F_{bpc} + B_{bh} + W_{bc} \tag{38} \]

\[ 0 = M_{bF_c} + M_{bpc} + M_{bh} \tag{39} \]

\[ I_y \ddot{\Omega}_c / \partial t + M_{bF_c} + M_{bpc} + M_{bh} \tag{40} \]

\[ 0 = M_{bF_c} + M_{bpc} + M_{bh} \tag{41} \]

The position vectors and grid velocity are computed by

\[ R = R_{initial} + V_c T_{b0} + \int_0^T \bar{w} \hat{k} dt \tag{42} \]
The simulation converges when the resistance coefficients reach asymptotic values, after which \( \tilde{w}_i = \Omega_z = 0 \) results from the force and moment balances. The position vectors and grid velocities are further simplified by \( \dot{w}_i = 0 \). The background grid is prescribed to surge at the same speed of the ship but without sinkage or trim.

The traditional way for a self-propulsion simulation is to mimic the experiment using a root finder module to determine the propeller revolution per second (RPS) \( R_{\text{PS}} \). However, this approach is disadvantageous from a numerical simulation point of view. First, even for a fixed RPS, the computations of the sinkage and trim are slow to converge. A root finder that can change RPS above and below the self-propelled point is needed and results in a very expensive computation; secondly and most important, this approach can only be used for a single \( \text{Fr} \), which requires many runs if a large range of \( \text{Fr} \) is of interest, and local features on the resistance curve may be lost if points were not chosen at those specific speeds. Herein, self-propulsion simulation using the absolute inertial earth-fixed coordinates is used to mimic a real self-propelled ship. For a typical steady-state computation, the ship speed is initially set to zero. When the propeller is turned on, the ship accelerates until it reaches a constant speed when the resistance force balances the thrust force from the propeller. The RPS is specified based on the target \( \text{Fr} \). The self-propelled \( \text{Fr} \), resistance coefficient, sinkage, and trim are all predicted.

A. Set RPS based on the target \( \text{Fr} \).
B. Estimate wake factor \( 1 - \tilde{w}_t \) from instantaneous velocity \( U_{\text{ship}} \).
C. Compute instantaneous advance coefficient \( J = U_{\text{ship}}(1 - \tilde{w}_t)/(nD_p) \).
D. Get thrust coefficient \( K_T \) and torque coefficient \( K_Q \) from \( J \) based on the open water curve.
E. Compute the weight that needs to be added for the instantaneous \( U_{\text{ship}} \); this weight is subtracted from the thrust force. This step is necessary if model-scale simulations will be used to predict full-scale conditions or if the experimental data to be simulated followed the same procedure.
F. Since \( K_T \) and \( K_Q \) are maximum for \( J = 0 \), the ship will accelerate.
G. Go to step B, repeat the above procedure until the final ship velocity is achieved.

The rigid body equations are simplified to

\[
m(\tilde{\gamma}X_i/\partial t + \tilde{w}_i\dot{\Omega}_i) = F_{fX} + F_{bX} + B_{bX} + W_{bX} + P_{bX} - \text{SFC}_{bX} \tag{44}
\]

\[
0 = F_{fY} + F_{bY} + B_{bY} + W_{bY} + P_{bY} - \text{SFC}_{bY} \tag{45}
\]

\[
m(\tilde{\gamma}X_i/\partial t - u_i\dot{\Omega}_i) = F_{fZ} + F_{bZ} + B_{bZ} + W_{bZ} + P_{bZ} - \text{SFC}_{bZ} \tag{46}
\]

\[
0 = M_{fX} + M_{bX} + M_{b\delta X} + M_{b\delta T_Y} \tag{47}
\]

\[
I_{i}(\tilde{\gamma}\Omega_i/\partial t = M_{fY} + M_{bY} + M_{b\delta Y} + M_{b\delta T_X} + M_{b\delta T_Y} + M_{b\delta T_Z} \tag{48}
\]

Note that the net moments caused by the two propellers' torque along the \( x \) and \( z \) axes are zero since the propellers are symmetric about the center plane (i.e., \( x-z \) plane) with counter-rotation direction. The position vector is computed as

\[ Y_i = X_i + Z_i \tag{43} \]
\[ R = R_{\text{initial}} + \int_0^T (u \dot{z} + w \dot{\phi}) dt \]  

with the grid velocity computed as in Eq. (43). The self-propulsion simulation converges when \( \frac{\partial R}{\partial t} = 0 \).

