Verification and validation has been completed for the use of computational fluid dynamics as a practical means of simulating captive manoeuvring model tests. Verification includes spatial and temporal refinement studies. Direct validation includes the comparison of individual steady drift and planar motion mechanism simulations to physical model test data. Rotating arm simulations are validated indirectly on the basis of manoeuvring derivatives developed from the PMM tests. The merits of steady and unsteady simulations are discussed.

KEY WORDS: Manoeuvring, Manoeuvering, Maneuvering, Computational Fluid Dynamics, CFD, Planar Motion Mechanism, Rotating Arm, Verification, Validation.

INTRODUCTION

While a basic level of manoeuvring performance is essential for the safe navigation of all vessels, the manoeuvring performance of a warship is an integral aspect of the vessel capability. High manoeuvring performance can be leveraged to effect a tactical advantage in combat situations, can improve the safety of ship-to-ship and fleet operations, and can improve performance in peacetime roles requiring launch and recovery operations or performance in high winds or seas.

The ability to accurately describe the manoeuvring performance of an existing warship can help the performance limitations to be better understood by the ship’s captain, so that the vessel’s capability can be exploited more effectively. This quality of information is typically obtained through manoeuvring trials, including standard tests such as turning circles, zig-zag maneuvers, crash stop maneuvers, spiral maneuvers, and more specialized maneuvers.

An accurate prediction of manoeuvring performance is also important at the design stage, when it remains possible to make design adjustments which can remedy deficiencies in the manoeuvring performance or further enhance the performance. However, trials data are not available for the first-of-class vessel at the design stage, and it is necessary to use other means to predict the manoeuvring performance.

Manoeuvring performance predictions can be made using physical model tests to directly simulate the trials maneuvers with a free-running model. Alternatively, either rotating arm tests or planar motion mechanism (PMM) tests can be conducted at model scale with the aim of developing mathematical models of the manoeuvring performance. Manoeuvring model tests are expensive due to the large testing program and the specialized equipment required. Manoeuvring model tests are therefore typically undertaken only late in the design cycle, and possibly not at all for some types of naval vessels.

In the absence of model test data, the necessary data to populate a mathematical manoeuvring model can be approximated using techniques such as regression analysis of data obtained from other (similar) vessels, strip-wise integration of estimated coefficients, and semi-empirical methods to separately account for the effects of appendages. The reliability of these methods is not high, because they often do not provide sufficient account of the specific geometry of the vessel, and the influence it has on the flow patterns and forces.

Even manoeuvring model tests results are subject to greater uncertainty than classic resistance and propulsion model tests. In many manoeuvring model tests, the measured loads include the load required to accelerate the vessel, and this must be removed.
from the measured loads to obtain the pure hydrodynamic loads. Such indirect measurements incur additional error. Also, manoeuvring forces include strong viscous components, so using Froude scaling to full scale may introduce error.

The prominence of viscous forces has also made it challenging to use numerical techniques to estimate the manoeuvring performance of a vessel. This is in contrast to other aspects of hydrodynamic performance, many of which can be predicted reasonably well under the assumption of inviscid potential flow. This combination of challenges has helped manoeuvring prediction to earn a reputation as a “black art”, with final results having an intricate dependency on coefficients that are difficult to estimate.

Meanwhile, Computational Fluid Dynamics (CFD) has developed from a research tool for use in specialized problems to a routine design tool. Integral aspects of this shift have been the dramatic increase in computational resources, technical developments, and their incorporation into commercially available CFD software. Important developments have spanned the whole range of the CFD workflow, including the direct import of 3D CAD geometry, automated meshing tools including local mesh refinement and specialized meshing models to resolve the turbulent boundary layer, the introduction of the volume of fluid (VOF) approach to modelling free surface flows, the collection of a wide range of physics models within a single program, and integrated post-processing. This has led to the routine use of CFD for vessel resistance calculations with good accuracy, and several publications on the use of CFD for more challenging problems.

The field of CFD prediction of ship manoeuvring is relatively new, but has incited the recent development of specific recommended procedures by ITTC (2011). These provide good insight into many of the important aspects of preparing such simulations, but due to the wide variety of techniques available the wording is non-prescriptive. The SIMMAN series of workshops have been a helpful means of collaboration in this field, allowing the comparison of several model test and CFD results.

CFD manoeuvring prediction techniques have ranged from simple single-phase flow (Zou, Larsson, and Orych, 2010) to more sophisticated two-phase models using levelset (Sakamoto, Carrica, and Stern, 2012a) or VOF (Simonsen et al., 2012) methods. Simulations have been done for free manoeuvres (Dubbioso, Durante, and Broglia, 2013) as well as captive manoeuvres (Sakamoto, Carrica, and Stern, 2012b). Carrica et al. (2013) indicate that the computation times for simulating free manoeuvres are about 50 times that required for captive simulations.

The accuracy of captive simulations has typically been better for steady conditions than for unsteady ones. Relatively early steady drift simulations by Jacquin et al. (2006) achieved between about 3% and 20% accuracy, depending on the drift angle, Froude number, vessel, and force component. Simonsen et al. (2012) obtained an accuracy of about 5% for steady drift CFD simulations on a 7 million cell mesh, and found about 2% change when coarsening the mesh to 3 million cells. Sakomoto (2009) completed simulations for the same vessel as used in the present work, and found accuracies of about 4%, 10%, and 2% for the longitudinal force, lateral force, and yaw moment in a 10° steady drift simulation, respectively. In the same work, the accuracy in an unsteady yaw and drift simulation was about 12% for the transverse force and yaw moment, and about 24% for the longitudinal force.

Objective

The objective of the present work has been to validate CFD as a reliable and practical technique for determining hull manoeuvring force coefficients. This includes direct validation of steady drift simulations and PMM simulations against model test data presented by Simonsen (2004), and indirect validation of rotating arm simulations by comparison of mathematical manoeuvring models. The objective of ensuring practicality of the technique from a ship designer’s perspective has implied limiting the computer hardware and simulation run time to that typically available to a ship designer.

METHODOLOGY

Conventions

All results are reported in a ship-fixed coordinate system with origin on centerline amidships, as shown in Fig. 1.

![Ship-fixed coordinate system](image)

The centreline amidships components of the vessel velocity are positive forward (surge, \( u \)), to starboard (sway, \( v \)), and clockwise
The accelerations ($u, v, r$) and the hydrodynamic surge and sway forces and yaw moment ($X, Y, N$) use the same convention.

The motions and accelerations were nondimensionalized using the magnitude of the vessel velocity, $U = \sqrt{u^2 + v^2}$, and the vessel length between perpendiculars, $L$, as follows:

$$
\begin{align*}
&u' = \frac{u}{U} \quad v' = \frac{v}{U} \quad r' = \frac{rL}{U} \\
&\dot{u}' = \frac{\dot{u}L}{U^2} \quad \dot{v}' = \frac{\dot{v}L}{U^2} \quad \dot{r}' = \frac{\dot{r}L}{U} 
\end{align*}
$$

The hydrodynamic forces and moment were nondimensionalized as follows:

$$
\begin{align*}
&X' = \frac{X}{\frac{1}{2} \rho U^2 LT_0} \quad Y' = \frac{Y}{\frac{1}{2} \rho U^2 LT_0} \quad N' = \frac{N}{\frac{1}{2} \rho U^2 L^2 T_0} 
\end{align*}
$$

where $\rho$ is the water density and $T_0$ is the calm-water draft.