The disadvantage of performing single Fr simulations is that to achieve a wide range of flow conditions will require many runs that are very time-consuming in terms of CPU and case setup.

### 3.2 Full-Fr Curves of Resistance and Propulsion

To increase the flexibility on diagnosing flow physics for a large range of flow conditions, this study develops a computational towing tank procedure that allows for a single run for the prediction of resistance and propulsion curves using the absolute inertial earth-fixed coordinates. The basic assumption is that at every instant the flow field is in a quasi steady-state, by virtue of imposing a very small acceleration so as to cover the desired velocity range during the computation. From the mathematical point of view, this means that the time derivatives of GDEs are negligible in a time-average sense. In many situations the ship transom is wet for low speeds and there is continuous vortex shedding from the ship transom corner, which forms naturally unsteady flows.

The shedding of the vortices can induce ship motions, oscillations of forces and moments, and deformation of the free surface. An unsteady constant speed computation can capture all these phenomena. The computational towing tank, being an unsteady solution of an accelerating ship, can also capture these effects as long as the characteristic times of these phenomena are much smaller than the characteristic ship acceleration time. This condition is easy to meet for vortex shedding and the free surface fluctuations, being high-frequency phenomena, but may be difficult for ship motions that have a long period. In summary, the computational towing tank can be regarded as basically a controlled transient calculation method.

The procedure starts with an appropriate choice of a reference velocity to nondimensionalize the equations of motion. For convenience we choose to use the ship speed at the point of maximum Fr to be achieved in the computation; thus

\[ \frac{U_{\text{ship}}}{\sqrt{gL}} \]

and thus the instantaneous ship Fr is

\[ Fr = U_{\text{ship}} \left( \frac{1}{\sqrt{gL}} \right) \]

Then the following has been carried out.

A. Specify the ramp time \( TR \) for either the ship speed (resistance curve) or the RPS of the propeller (powering curve) to control the acceleration of the ship. The ramp function could be a linear, quadratic, or other continuous smooth function. A linear function is used herein for the ship speed changing from 0 to 1 as follows:

\[ V_C = U_{\text{ship}} \left( \frac{t}{T_R} \right) \]

and the propeller RPS as

\[ \text{RPS} = \text{RPS}_{\text{max}} + \left( \frac{t}{T_R} \right) (\text{RPS}_{\text{max}} - \text{RPS}_{\text{min}}) \]

B. Specify the time-step size based on \( TR \), acceptable error, and computational cost.

C. Perform computation (note \( V_C \) in Eq. (42) needs to be inside the integral as it is a function of time).

D. Report the resistance curve, powering curve, and ship motions during the computation.

E. Estimate the total distance traveled and the time needed. For a linear ramp function the total dimensional distance the ship travels can be computed from

\[ D^* = L^* \int_0^{T_R} U_{\text{ship}} dt = L^* \int_0^{T_R} \frac{U_{\text{ship}}}{T_R} dt = \frac{T_R L^*}{2} \]

The total dimensional time \( T^* \) required for conducting such an experiment can be estimated by

\[ T^* = \frac{T_R L^*}{U_{\text{max}}^*} = \frac{T_R}{U_{\text{max}}^*} \sqrt{\frac{L^*}{gL^*}} \]

F. Analyze the free surface wave elevation, ship motions, forces and moments, and other flow details at any Fr of interest.

The optimum \( T_R \) involves a trade-off between computation speed and accuracy. The characteristic times of the main unsteady processes related to the acceleration of the ship are the time necessary to develop the boundary layer and the Kelvin wave pattern on the ship's wave pattern in the vicinity of the ship. Based on the authors’ previous CFD simulations, these two times are in the order of the time needed to cover one ship length. So during this time we would like the change of ship speed to be small, to obtain essentially constant speed values of resistance coefficient, sinkage, and trim. If the acceleration to the full speed (\( U_{\text{max}}^* = 1 \) corresponds to \( Fr_{\text{max}} \)) is linear as in Eq. (53), then to cover one ship length the ship needs to travel a time \( \Delta t \) obtained from

\[ 1 = \int_{t}^{t+\Delta t/2} \frac{t}{T_R} dt = \frac{t}{T_R} \Delta t \]

and using Eqs. (53) and (57) we obtain the change in Fr while the ship traveled one ship length

\[ \Delta Fr = \frac{Fr_{\text{max}}^2}{T_R Fr} \]

which shows that the change in Fr while the ship travels one ship length is proportional to the inverse of \( T_R \) and the instantaneous Fr. Figure 3 shows \( \Delta Fr \) as a function of Fr for the two ramp times used in the resistance simulation. It clearly demonstrates that the single-curve procedure will not work for very low Fr (Fr<0.1), unless an impractically large \( T_R \) is used. This conclusion may be misleading since at low Fr the near-field wave pattern will develop faster than in one ship length. In addition, since we are slowly increasing the ship speed, the solution is only changing slightly when the ship travels one ship length, and thus the characteristic time adopted for this analysis may be too long. Figure 3 also shows that \( T_R \) has little effect for Fr>0.4. Further discussion on the subject is presented in the next sections.
4 Validation for High-Speed Transom Ship

Following the procedure of computational towing tank, single-run full-Fr curves for resistance and self-propulsion are performed for the Athena bare hull with a skeg (BH hereinafter) and a fully appended twin-screw Athena (AH hereinafter), respectively. As shown in Table 1 for BH, two ramp times are used to investigate the sensitivity of different ramp times on the prediction of resistance coefficients and motions, with the long ramp time twice the short one. For the long ramp, a larger time step is used to make the whole curve run at an affordable computational cost. For AH, the long ramp time is used to slowly increase the propeller RPS that will accelerate the ship for a whole range of speeds (Fr = 0–1). The ship lengths that the ship travels for each simulation are also summarized and compared in Table 1 discussed later.

4.1 Experimental Data and Simulation Conditions. Towing tank experiments for resistance of the high-speed transom stern ship Athena with a skeg were conducted on a 1/8.25-scale model by Jenkins [20]. Measurements of resistance, wave resistance, sinkage and trim, and wave elevation along the hull and at several stations were made for Fr ranging from 0.28 to 1.00. The same model but appended with twin rudders, stabilizers, skeg, shafts, and struts was used to measure resistance and powering characteristics [21]. Experimental data include sinkage and trim at 17 different Fr ranging from 0.336 to 0.839, open water curves for the propellers (model 4710), with the corresponding propeller RPS, thrust deduction, and wake factors. Details of the simulation conditions are summarized in Table 1.

4.2 Domain, Grid Topology, and Boundary Conditions. Figures 4 and 5 show the grid topologies for the BH and AH cases, respectively. Body-fitted “O” or “double-O” type grids are generated for the ship hull and appendages, with extensive use of overset grids. Cartesian background grids are used to better specify the boundary conditions from the ship hull and refined near the free surface to resolve the wave field. To resolve the wake, a Cartesian refinement block is included. Though shown in full domain in Fig. 5, only half domain has been computed taking advantage of the symmetry of the problem about the center plane y = 0. Pitch and heave motions are allowed to predict the final sinkage and trim. Only the grids attached to the hull move in these two degrees of freedom, but all grids move with the surge speed of the ship, following the forward motion. The boundary conditions for all the variables are shown in Table 2.