**Motion**

The various types of motions applied in the CFD simulations are illustrated in Fig. 2. The motions in the unsteady PMM simulations were set using the same motion equations as were achieved by the PMM mechanism used in the model tests (Simonsen, 2004). The carriage speed, $U_c$, was kept fixed based on the target Froude number of 0.28. The model heading, $\psi$, with respect to the direction of travel of the carriage included a steady drift component, $\beta$, and an oscillatory component, $\psi_o$, as shown in Fig. 1. Due to the construction of the mechanism, the heading was not exactly sinusoidal.

---

**Fig. 2. Vessel and mesh motions for simulation of planar-motion mechanism tests and rotating arm tests.**
ψ = β + ψ₁
= β – tan⁻¹(acos ωt)

(4)

where \(a = Y_{mm}/R\), \(Y_{mm}\) is a case-specific mechanism setting, \(R = 0.5\) m, \(ω = 2π/T\) is the motion frequency in rad/s, and \(T\) is the motion period. The transverse position of the model, \(x\), measured perpendicular to the direction of travel of the carriage, is sinusoidal:

\[ η_{PMM} = 2S_{mn}(1 - \sin ωt) \]

(5)

where \(S_{mn}\) is a case-specific mechanism setting. The shift in the mean value of \(η_{PMM}\) is applied to facilitate the initialization of the CFD simulations. The velocity components of the vessel in the ship coordinate system were obtained by differentiating the PMM motions in time, then transforming to ship coordinates:

\[ r = \frac{dφ}{dt}, \quad v_{PMM} = \frac{dη_{PMM}}{dt} \]

(6)

\[ v = v_{PMM} \cos ψ - U_C \sin ψ \]

(7)

\[ u = v_{PMM} \sin ψ + U_C \cos ψ \]

(8)

The time rates of change of these were then obtained by differentiating in time. These include the influence of changing \(ψ\) as in Simonson (2004), so do not include centripetal acceleration.

\[ \dot{r} = \frac{dv}{dt}, \quad \dot{v} = \frac{dv}{dt}, \quad \dot{u} = \frac{du}{dt} \]

(9)

For the steady drift simulations, the drift angle and carriage speed were specified, the oscillatory terms are zero, and the above reduces to \(r = 0\), \(v = U_C \sin β\), \(u = U_C \cos β\), and no accelerations.

For the rotating arm simulations, the motion was specified based on the vessel speed, \(U\), and rate of rotation, \(r\). In the global coordinate system the vessel followed a circular path of radius \(L/r\), such that \(v = U \sin β\), \(u = U \cos β\) and the rates of change of these are zero.

**Numerical Technique**

CFD simulations have been carried out using STAR-CCM+ by CD-adapco. Version 9.06.011 was used for the PMM and rotating arm simulations, while version 8.06.007 was used for the steady drift simulations. The technique numerically solves the Reynolds-Averaged Navier-Stokes equations. Turbulent effects were simulated using Menter’s (1994) shear-stress transport \(k - \omega\) turbulence model with a blended (all-\(y^+\)) near-wall approach. Both water and air were modeled using the volume of fluid approach. The simulations were configured such that the High-Resolution Interface Capturing (HRIC) method was used over most of the domain. In some local areas with small cells this transitioned to a second-order free surface capturing scheme to maintain numerical stability.

The computational meshes were unstructured, and predominantly composed of hexahedral cells, with localized refinements achieved by cell subdivision, and trimming of the cells to fit the boundaries. Prismatic layers conforming to the vessel surface were used to efficiently resolve the boundary layer, with a near-wall cell size set to achieve a \(y^+\) value of approximately 50. This is consistent with a log-law turbulent boundary approach. The mesh included refinement in the vertical direction to accurately capture the air/water interface, additional refinement of the free surface in way of the vessel wake, general refinement near the vessel to capture vortices, and localized refinements near the bilge keels, aft, and bow wave. The detailed shape of the refinement blocks were adjusted for different vessel motion types and amplitudes, so the flow features of interest remained within the refined mesh. This included, for example, consideration for the convection of the free surface wake across the mesh. Aggressive mesh coarsening towards the outer boundaries and a wave damping function were used to effectively eliminate the waves before they reached the boundaries, while avoiding wave reflections. Typical cell counts were 2.4 M, 4.2 M, and 5.7 M for the PMM, steady drift, and rotating arm simulations, respectively.

All of the simulations were solved using an implicit unsteady approach. In the unsteady simulations, the time step was typically chosen to be \(T/2880\), where \(T\) is the motion period. This achieved a Courant-Friedrichs-Lewy number (CFL) less than unity in most cells. Ten iterations were used per time step. To help reach a repeatable cyclic response quickly, a steady initialization phase preceded the unsteady part of each PMM simulation; the unsteady motion then initiated with \(t = T/4\) to avoid instantaneous changes in the sway rate. This is shown for the three unsteady simulation types in Fig. 2. The unsteady simulations were run until good cyclic repeatability was achieved; in most cases this was within 1.6 motion periods.

As compared to the PMM simulations, the steady drift and rotating arm simulations used fewer iterations per time step and larger time steps to efficiently achieve a steady solution. The vessel travelled at least thirteen ship lengths to achieve steady results with good convergence.
Sinkage and Trim
Where available, model test data were used to specify the
dynamic sinkage and trim for the CFD simulations. This included
steady drift and PMM simulations. For the PMM simulations,
time-averages of the model test sinkage and trim were used.
Publications by Simonsen (2004) and Agdrup (2004), both based
on the same set of model tests, were combined to determine the
correct sinkage and trim.

For the rotating arm simulations, the sinkage and trim were
determined via CFD. A manual iteration technique was used to
successively improve estimates of the sinkage and trim, such that
the net vertical force and trimming moment were reduced. Based
on calm-water hydrostatics, the running conditions were achieved
within tolerances of 0.7 mm for draft at the longitudinal centre of
flotation, and 0.002° trim.

Post-Processing
The primary results of the simulations are the manoeuvring
forces, X and Y, and yaw moment, N. These were integrated on
the hull surface including both pressure and shear stress
components.

For the unsteady simulations, only the results from the last motion
period in the simulation were used for further processing. A third-
order Fourier series was fit to each result. Using \( X' \) as an example,
\[
X'(t) \approx C_0 + \sum_{k=1}^{3} \left( C_{k,s} \cos k \omega t + C_{k,s} \sin k \omega t \right) \tag{10}
\]
Use of the Fourier coefficients facilitates the comparison of the
results time-history from a single simulation against the model
test data. Fourier coefficients would also be required if the
manoeuvring derivatives were to be computed by harmonic
analysis, so their use here is considered to maintain a focus on the
features that are important to a mathematical manoeuvring model.