![Fig. 4 Simulation domain, grids, and boundary conditions for Athena bare hull with skeg](image-url)
5 Verification and Validation

A verification and validation (V & V) study is conducted for the towed BH for resistance coefficient, sinkage, and trim at Fr = 0.48. The V & V methodology and procedures follow Stern et al. [22] with an improved correction factor formula for grid/time-step uncertainty estimates [23]. The simulation numerical uncertainty $U_{SN}$ is composed of iterative $U_I$, grid $U_G$, and time-step $U_T$ errors

$$U_{SN} = U_I + U_G + U_T$$

(59)

The comparison error $E$ is defined by the difference between the data $D$ and simulation $S$ values as follows:

$$E = D - S$$

(60)

The validation uncertainty is defined as

$$U_V = \sqrt{U_{SN}^2 + U_D^2}$$

(61)

where $U_D$ is the uncertainty of the experimental fluid dynamics (EFD) data. When $E$ is within $\pm U_V$, solutions are validated at the intervals of $U_V$. Validation of the full-Fr curves for resistance coefficient, sinkage, and trim for BH is conducted by assuming that the same interval of $U_D$ based on the grid study for BH at Fr = 0.48 is valid for the whole Fr range.

Table 3 summarizes all the grids used for V & V for BH at Fr = 0.48, with $y_1^*$ of the first grid point away from the wall. The primary objective of this study is full-Fr curves of resistance and propulsion. To maintain an affordable computational cost with a reasonable accuracy, grid 5 is picked for most of the results. Investigation of the issue of approaching the asymptotic range is conducted by systematically refining grid 5 to grid 1 and coarsening grid 5 to grid 7 with a grid refinement ratio $r_G = 2^{1/4}$. This allows nine sets of grids for verification including five sets with $r_G = 2^{1/4}$, three sets with $r_G = 2^{1/2}$, and one set with $r_G = 2^{3/4}$.

For AH, a total of $2.2 \times 10^6$ grid points are split into 24 blocks with an average of 90,891 grid points per processor.

5.1 Iterative and Statistical Convergence. Parametric studies on the nonlinear iterations for each time step ensure iterative convergence at each time step. The simulation is then advanced to the next time step. Results show that by increasing the nonlinear iterations from 4 to 5, the difference for the resistance coefficient $C_{TX} = C_{TX} + C_{pX}$ is less than 1%. Simulations used five nonlinear iterations. The distribution of iterative errors $0.1\% S_{line} \leq U_I \leq 0.6\% S_{line}$ for grids 1 to 7 is shown in Figs. 6(b) and 6(d) for resistance coefficients and motions of BH, respectively. $U_I$ are of the same order of magnitude for all grids, which suggests that it is mainly determined by the iterative method applied and independent of grid resolutions. The criterion for statistical convergence of $C_{TX}$ is that the oscillations/fluctuations of the running mean are less than 0.4% of the mean value, ensured for all cases run.
5.2 Verification, Validation, and Discussion of Integral Variables at Fr=0.48.

Tables 4 and 5 show verification and validation studies for monotonically converged solutions for $C_{TX}$ and ship motions, respectively. $RG$ is defined as the ratio of solution changes for medium-fine and coarse-medium solutions, which defines the convergence conditions. $CG$ is the correction factor and $PG$ is the observed order of accuracy. $1-CG$ indicates how far the solution is from the asymptotic range where $CG = 1$.

$C_{TX}$ monotonically converges on grids (2,4,6), (1,3,5), (4,5,6),

<table>
<thead>
<tr>
<th>Grids</th>
<th>Refinement ratio</th>
<th>$RG$</th>
<th>$PG$</th>
<th>$1-CG$</th>
<th>Correction factor</th>
<th>GCI</th>
<th>$E$ (%)</th>
<th>$U_Y$ (%)</th>
<th>$U_D$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2, 4, 6</td>
<td>$\sqrt[3]{2}$</td>
<td>0.63</td>
<td>1.32</td>
<td>0.42</td>
<td>4.90</td>
<td>3.34</td>
<td>1.83</td>
<td>5.21</td>
<td>1.5</td>
</tr>
<tr>
<td>1, 3, 5</td>
<td>$\sqrt[3]{2}$</td>
<td>0.40</td>
<td>2.66</td>
<td>-0.51</td>
<td>3.59</td>
<td>0.72</td>
<td>2.10</td>
<td>3.96</td>
<td>1.5</td>
</tr>
<tr>
<td>4, 5, 6</td>
<td>$\sqrt[4]{2}$</td>
<td>0.97</td>
<td>0.16</td>
<td>0.93</td>
<td>125.2</td>
<td>52.7</td>
<td>0.24</td>
<td>125.5</td>
<td>1.5</td>
</tr>
<tr>
<td>3, 4, 5</td>
<td>$\sqrt[4]{2}$</td>
<td>0.80</td>
<td>1.27</td>
<td>0.41</td>
<td>7.23</td>
<td>4.98</td>
<td>1.23</td>
<td>7.47</td>
<td>1.5</td>
</tr>
<tr>
<td>2, 3, 4</td>
<td>$\sqrt[3]{2}$</td>
<td>0.60</td>
<td>2.98</td>
<td>-0.64</td>
<td>8.73</td>
<td>1.07</td>
<td>1.83</td>
<td>9.02</td>
<td>1.5</td>
</tr>
<tr>
<td>1, 2, 3</td>
<td>$\sqrt[2]{2}$</td>
<td>0.50</td>
<td>4.00</td>
<td>-1.42</td>
<td>1.11</td>
<td>0.58</td>
<td>2.10</td>
<td>1.88</td>
<td>1.5</td>
</tr>
</tbody>
</table>