Calculation of Manoeuvring Derivatives
The development of a mathematical manoeuvring model based
on captive manoeuvring tests or simulations is not standardized
by ITTC. Here, manoeuvring derivatives have been calculated by
various techniques for use in the traditional Taylor series
representation of manoeuvring force coefficients, as typically
used in mathematical manoeuvring models (Lewis, 1989). The
Taylor series representations of the force coefficients are as
follows, for the non-dimensional surge force, sway force, and
yaw moment, respectively:
\[
X'(\hat{u}', \hat{v}', \hat{r}') \approx X_0 + X_0 \hat{u}' + \frac{1}{2} X_{uv} v'^2 + \frac{1}{12} X_{ur} r'^2 + \frac{1}{6} X_{rr} r'^3 \tag{11}
\]
\[
Y'(\hat{v}', \hat{r}', \hat{v}', \hat{r}') \approx Y_0 \hat{v}' + \frac{1}{2} Y_{uv} v'^2 r' + \frac{1}{24} Y_{ur} r'^3 + \frac{1}{8} Y_{vv} v'^4 + \frac{1}{4} Y_{rv} v'^2 r'^2 \tag{12}
\]
\[
N'(\hat{v}', \hat{r}', \hat{v}', \hat{r}') \approx N_0 + \frac{1}{2} N_{uv} v'^2 r' + \frac{1}{24} N_{ur} r'^3 + \frac{1}{8} N_{vv} v'^4 + \frac{1}{4} N_{rv} v'^2 r'^2 \tag{13}
\]
where a shorthand has been used to represent the partial
derivatives of the manoeuvring coefficients (“manoeuvring
derivatives”). For example,
\[
X'_{vr} = \frac{\partial^2 X'}{\partial v' \partial r'} \tag{14}
\]
These series expansions include only the hydrodynamic effects,
and not the influence of the vessel inertia that is ultimately
included in a mathematical manoeuvring model. Also, they
include only the terms up to third order which meet the centreline
symmetry requirements.

Due to the definition of the time rates of change \( \hat{u}' \), \( \hat{v}' \), and \( \hat{r}' \), the
influence of centripetal accelerations are not expressly included
in the added mass terms. Because centripetal acceleration, \( U_r \), is
closely correlated with the yaw rate, \( r \), it is difficult to discern
which of the two is the cause for observed force fluctuations. The
confusion was avoided by removing the centripetal accelerations.
The associated errors are expected to be small because only a
relatively small mass of fluid actually moves at or close to the
same speed as the vessel.

A regression technique was used to determine the manoeuvring
derivatives in equations (11) through (12) based on a given
ensemble of simulations. This involved minimizing the weighted
sum of the square of the differences between the CFD results and the
Taylor series approximation. This was completed individually for
\( X', Y', \) and \( N' \). Using \( X' \) as an example,
\[
SSE_X = \sum_i W_i \left( X'(\hat{u}', \hat{v}', \hat{r}') - X'_{CFD,i} \right)^2 \tag{15}
\]
where \( i \) denotes the \( i \)th datum, \( X'(u'_i, v'_i, r'_i) \) is the Taylor series expansion evaluated in the corresponding condition, and \( X'_{\text{CFD}} \) is the corresponding CFD result. The weights, \( w_i \), were selected for each simulation as \( 1/n \), where \( n \) is the number of data used from the simulation. In the unsteady simulations, each time step (in the final motion period) was considered a datum; the converged result is the only datum provided by a steady simulation. A more easily interpreted measure of the quality of the fit is the weighted root-mean-square error, \( E_{\text{RMS}} \). For \( X' \), this is

\[
E_{\text{RMS},X} = \sqrt{\frac{\sum_i (w_i \text{SSE})_X}{\sum_i w_i}}
\]  

(15)

The minimization of \( \text{SSE}_X \) with respect to the manoeuvring derivatives was accomplished by solving

\[
\frac{\partial \text{SSE}_X}{\partial D_X} = 0
\]  

(16)

for \( D_X \), the vector of manoeuvring derivatives appearing in the Taylor series expansion of \( X' \). This reduces to a linear system having the same size as \( D_X \). A similar technique was applied to \( Y' \) and \( N' \).

In determining the manoeuvring derivatives, the source data was generally taken to be either (a) the unsteady PMM model tests, (b) the unsteady PMM simulations or (c) the combination of steady drift and steady rotating arm simulations.

SIMULATION CONDITIONS

Vessel Description

The subject vessel for this work is the DTMB 5415 destroyer model used in the physical model tests reported by Simonsen (2004). The model has a typical fine hull form for a fast displacement naval monohull, and includes a sonar bulb below baseline and bilge keels. No rudder, propellers, shafts, or other appendages were included in either the model tests or the CFD simulations. The model was built to a notional scale of 1:35.48. Particulars of the model are shown in Table 1, and a visualization of the hull is shown in Fig. 3. The hull form and bilge keel design were obtained in electronic format from FORC Technology (2014), from which 3D geometry was developed for input into the CFD software.

Table 1. DTMB 5415 destroyer model particulars

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars, ( L )</td>
<td>4.0023</td>
<td>m</td>
</tr>
<tr>
<td>Length on waterline</td>
<td>4.0083</td>
<td>m</td>
</tr>
<tr>
<td>Beam on waterline, ( B )</td>
<td>0.5382</td>
<td>m</td>
</tr>
<tr>
<td>Calm water draft, ( T_o )</td>
<td>0.1736</td>
<td>m</td>
</tr>
<tr>
<td>Calm water trim</td>
<td>0</td>
<td>°</td>
</tr>
<tr>
<td>Volumetric displacement</td>
<td>0.1897</td>
<td>m³</td>
</tr>
</tbody>
</table>

Fig. 3. DTMB 5415 destroyer hull geometry

Physical Constants

In the interest of accurately replicating the model tests, the physical constants reported by Simonsen (2004) were used in the CFD analysis. The only exception to this is gravitational acceleration, for which value of 9.806 m/s² was used in the present work; Simonsen (2004) did not measure the local gravitational acceleration, and assumed a value of 9.81 m/s² in the model test analysis.

Motion Conditions

All simulations were carried out at a Froude number of 0.28. The simulation matrix was developed with the aim of reproducing the steady drift and PMM physical model tests presented by Simonsen (2004). While this does not include rotating arm model tests, the simulation matrix includes rotating arm simulations intended to capture the same range of motion as experienced within the PMM test matrix.

The simulation matrix, including both steady and unsteady simulations, is included in Table 2. Steady simulations were carried out using each of the conditions marked with an “S” in the table, where the condition is described by the corresponding drift angle and non-dimensional yaw rate. The highest non-dimensional yaw rate of \( r' = 0.75 \) corresponds to a turning diameter of 2.76L. The steady cases with drift angles of 0°, 9°, 10°, and 11° were repeated using draft and trim from the model test results, and draft and trim as determined by iterative CFD analyses.

Unsteady simulations were carried out using each of the combinations marked with “U” in the table. In cases of unsteady pure sway (i.e. no yaw), the sine of the drift angle shown gives the nominal amplitude of the non-dimensional sway rate. For coupled unsteady yaw and drift cases, the drift angle shown in the table is steady throughout the simulation. In all unsteady cases, the yaw rate shown in the table is taken as the nominal amplitude. The actual values of \( S_{nm} \) and \( Y_{nm} \) used to achieve the nominal unsteady motions shown in Table 2 are as presented by Simonsen (2004). The PMM motion period was based on seven cycles per minute in most cases, and nine cycles per minute for the pure yaw cases with non-dimensional yaw rate amplitudes of 0.6 and 0.75.
Validation has focused on the cases where “S” or “U” is shown in bold italics in Table 2; in these cases the corresponding steady or unsteady model tests included 12 repeat runs. Verification has focused on the cases where “S” or “U” is underlined in the table.

**CFD RESULTS**

Many results were derived from the various CFD simulations. A few samples of these are included here, mainly for verification and validation purposes. The ensembles of steady manoeuvring force and moment results are shown later in the context of determining the manoeuvring derivatives.

**Flow Features**

**Steady Simulations.** The steady drift simulations predicted vortex generation at the bow by the sonar bulb, at each of the bilge keels, and at the skeg. The bulb vortex was predicted to be both large and strong, and at some modest drift angles it passed under the starboard bilge keel. In these cases, the top part of the bow vortex caused the transverse flow component in way of the bilge keel tip to be from starboard to port, rather than the opposite trend generally observed elsewhere (see Fig. 7). The effects of this reversal would not be predicted by a simpler methodology that does not consider interaction effects. An example of this is shown in the underwater view in Fig. 4. It includes vortex identification using iso-surfaces of the Q-criterion at $Q = 30$, determined following Sakomoto (2009). The colours of the vortex are based on dimensionless helicity, and indicates the direction of rotation: red vortices are rotating clockwise when viewed from downstream. While this also identifies vortex structures in the breaking wave and against the hull surface, they are not as important as the vortices generated by the bulb, bilge keels, and skeg.

Larger drift angles were found to cause breaking waves to form, the most substantial of which is a plunging breaking bow wave on the port (windward) side. It was found to generate air cavities that caused air to be swept down and aft. This is shown via the volume fraction in Fig. 5, with water in red, transitioning to air in blue. Local refinement of the mesh in the vicinity of the bow wave allowed the visualization showing the stern on the first page of this paper to be developed, but accurate resolution of these small flow phenomena was not considered important to the overall accuracy of the larger scale hull manoeuvring forces.

**Fig. 4. Steady vortex structures at 10° drift (top), steady $r' = 0.3$ (middle), and 10° drift with $r' = -0.3$ (bottom), view from below**

<table>
<thead>
<tr>
<th>Drift angle (amplitude) [°]</th>
<th>Non-dimensional yaw rate (amplitude)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>-0.3</td>
</tr>
<tr>
<td>0</td>
<td>S</td>
</tr>
<tr>
<td>2</td>
<td>SU</td>
</tr>
<tr>
<td>4</td>
<td>U</td>
</tr>
<tr>
<td>6</td>
<td>S</td>
</tr>
<tr>
<td>9</td>
<td>S</td>
</tr>
<tr>
<td>10</td>
<td>S</td>
</tr>
<tr>
<td>11</td>
<td>S</td>
</tr>
<tr>
<td>12</td>
<td>S</td>
</tr>
<tr>
<td>16</td>
<td>S</td>
</tr>
<tr>
<td>20</td>
<td>S</td>
</tr>
</tbody>
</table>

**Table 2. Simulation matrix**

Oldfield Prediction of Warship Manoeuvring Coefficients using CFD 7
The rotating arm simulations also predicted several vortices, breaking waves, and bubble sweep-down. The bow vortex was again found to be larger than the others, but in rotating arm cases with no drift it rotated in the direction opposite to the bilge keel and skeg vortices. This is consistent with the opposite direction of the transverse flow component in the forward and aft parts of the ship. The bulb vortex was not generated in the drift and yaw cases where $\beta$ and $r'$ were both positive, as the local angle of attack at the bow was near zero. In the combined case with negative $r'$, the bulb and leeward bilge keel vortices switched direction, and the bilge keel vortex detached and was swept under the bulb vortex. This is shown in the bottom of Fig. 4.

Many of the flow phenomena predicted in the steady simulations with large drift angles or yaw rates were inherently unsteady. As such, a true steady solution to the RANS equations does not exist. The computational demands of achieving a time-accurate solution to resolve these flow features is much greater than that required to obtain reasonable convergence of the overall and nominally steady flow, and has not been attempted here.

Unsteady Simulations and Comparison. The results of the unsteady simulations are in many ways similar to those of the steady simulations, except that they show less spatial detail due to the coarser mesh used, and they include unsteady effects.

Acceleration effects are a primary effect of unsteady flow, and have traditionally been approximated as linear, inviscid, added-mass effects. The unsteady CFD simulations, however, show evidence of more complicated phenomena. Examples of this include breaking waves and vortices present at the instant when the sway and yaw velocities are zero and accelerations are highest.

At the instant in an unsteady simulation when the accelerations are zero, the vessel motions are nominally the same as in a steady simulation. In such cases, the difference between the unsteady flow field and that obtained in a corresponding steady simulation is largely due to the influence of the prior flow on the instantaneous unsteady flow (differences in numerical errors also play a role). This influence of the flow history is case-specific. An example that illustrates it well is the comparison between the zero-drift rotating arm simulation at $r' = 0.75$ to the pure yaw PMM simulation at the same nominal amplitude, and a phase angle of $\omega t = \pi/2$ (or 90°). These two cases are contrasted in the underwater view in Fig. 6. The free surface elevation is shown in shades ranging from dark blue (low) to white (high). The pressure disturbance caused by the vessel’s motion was determined by subtracting the zero-speed hydrostatic component from the pressure, and is plotted on the hull surface.

While the general trends in the upper and lower parts of Fig. 6 are similar, some details are different. The steady case has well-developed vortices due to the bilge keels, skeg, and bulb, while the vortices in the unsteady case were still growing in strength and longitudinal extent, and their transverse position was changing in time. This caused the unsteady result to have a smaller low pressure region near the port bilge keel and a smaller low-pressure streak trailing aft of the bulb on the port side of the bow. The longer bulb and bilge keel vortices in the steady case also resulted in a low-pressure streak inboard of the port bilge keel, between its aft end and the skeg; it is not present in the unsteady result. The unsteady result has a smaller wave trough to port of the bulb, a steeper rise to a narrower crest immediately aft of it, and more interference around amidships from the closer wake on the port side. These relate to the substantially different distribution of the pressure disturbance on the port side, including much lower pressures near the port shoulder. The differences on the starboard side are more subtle. Although the spray and details of the wave breaking phenomena are also different, more refined simulations would be required to resolve them accurately.
fixed to prevent changes in the modelling errors associated with the turbulence wall treatment. The time step was adjusted such that the CFL number remained the same in all three meshes.

The fine mesh solution resolved vortex strengths well, predicted secondary vortices, and showed some of the smaller features within the breaking bow wave. The coarse mesh tended to be diffusive and smear out larger scale features, and the medium mesh captured large-scale features but smeared the details. As an example of this, the component of the vorticity vector in the direction parallel to the carriage motion is shown on a midship cross-section in Fig. 7, for all three meshes.

The mesh dependence of the forces was about the same order of magnitude as the force fluctuations near the end of a simulation. The converged forces and this temporal convergence error have therefore been estimated based on weighted averaging of the force time-history, and are shown in Table 3. Rigorous consideration for the unknown error that may be associated with imperfect temporal refinement was not attempted.

![Fig. 7. Vorticity predicted with coarse (top), medium (middle), and fine (bottom) meshes](image)

The bottom row of Table 3 shows fine-mesh spatial discretization relative error, or the grid convergence index, $GC_{\text{fine}}^{21}$, calculated following the method of Celik et al. (2008). The spatial error calculation has been repeated using every combination of upper or lower bound estimates of the converged forces on each mesh. From this, the minimum and maximum estimates of $GC_{\text{fine}}^{21}$ were identified. As shown in the bottom row of the table, the maximum estimate of the spatial discretization error is 0.14% for $X'$, 3.0% for $Y'$, and 1.6% for $N'$. This can be considered to represent the combined uncertainty due to both imperfect convergence in time and spatial discretization.