Fig. 6 Verification for resistance coefficients and motions for Athena bare hull with skeg (Fr=0.48): (a) resistance coefficients, (b) relative change $\varepsilon_n=[|S_n-S_{n+1}|/S_1]\times 100$ and iterative errors for resistance coefficients, (c) sinkage and trim, and (d) relative change $\varepsilon_n$ and iterative errors for sinkage and trim.
Table 5 Verification and validation study for motions of Athena bare hull with skeg (Fr = 0.48). $U_g$ is $%S_1$, $%S_2$, or $%S_4$. Others are $%EFD$ (sinkage is −0.00341 and trim is 0.7105 deg).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Grids</th>
<th>Refinement ratio</th>
<th>$R_g$</th>
<th>$P_g$</th>
<th>$1 − C_g$</th>
<th>$U_g$ (%)</th>
<th>$E$ (%)</th>
<th>$U_V$ (%)</th>
<th>$U_D$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sinkage</td>
<td>1, 3, 5</td>
<td>$\frac{5}{1}$</td>
<td>0.31</td>
<td>3.4</td>
<td>−1.25</td>
<td>1.80</td>
<td>1.5</td>
<td>29.4</td>
<td>29.3</td>
</tr>
<tr>
<td>Sinkage</td>
<td>2, 3, 4</td>
<td>$\frac{2}{1}$</td>
<td>0.13</td>
<td>12</td>
<td>−15.92</td>
<td>1.37</td>
<td>0.6</td>
<td>29.3</td>
<td>29.3</td>
</tr>
<tr>
<td>Trim</td>
<td>1, 3, 5</td>
<td>$\frac{5}{1}$</td>
<td>0.48</td>
<td>2.13</td>
<td>−0.09</td>
<td>4.67</td>
<td>13</td>
<td>9.66</td>
<td>8.1</td>
</tr>
<tr>
<td>Trim</td>
<td>4, 5, 6</td>
<td>$\frac{4}{1}$</td>
<td>0.86</td>
<td>0.89</td>
<td>0.60</td>
<td>42.87</td>
<td>3.7</td>
<td>45.2</td>
<td>8.1</td>
</tr>
<tr>
<td>Trim</td>
<td>2, 3, 4</td>
<td>$\frac{2}{1}$</td>
<td>0.53</td>
<td>3.69</td>
<td>−1.16</td>
<td>8.91</td>
<td>12</td>
<td>12.81</td>
<td>8.1</td>
</tr>
</tbody>
</table>

(3,4,5), (2,3,4), and (1,2,3), oscillatory diverges on grids (5,6,7), and monotonically diverges on grids (3,5,7) and (1,4,7). All the monotonically or oscillatorily diverged solutions involve the coarsest grid 7, which is likely due to the insufficient grid resolution. As shown in Fig. 6(a), $C_{TX}$ and $C_{PX}$ for grid 7 do not follow the trend as shown for grids 6–1. Compared to resistance coefficients, sinkage and trim are more difficult to converge in grid. Sinkage monotonically converges on grids (1,3,5) and (2,3,4) and monotonically diverges on all other grid studies. Trim monotonically converges on grids (1,3,5), (4,5,6), (2,3,4), and (1,2,3), monotonically diverges on grids (1,4,7), (3,5,7), (2,4,6), and (3,4,5), and oscillatorily diverges on grids (5,6,7).

As shown in Table 4, solutions for $C_{TX}$ are far away from the asymptotic range as $1 − C_g$ oscillates within a large range of values $−1.42 ≤ 1 − C_g ≤ 0.93$. Grid study (3,4,5) is closest to the asymptotic range. Excluding the highest $U_g$ on the coarse grids (4,5,6) and the lowest $U_g$ on the finest grids (1,2,3), the average $U_g$ is 6.11% and the average $U_V$ at the intervals of 1.05% and 29.4%, respectively. Further grid refinement does not reduce the interval of validation since $U_{SN} ≤ U_D$. Although trim has a larger $U_g$, $E$ (10.43% on average) is much larger than that of sinkage and trim is only validated on grids (4,5,6) and (2,3,4) at an average interval of 29% for $U_V$.

5.3 Validation of Full-Fr Resistance Curve Simulations.

Figure 7 shows resistance coefficient, sinkage, and trim BH predictions using the single-run approach with ramp times of 50 and 100 dimensionless seconds, and experimental data for $0 < Fr < 1$. Analysis of Eq. (58) shows that the error on the Fr evaluation is larger at lower speeds. As mentioned earlier, it is reasonable to think that since the ship is accelerated slowly, the solution at any speed starts from a solution that is very close, and thus the one ship length necessary for the boundary layer and wave field to develop evaluated using Eq. (58) may be largely underestimated. This appears to be confirmed by Fig. 7. It shows that for the acceleration ramp time of 100 dimensionless seconds the solution is close to that obtained through steady-state computations and experimental data except for $0.4 < Fr < 0.55$. Beyond Fr=0.55, there is no noticeable difference of resistance coefficient between the two ramps. As shown in Fig. 7, the sinkage matches the experimental data well for both ramps with the long ramp a little closer. For Fr<0.35, the trim predicted by the long ramp agrees very well with the EFD data. For Fr>0.35, all trim values are underpredicted by 10%, possibly due to the different levels of convergence compared with steady-state computations as the latter agrees well with the EFD data. Since no experimental information was available, the authors used $Z_{CG}=0.00664$ for the center of gravity, which is chosen by testing a range of $Z_{CG}$ values for Fr=0.48 and comparing resistance coefficient, sinkage, and trim with experimental data. $X_{CG}$ equals to zero as the geometry is symmetric with respect to the center plane $y=0$. $X_{CG}=0.5688$ is calculated based on the balance of pitching moment for static conditions. Two steady-state cases for BH at Fr=0.48 and Fr =0.8 are computed by towing the ship at a constant speed and predicting resistance coefficient, sinkage, and trim. In addition, six steady-state solutions are available in literature [24] computed fixing sinkage and trim to experimental values. Predictions of these CFD computations agree very well with the EFD data.

Distributions of the comparison errors and validation uncertainties $(E \pm U_V)$ versus Fr are shown in Fig. 8 for $C_{TX}$ sinkage, and trim for the two different $T_p$. The average comparison errors $\bar{E}$, average validation uncertainties $U_{U_{TX}} U_{U_D}$ [20], $U_{SN}$ and $U_{SN} U_D$ are presented in Table 6. $U_g$ of grids (3,4,5) and (1,3,5) are selected for resistance coefficient and motions, respectively.
the increase of Fr, the resistance coefficient $E$ shifts from $-5\%$ to $+5\%$ and remains small for $Fr>0.6$. The sinkage $E$ shows oscillations around zero with the maximum at $Fr=1$. The smallest $E$ for trim is for $Fr<0.4$. It reaches the maximum $20\%$ at $Fr=0.4$ followed by a decrease for $0.41<Fr<0.65$ and a gradual increase for $Fr>0.65$. Compared to the short ramp time, the long ramp time helps reduce $|E|$ by $34.4\%$, $4.9\%$, and $8.6\%$ for resistance coefficient, sinkage, and trim, respectively. For both ramp times, resistance coefficient and sinkage are validated for the whole Fr range at $U_V$ of $8.5\%$ and $34.3\%$ of the EFD data, respectively. Trim is only validated for $Fr<0.4$ at $U_V$ of $7.0\%$. $U_{SN}/U_D$ are $5.6$, $0.026$, and $0.54$ for resistance coefficient, sinkage, and trim, respectively.