<table>
<thead>
<tr>
<th>$X'$</th>
<th>$Y'$</th>
<th>$N'$</th>
</tr>
</thead>
<tbody>
<tr>
<td>-0.020411 ±0.000034</td>
<td>0.059620 ±0.00011</td>
<td>0.029370 ±0.000026</td>
</tr>
<tr>
<td>-0.019644 ±0.000026</td>
<td>0.059937 ±0.000083</td>
<td>0.029790 ±0.000026</td>
</tr>
<tr>
<td>-0.019596 ±0.000028</td>
<td>0.059182 ±0.000074</td>
<td>0.029736 ±0.000029</td>
</tr>
<tr>
<td>$GC_{\text{fine}}^{21}$ (min., expected, max.)</td>
<td>0.0004% 0.027% 0.14%</td>
<td>0.26% 1.0% 3.0%</td>
</tr>
</tbody>
</table>

Unsteady Mesh Refinement. A mesh refinement study was carried out for the unsteady yaw and drift PMM case with $r = 0.3$ and a drift angle of $10^\circ$. This followed a procedure similar to that used for the steady drift case, but the cell counts were smaller to allow practical calculation times. The cell counts were 0.98M, 2.4M, and 6.1M. The time step was adjusted to maintain the same CFL on all three meshes, based on a time step of $T/5760$ on the medium mesh.

The general mesh dependencies seen in the steady mesh refinement study were also present in the unsteady cases, including resolution of vortex strengths and lengths, breaking waves, and bubble generation. As the meshes used were not as fine as those used for the steady mesh refinement study, the flow features were not as well resolved in the unsteady case. The unsteady simulations included mesh-dependent evidence of bubbles generated near the bow, swept down and aft, trapped inboard of the bilge keels, then released at the aft end of the bilge keel later in the simulation.

As in the steady case, the method of Celik et al. (2008) has been used to calculate the fine-mesh spatial discretization relative error, $GC_{\text{fine}}^{21}$, and the apparent order. This has been done for each of the Fourier coefficients of the non-dimensional forces individually, as shown in Table 4. For consistency with the comparison with model tests, the relative error has been calculated by dividing the absolute error ($GC_{\text{fine}}^{21} \times$ fine mesh result) by the maximum absolute value reported in the mean third-order Fourier fit to the experimental data. The relative error therefore shows where the inaccuracy in the coefficients is...
predict the spatial discretization error in divergent, so more mesh refinement is required to accurately capture all of the relevant flow features, so the results from it are not as useful in determining how accurately these features have been captured on the finer meshes. The relative error computed for the coefficients noted as divergent is therefore an indication of oscillatory convergence or an indication that the mesh refinement is not yet in the asymptotic range. For example, the coarse mesh may be too coarse to capture all of the relevant flow features, so the results from it are not as useful in determining how accurately these features have been captured on the finer meshes. The relative error computed for the coefficients noted as divergent is therefore of questionable accuracy. The apparent order also gives an indication of the accuracy of the error calculation: $GC_{time}^{k_{1}}$ is predicted more accurately when the apparent order is closer to the expected value of two.

The largest Fourier coefficients of $X'$ have been identified as divergent, so more mesh refinement is required to accurately predict the spatial discretization error in $X'$. The results for $Y'$ and $N'$ are more typically convergent, but some relative errors on the order of 10% indicate that the use of coarser meshes had an important influence on accuracy.

### Unsteady Time Refinement

A time refinement study was carried out using the same case as in the mesh refinement study: the unsteady yaw and drift PMM case with $r^i = 0.3$ and a drift angle of $10^\circ$. This included time steps of $T/384$, $T/2880$, and $T/5670$.

As compared to the first time step, the others represent refinements of 7.5 and 15 times, respectively. The coarsest time step was selected considering time-accuracy of flow phenomena having a temporal scale on the order of the motion period, and is consistent with that used by Sakamoto (2009). The finer time steps were selected on the basis of achieving a CFL number on the order of one or one-half, respectively, on the free surface near the vessel. With a time step of $T/2880$, the influence of switching between a first- and second-order accurate time stepping scheme was also investigated. It would normally be expected that the best time accuracy would be achieved with the finest time step and the highest order of accuracy, although guidance from CD-adapco indicates that the 1st order method is appropriate for many unsteady problems.

The choice of temporal method had a strong influence on flow aeration against the hull. This is depicted in Fig. 8, which shows, from top to bottom, time steps and orders of accuracy of $T/384$ 2nd order, $T/2880$ 1st order, $T/2880$ 2nd order, and $T/5670$ 2nd order. This is shown at a phase of 270°, when the yaw rate is large and opposing the superimposed drift angle at the bow. It shows that the longest time step resulted in dramatically increased flow aeration, and the two finer time steps resulted in similar amounts of flow aeration although the details of the aeration pattern are different. The use of the first order temporal method did not dramatically change the amount of flow aeration, but resulted in smooth streaks rather than defined bubbles.
Validation

Steady Drift. Validation of the steady drift force coefficients has been done through comparison to the physical model test results presented by Simonsen (2004) and Agdru (2004). Table 6 shows the relative difference with respect to the model tests, where positive values indicate the CFD result has a smaller magnitude than the physical model test result. The model test and CFD steady drift results are also included in Figs. 9-11 (see markers denoted by $r' = 0$). As in the verification above, this is based on weighted averages of the force time-histories from the steady simulations.

Table 6. Steady drift relative difference from model tests

<table>
<thead>
<tr>
<th>Drift angle</th>
<th>$X'$</th>
<th>$Y'$</th>
<th>$N'$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$0^\circ$</td>
<td>10.4%</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>$2^\circ$</td>
<td>5.8%</td>
<td>3.2%</td>
<td>5.5%</td>
</tr>
<tr>
<td>$6^\circ$</td>
<td>3.7%</td>
<td>0.4%</td>
<td>1.1%</td>
</tr>
<tr>
<td>$9^\circ$</td>
<td>1.2%</td>
<td>1.2%</td>
<td>0.5%</td>
</tr>
<tr>
<td>$10^\circ$</td>
<td>0.7%</td>
<td>4.1%</td>
<td>2.8%</td>
</tr>
<tr>
<td>$11^\circ$</td>
<td>6.4%</td>
<td>4.4%</td>
<td>2.6%</td>
</tr>
<tr>
<td>$12^\circ$</td>
<td>3.9%</td>
<td>3.3%</td>
<td>1.0%</td>
</tr>
<tr>
<td>$16^\circ$</td>
<td>7.3%</td>
<td>4.2%</td>
<td>6.7%</td>
</tr>
<tr>
<td>$20^\circ$</td>
<td>2.5%</td>
<td>3.7%</td>
<td>1.8%</td>
</tr>
</tbody>
</table>

The CFD results for $X'$ appear to under-predict the model test results at low drift angles. It is possible that this is partly influenced by the accuracy of the model test results. Similar model tests of a larger model of the same vessel presented by Benedetti et al. (2006) compare better with the CFD result for $X'$, with a relative difference of 0.7% at $0^\circ$ drift, gradually increasing to 1.9% at $6^\circ$ drift. At $16^\circ$ drift, the model tests showed poor symmetry for port and starboard drift angles.