For the long ramp time at $Fr=0.48$, $|E|$ and $U_V$ for $C_{TN}$ are $2.9\%$ and $7.5\%$, respectively, compared with the smaller average $|E|$ (2%) and $U_V$ (5.5%) for a single $Fr$ V & V. $E$ and $U_V$ for sinkage are $4\%$ and $34.3\%$, respectively, compared with the smaller average $|E|$ (1.05%) and $U_V$ (29.4%) for a single $Fr$ V & V. $E$ and $U_V$ for trim are $13.7\%$ and $7.0\%$, respectively, compared to a smaller average $|E|$ (10.43%) and a much larger $U_V$ (29%) for a single $Fr$ V & V. $E$ and $U_V$ for $C_{TN}$ are computed based on the EFD data. However, to avoid misleading values for $E$ and $U_V$ due to very small magnitudes for high Fr sinkage and low Fr trim, all percentages of the sinkage and trim for the full-Fr curve are computed based on their dynamic range, of which $U_D$ for sinkage is large due to the small dynamic range.

Using Eq. (55), the total distance the ship travels are 25 and 50 ship lengths for the ramp times of 50 and 100 dimensionless seconds, respectively. The length of the model Athena R/V is $L=5.69$ m. To run 25 and 50 ship lengths in experiments will require $142.25$ m and $284.5$ m, respectively. Using Eq. (56) and noting $Fr_{max}=1.0$ for BH, the total time required for conducting such an experiment are $38.1$ s and $76.2$ s for the CFD ramp times of 50 and 100 dimensionless seconds, respectively. In addition, a variably controlled speed would be necessary for the carriage, which may be cumbersome.

A typical single Fr number run takes 51 wallclock hours on an IBM P4 using 24 processors. If simulations are conducted at all the Fr where EFD data are available for BH, the time saved using a single run is 68.6%.

Figure 9 shows the free surface wave field predicted at $Fr=0.43$ using a ramp time of 100 dimensionless seconds. Compared with the unsteady RANS computation for Athena bare hull at $Fr=0.43$ [25], the overall flow pattern is similar except for the breaking bow wave, due to the insufficient discretization on the coarse grid applied.

## 5.4 Validation of Full-Fr Propulsion Curve Simulation.

Figure 10 shows the whole powering curve for the AH using a ramp time of 100 dimensionless seconds. The independent variable is the propeller RPS, which is slowly increased as a function of time (Eq. (54)). All the dependent variables, including the resistance coefficient, $Fr$, and sinkage and trim are predicted. It is observed that overall both the computational towing tank approach and the steady-state computations agree well with the EFD data. Predicted $Fr$ are $2.6\%$ lower than the EFD data, which may be caused by the simplified body-force propeller model applied and the uncertainties of the EFD data. The predicted ship sinkage and trim agree well with EFD for a low-speed range ($Fr<0.45$) but are overpredicted for $0.45<Fr<0.83$, the magnitude of the overprediction increasing with increasing Fr. This is likely due to the complexity of the wave splashing and breaking, bubble entrainment, and free surface turbulence for high-speed ship flows, which are not modeled herein. Predicted resistance coefficient agrees very well with EFD, except for RPS $<$ 13.6 ($Fr<0.37$), where the difference between CFD and EFD is up to $13\%$. The nondimensional propeller thrust force is higher than the resistance coefficient within this range, which suggests that the very slow ship speed at the beginning of the whole powering curve simulation does not provide sufficient time for the wave pattern and boundary layer to be fully developed.
Three steady-state points were simulated for \( Fr = 0.432, 0.575, \) and 0.839 and compared with the EFD data. The thrust, torque, and advance coefficients of the propeller are computed based on the target \( Fr \). When the resistance force balances the total thrust force from the propeller minus the skin friction correction, the acceleration of the ship equals zero and the ship speed becomes a constant. This self-propelled speed thus \( Fr \) and the predicted sinkage and trim were then compared with the target values (EFD data). The predictions of the self-propelled \( Fr \) are within 2.1% of the EFD values. Sinkage and trim differences are less than 11% for all \( Fr \), with the exception of the very low sinkage observed at the highest \( Fr \), for which the relative error is misleading since the absolute value is properly predicted by CFD.

### 6 Conclusions and Future Work

The implementation of absolute inertial earth-fixed coordinates \( (X, Y, Z, t) \) in the RANS/DES solver CFDSHIP-IOWA allows de vel-
Table 7  Steady-state computations for the self-propelled fully appended Athena. %EFD for Fr and %EFD dynamic range for sinkage and trim.

<table>
<thead>
<tr>
<th>Fr</th>
<th>Sinkage (deg)</th>
<th>Trim (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>EFD</td>
<td>0.432</td>
<td>-0.0036</td>
</tr>
<tr>
<td>CFD</td>
<td>0.423</td>
<td>-0.0038</td>
</tr>
<tr>
<td>E</td>
<td>2.1%</td>
<td>6.2%</td>
</tr>
<tr>
<td>EFD</td>
<td>0.575</td>
<td>-0.00335</td>
</tr>
<tr>
<td>CFD</td>
<td>0.567</td>
<td>-0.00305</td>
</tr>
<tr>
<td>E</td>
<td>1.4%</td>
<td>9.4%</td>
</tr>
<tr>
<td>EFD</td>
<td>0.839</td>
<td>-8.0 \times 10^{-4}</td>
</tr>
<tr>
<td>CFD</td>
<td>0.830</td>
<td>-3.0 \times 10^{-4}</td>
</tr>
<tr>
<td>E</td>
<td>1.1%</td>
<td>15.6%</td>
</tr>
</tbody>
</table>

Sensitivity study of the ramp time conducted for a full-Fr resistance curve shows that $C_{TX}$ and ship motions predicted using the long ramp time are closer to the EFD data than the short one. This is consistent with the validation of the full-Fr resistance curve, which shows that the long ramp time reduces the average comparison errors $|E|$ by 34.4%, 4.9%, and 8.6% for $C_{TX}$ sinkage, and trim, respectively, compared to the use of the short ramp time. For both ramp times, validation is achieved at intervals of validation $(U_{95})$ 8.5% and 34.3% for $C_{TX}$ sinkage, and trim, respectively, but not achieved for trim except for Fr<0.4 at $U_{95}$ of 7.0%.

The overall intervals of $|E|$ for resistance coefficient, sinkage, and trim are consistent with previous studies [4,7,26]. For $C_{TX}$, overall results of the 2005 and 2000 workshop results indicate that the interval of $|E|$ is 4% [26]. Other typical values for $|E|$ is 8% for High Speed Sealift (HSSL)-Delft [4] and 4.3% for 5512 [7]. For sinkage, $|E|$ is 23% for HSSL-Delft and 7.4% for 5512. For trim, $|E|$ is 17% for HSSL-Delft and 10.4% for 5512. $U_{SN}$ for $C_{TX}$ is at the same interval as other studies while $U_{SN}$ for ship motions are rarely evaluated by previous studies.

Overall results prove that for medium and high Froude numbers the computational towing tank is an efficient and accurate tool to predict curves of resistance and propulsion for ship flows using a single run, including time savings on case setup and CPU hours compared to multiple runs and accuracy and flexibility on diagnosing flow physics. The procedure is not possible or highly difficult using a physical towing tank suggesting a potential of using the computational towing tank to aid the design process. However, a full curve simulation for low Fr can be very expensive in terms of computational time due to the long ramp time needed to accelerate the ship and resolve the unsteadiness of the flow.

Future work is to investigate the effects of different ramp time functions and initial conditions and improve the propeller model using a more complex propeller model (for instance, the Yamazaki model [27]) that accounts for the interaction between ship hull and propeller.

Acknowledgment

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