Table 5. Unsteady yaw and drift temporal discretization error

<table>
<thead>
<tr>
<th>Force</th>
<th>Fourier coefficient $\rightarrow$</th>
<th>$C_0$</th>
<th>$C_{1,c}$</th>
<th>$C_{1,s}$</th>
<th>$C_{2,c}$</th>
<th>$C_{2,s}$</th>
<th>$C_{3,c}$</th>
<th>$C_{3,s}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$X'$</td>
<td>T/384, 2nd order</td>
<td>-2.093e-2</td>
<td>0.800e-3</td>
<td>-6.239e-4</td>
<td>6.474e-4</td>
<td>-3.265e-4</td>
<td>-2.405e-4</td>
<td>10.04e-5</td>
</tr>
<tr>
<td></td>
<td>T/2880, 1st order</td>
<td>-2.118e-2</td>
<td>0.933e-3</td>
<td>-1.505e-4</td>
<td>6.776e-4</td>
<td>-2.251e-4</td>
<td>-4.008e-4</td>
<td>8.328e-5</td>
</tr>
<tr>
<td></td>
<td>T/2880, 2nd order</td>
<td>-2.110e-2</td>
<td>1.061e-3</td>
<td>-3.346e-4</td>
<td>8.030e-4</td>
<td>-3.039e-4</td>
<td>-2.597e-4</td>
<td>7.071e-5</td>
</tr>
<tr>
<td></td>
<td>T/5670, 2nd order</td>
<td>-2.112e-2</td>
<td>0.963e-3</td>
<td>-2.179e-4</td>
<td>6.395e-4</td>
<td>-1.800e-4</td>
<td>-3.827e-4</td>
<td>5.924e-5</td>
</tr>
<tr>
<td></td>
<td>Relative error</td>
<td>0.09%</td>
<td>0.42%</td>
<td>0.50%</td>
<td>0.70%</td>
<td>0.53%</td>
<td>0.52%</td>
<td>0.05%</td>
</tr>
<tr>
<td></td>
<td>T/2880, 1st order</td>
<td>6.427e-2</td>
<td>-6.749e-3</td>
<td>-2.613e-2</td>
<td>-6.693e-3</td>
<td>-1.108e-4</td>
<td>3.995e-4</td>
<td>7.823e-4</td>
</tr>
<tr>
<td></td>
<td>T/2880, 2nd order</td>
<td>6.466e-2</td>
<td>-6.376e-3</td>
<td>-2.619e-2</td>
<td>-6.643e-3</td>
<td>-3.694e-4</td>
<td>3.126e-4</td>
<td>5.455e-4</td>
</tr>
<tr>
<td></td>
<td>T/5670, 2nd order</td>
<td>6.480e-2</td>
<td>-6.594e-3</td>
<td>-2.604e-2</td>
<td>-6.795e-3</td>
<td>-1.504e-4</td>
<td>2.725e-4</td>
<td>8.407e-4</td>
</tr>
<tr>
<td></td>
<td>Relative error</td>
<td>0.13%</td>
<td>0.20%</td>
<td>0.14%</td>
<td>0.14%</td>
<td>0.21%</td>
<td>0.04%</td>
<td>0.28%</td>
</tr>
<tr>
<td>$N'$</td>
<td>T/384, 2nd order</td>
<td>3.121e-2</td>
<td>-4.993e-3</td>
<td>-1.849e-2</td>
<td>-1.029e-3</td>
<td>2.988e-4</td>
<td>3.529e-4</td>
<td>3.222e-4</td>
</tr>
<tr>
<td></td>
<td>T/2880, 1st order</td>
<td>3.084e-2</td>
<td>-4.689e-3</td>
<td>-1.887e-2</td>
<td>-1.386e-2</td>
<td>0.271e-4</td>
<td>0.546e-4</td>
<td>1.586e-4</td>
</tr>
<tr>
<td></td>
<td>T/2880, 2nd order</td>
<td>3.088e-2</td>
<td>-4.630e-3</td>
<td>-1.885e-2</td>
<td>-1.244e-2</td>
<td>1.568e-4</td>
<td>1.560e-4</td>
<td>2.602e-4</td>
</tr>
<tr>
<td></td>
<td>T/5670, 2nd order</td>
<td>3.086e-2</td>
<td>-4.627e-3</td>
<td>-1.898e-2</td>
<td>-1.368e-2</td>
<td>0.199e-4</td>
<td>0.294e-4</td>
<td>1.968e-4</td>
</tr>
<tr>
<td></td>
<td>Relative error</td>
<td>0.04%</td>
<td>0.005%</td>
<td>0.23%</td>
<td>0.22%</td>
<td>0.25%</td>
<td>0.23%</td>
<td>0.11%</td>
</tr>
</tbody>
</table>
A more rigorous comparison has been made at 10° drift, where 12 repeat model tests were carried out as well as the CFD mesh refinement study. Table 7 shows the difference between the two alongside the approximate uncertainty in each range of the model test results. This shows that for all force components, the difference between the CFD and model test results is smaller than the sum of the two errors ($\frac{GCI_{line}}{2}$ and model test range/2).

Table 7. 10° Steady drift validation

<table>
<thead>
<tr>
<th></th>
<th>$X'$</th>
<th>$Y'$</th>
<th>$N'$</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFD</td>
<td>-0.0196</td>
<td>0.0592</td>
<td>0.0297</td>
</tr>
<tr>
<td>Model minimum</td>
<td>-0.0200</td>
<td>0.0600</td>
<td>0.0299</td>
</tr>
<tr>
<td>Model mean</td>
<td>-0.0195</td>
<td>0.0618</td>
<td>0.0306</td>
</tr>
<tr>
<td>Model maximum</td>
<td>-0.0175</td>
<td>0.0640</td>
<td>0.0318</td>
</tr>
<tr>
<td>$GCI_{line}$ (maximum)</td>
<td>0.14%</td>
<td>3.0%</td>
<td>1.6%</td>
</tr>
<tr>
<td>Model test relative range/2</td>
<td>6.6%</td>
<td>3.2%</td>
<td>3.2%</td>
</tr>
<tr>
<td>Relative difference</td>
<td>-0.7%</td>
<td>4.1%</td>
<td>2.8%</td>
</tr>
</tbody>
</table>

Fig. 9. Variation of $X'$ with drift angle and yaw rate

Fig. 10. Variation of $Y'$ with drift angle and yaw rate

Fig. 11. Variation of $N'$ with drift angle and yaw rate
Unsteady Simulations. Validation of the unsteady simulations has been done through comparison to the physical model test results presented by Simonsen (2004) and Agdrupt (2004). Several of the model test force time-histories included considerable high-frequency oscillations. This validation has therefore focused on the most reliable source data: the three unsteady conditions in which 12 repeat model tests were completed. As shown in Tables 8-10, these include pure sway, pure yaw, and combined yaw and drift cases, using motion amplitudes of 10° drift and \( r' = 0.3 \).

Table 8. Unsteady validation, \( X' \)

<table>
<thead>
<tr>
<th>Motion</th>
<th>Fourier Coef.</th>
<th>( C_0 )</th>
<th>( C_{1,s} )</th>
<th>( C_{1,e} )</th>
<th>( C_{2,s} )</th>
<th>( C_{2,e} )</th>
<th>( C_{3,s} )</th>
<th>( C_{3,e} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pure sway</td>
<td>Model min.</td>
<td>(-2.03e-2)</td>
<td>(3.01e-4)</td>
<td>(-2.14e-4)</td>
<td>(-8.14e-4)</td>
<td>(6.15e-4)</td>
<td>(-13.9e-5)</td>
<td>(-7.40e-5)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Model mean</td>
<td>(-2.01e-2)</td>
<td>(3.69e-4)</td>
<td>(-1.45e-4)</td>
<td>(-6.78e-4)</td>
<td>(7.42e-4)</td>
<td>(-3.98e-5)</td>
<td>(0.013e-5)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Model max.</td>
<td>(-1.99e-2)</td>
<td>(4.46e-4)</td>
<td>(-0.254e-4)</td>
<td>(-5.09e-4)</td>
<td>(8.54e-4)</td>
<td>(7.62e-5)</td>
<td>(5.06e-5)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Model range/2</td>
<td>0.84%</td>
<td>0.34%</td>
<td>0.44%</td>
<td>0.71%</td>
<td>0.56%</td>
<td>0.50%</td>
<td>0.29%</td>
</tr>
<tr>
<td>Pure sway</td>
<td>CFD</td>
<td>(-1.88e-2)</td>
<td>(-0.346e-4)</td>
<td>(0.167e-4)</td>
<td>(-4.76e-4)</td>
<td>(-2.37e-4)</td>
<td>(-1.14e-5)</td>
<td>(-1.21e-5)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Difference</td>
<td>(6.03%)</td>
<td>(-1.88%)</td>
<td>0.75%</td>
<td>(0.94%)</td>
<td>(-4.55%)</td>
<td>(0.13%)</td>
<td>(-0.06%)</td>
</tr>
</tbody>
</table>

Table 9. Unsteady validation, \( Y' \)

<table>
<thead>
<tr>
<th>Motion</th>
<th>Fourier Coef.</th>
<th>( C_0 )</th>
<th>( C_{1,c} )</th>
<th>( C_{1,s} )</th>
<th>( C_{2,c} )</th>
<th>( C_{2,s} )</th>
<th>( C_{3,c} )</th>
<th>( C_{3,s} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pure sway</td>
<td>Model min.</td>
<td>(-0.533e-4)</td>
<td>(5.40e-2)</td>
<td>(-3.50e-2)</td>
<td>(-3.69e-5)</td>
<td>(-28.8e-5)</td>
<td>(2.40e-3)</td>
<td>(-5.54e-4)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Model mean</td>
<td>(5.59e-4)</td>
<td>(5.47e-4)</td>
<td>(-3.38e-2)</td>
<td>(12.4e-5)</td>
<td>(-12.4e-5)</td>
<td>(2.65e-3)</td>
<td>(-3.51e-4)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Model max.</td>
<td>(10.4e-4)</td>
<td>(5.51e-2)</td>
<td>(-3.26e-2)</td>
<td>(36.6e-5)</td>
<td>(5.72e-5)</td>
<td>(2.78e-3)</td>
<td>(-2.65e-4)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Model range/2</td>
<td>0.83%</td>
<td>0.85%</td>
<td>1.81%</td>
<td>0.31%</td>
<td>0.26%</td>
<td>0.28%</td>
<td>0.22%</td>
</tr>
<tr>
<td>Pure sway</td>
<td>CFD</td>
<td>(0.055e-4)</td>
<td>(5.52e-2)</td>
<td>(-3.40e-2)</td>
<td>(-5.71e-5)</td>
<td>(-1.70e-5)</td>
<td>(2.32e-3)</td>
<td>(38.2e-4)</td>
</tr>
<tr>
<td>Pure sway</td>
<td>Difference</td>
<td>(-0.84%)</td>
<td>(0.84%)</td>
<td>(-0.31%)</td>
<td>(-0.28%)</td>
<td>(0.16%)</td>
<td>(-0.51%)</td>
<td>(0.59%)</td>
</tr>
</tbody>
</table>

Table 9. Unsteady validation, \( Y' \)

<table>
<thead>
<tr>
<th>Motion</th>
<th>Fourier Coef.</th>
<th>( C_0 )</th>
<th>( C_{1,c} )</th>
<th>( C_{1,s} )</th>
<th>( C_{2,c} )</th>
<th>( C_{2,s} )</th>
<th>( C_{3,c} )</th>
<th>( C_{3,s} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pure yaw</td>
<td>Model min.</td>
<td>(-3.80e-4)</td>
<td>(-7.23e-3)</td>
<td>(-1.80e-2)</td>
<td>(-25.1e-5)</td>
<td>(-4.03e-5)</td>
<td>(3.17e-4)</td>
<td>(2.07e-4)</td>
</tr>
<tr>
<td>Pure yaw</td>
<td>Model mean</td>
<td>(2.02e-5)</td>
<td>(-4.73e-3)</td>
<td>(-1.77e-2)</td>
<td>(-9.19e-5)</td>
<td>(15.6e-5)</td>
<td>(4.29e-4)</td>
<td>(2.65e-4)</td>
</tr>
<tr>
<td>Pure yaw</td>
<td>Model max.</td>
<td>(3.68e-4)</td>
<td>(-4.38e-3)</td>
<td>(-1.73e-2)</td>
<td>(6.33e-5)</td>
<td>(39.2e-5)</td>
<td>(5.30e-4)</td>
<td>(3.03e-4)</td>
</tr>
<tr>
<td>Pure yaw</td>
<td>Model range/2</td>
<td>1.97%</td>
<td>1.96%</td>
<td>2.05%</td>
<td>0.83%</td>
<td>1.14%</td>
<td>0.56%</td>
<td>0.25%</td>
</tr>
<tr>
<td>Pure yaw</td>
<td>CFD</td>
<td>(-1.14e-4)</td>
<td>(-3.17e-3)</td>
<td>(-1.56e-2)</td>
<td>(6.50e-5)</td>
<td>(-0.868e-5)</td>
<td>(-1.82e-4)</td>
<td>(9.82e-4)</td>
</tr>
<tr>
<td>Pure yaw</td>
<td>Difference</td>
<td>(-0.71%)</td>
<td>(8.23%)</td>
<td>(10.82%)</td>
<td>(0.83%)</td>
<td>(-0.87%)</td>
<td>(-3.22%)</td>
<td>(3.78%)</td>
</tr>
</tbody>
</table>

Table 9. Unsteady validation, \( Y' \)

<table>
<thead>
<tr>
<th>Motion</th>
<th>Fourier Coef.</th>
<th>( C_0 )</th>
<th>( C_{1,c} )</th>
<th>( C_{1,s} )</th>
<th>( C_{2,c} )</th>
<th>( C_{2,s} )</th>
<th>( C_{3,c} )</th>
<th>( C_{3,s} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yaw and drift</td>
<td>Model min.</td>
<td>(6.66e-2)</td>
<td>(-9.45e-3)</td>
<td>(-3.06e-2)</td>
<td>(-7.46e-3)</td>
<td>(-1.60e-3)</td>
<td>(1.62e-3)</td>
<td>(-1.63e-4)</td>
</tr>
<tr>
<td>Yaw and drift</td>
<td>Model mean</td>
<td>(6.83e-2)</td>
<td>(-8.52e-3)</td>
<td>(-3.01e-2)</td>
<td>(-7.04e-3)</td>
<td>(-1.13e-3)</td>
<td>(1.89e-3)</td>
<td>(0.986e-4)</td>
</tr>
<tr>
<td>Yaw and drift</td>
<td>Model max.</td>
<td>(6.89e-2)</td>
<td>(-8.16e-3)</td>
<td>(-2.87e-2)</td>
<td>(-6.65e-3)</td>
<td>(-67.1e-3)</td>
<td>(2.10e-3)</td>
<td>(2.80e-4)</td>
</tr>
<tr>
<td>Yaw and drift</td>
<td>Model range/2</td>
<td>1.08%</td>
<td>0.61%</td>
<td>0.88%</td>
<td>0.38%</td>
<td>0.43%</td>
<td>0.43%</td>
<td>0.22%</td>
</tr>
<tr>
<td>Yaw and drift</td>
<td>CFD</td>
<td>(6.44e-2)</td>
<td>(-6.13e-3)</td>
<td>(-2.67e-2)</td>
<td>(-6.74e-3)</td>
<td>(-93.4e-3)</td>
<td>(-0.077e-3)</td>
<td>(1.66e-4)</td>
</tr>
<tr>
<td>Yaw and drift</td>
<td>Difference</td>
<td>(-3.65%)</td>
<td>(2.24%)</td>
<td>(3.24%)</td>
<td>(0.28%)</td>
<td>(0.18%)</td>
<td>(-1.84%)</td>
<td>(0.06%)</td>
</tr>
</tbody>
</table>
Due to symmetry considerations, one would expect that the first-
little importance.

model test results than the other Fourier coefficients of

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and the source data generally pertain to details such as higher-order oscillations, amplitude-dependent phase shifts, or nonlinear effects at large vessel motions.

There was more difficulty in fitting the Taylor series to the \( X' \) results. The achieved \( \varepsilon_{\text{RMS}} \) is relatively large: typically about one fifth of typical values for \( X' \). This can be attributed to some combination of experimental error, uncertainty in the CFD technique, and more complicated fluctuations than can be represented by the Taylor series in equation (11). Investigations have suggested that the fit can be improved by including terms such as \( X'_{\text{pp}} v^2 r^2/2 \), \( X'_{\text{pp}} v^2 r^2/2 \), and \( X'_{\text{pppr}} v^2 r^2/4 \).

Comparison between the manoeuvring derivatives developed from different source data is a high-level means of validating the consistency of the different approaches. Many of the more important manoeuvring derivatives show little dependence on which data were used in their determination. Larger variations are typically seen in the higher order derivatives, which are traditionally difficult to predict and to which ship manoeuvres tend to be less sensitive.

A particularly useful application of the manoeuvring derivatives is for indirect validation of the rotating arm CFD simulations. The model tests by Simonsen (2004) did not include rotating arm tests, so direct validation was not carried out. Instead, the manoeuvring derivatives developed from the PMM model tests were used to predict the forces that would be developed in the rotating arm simulation conditions. The results of this indirect validation are shown in Figs. 9-11, where the fit to the unsteady PMM model tests are represented by smooth curves, and the steady CFD results are shown with markers. For reference, the steady drift model test results are also shown with dots.

The results for \( X' \) in Fig. 9 show some disagreement between the Taylor series fit to the PMM model tests and all of the steady results, including both CFD and model tests. It can be seen in the upper plot that the agreement between the steady drift model tests and the steady drift CFD simulations is substantially better than the comparison of either to the Taylor series. This suggests some underlying theoretical issues or model test accuracy issues in performing such a comparison. For example, the time-varying sinkage and trim in the PMM model tests may have quite a large influence on \( X' \). Support for this possibility can be seen in the duplicate CFD results that are shown in for the pure drift cases at drift angles of 0°, 9°, 10°, and 11°. These are due to repetition of calculations using the model test sinkage and trim, and using sinkage and trim as found by CFD. As the agreement in the sinkage and trim values are within 1.5 mm and 0.3°, respectively, this indicates a sensitivity of \( X' \) to sinkage and trim. Considering the above, the fair agreement between the Taylor series fit to the PMM model tests and the rotating arm CFD simulations is therefore not a conclusive indication of inaccuracy in the CFD technique.

In contrast, Figs. 10 and 11 show very good agreement for the \( Y' \) and \( N' \) results of the rotating arm simulations. The only deviations of note are in particularly extreme motion conditions. The discrepancy at \( r' = 0.3 \) and near 10° drift is not of much practical importance because this condition with the bow yawed away from the direction of turn is very uncommon in a real ship manoeuvre. The discrepancy at drift angles in excess of 11° are expected because the PMM test matrix did not include such high drift angles. The non-smooth trend shown by the CFD results near \( r' = 0.6 \) and no drift may be associated with imperfect convergence of that simulation, or challenges with predicting increasingly unsteady hydrodynamic phenomena with a steady simulation.

| Table 11. Derivatives of \( X' \) |
|-----------------|---------|---------|-----------|-----------|-----------|-----------|
| Source Data     | \( \varepsilon_{\text{RMS}} \) | \( X'_{\phi} \) | \( X'_{\phi} \) | \( X'_{\psi} \) | \( X'_{\tau} \) | \( X'_{\psi} \) |
| Steady CFD      | 1.21e–3 | −0.01606| −0.2970   | −0.06525   | 0.00997   |
| Unsteady CFD    | 1.12e–3 | −0.01634| −0.00774  | −0.2390    | −0.03931  | 0.00926   |
| Unsteady model test | 9.13e–4 | −0.01839| −0.01019  | −0.1683    | −0.03130  | 0.01889   |

| Table 12. Derivatives of \( Y' \) |
|-----------------|---------|---------|-----------|-----------|-----------|-----------|
| Source Data     | \( \varepsilon_{\text{RMS}} \) | \( Y'_{\phi} \) | \( Y'_{\phi} \) | \( Y'_{\psi} \) | \( Y'_{\tau} \) | \( Y'_{\psi} \) |
| Steady CFD      | 2.25e–3 | −0.2922 | −0.06088  | −10.03     | −0.1840   | −0.9790   | −0.8131   |
| Unsteady CFD    | 1.53e–3 | −0.1162 | −0.01137  | −0.2634    | −0.04403  | −14.33    | −0.3611   | −2.791    | −1.663   |
| Unsteady model test | 3.56e–4 | −0.1147 | −0.01394  | −0.2683    | −0.05555  | −14.77    | −0.3337   | −2.687    | −1.953   |

| Table 13. Derivatives of \( N' \) |
|-----------------|---------|---------|-----------|-----------|-----------|-----------|
| Source Data     | \( \varepsilon_{\text{RMS}} \) | \( N'_{\phi} \) | \( N'_{\phi} \) | \( N'_{\psi} \) | \( N'_{\tau} \) | \( N'_{\psi} \) |
| Steady CFD      | 9.74e–4 | −0.1651 | −0.04370  | −1.679     | −0.2360   | −0.9924   | −0.1361   |
| Unsteady CFD    | 4.31e–4 | −0.01191| −0.008944 | −0.1545    | −0.04527  | −3.673    | −0.2184   | −1.072    | −0.2895  |
| Unsteady model test | 1.13e–3 | −0.01339| −0.01003  | −0.1626    | −0.04347  | −3.185    | −0.2688   | −1.442    | −0.3369  |
DISCUSSION
A matter of practical importance to ship designers is the question of whether steady or unsteady CFD simulations are more appropriate for developing manoeuvring derivatives, and thereby predicting manoeuvring performance.

The validation results show that better agreement with the model tests was achieved for steady CFD simulations. The analysis above shows this can primarily be attributed to the assumption of constant sinkage and trim and the use of coarser meshes in the unsteady simulations. The difference in mesh size was motivated by the additional computational effort required for an unsteady simulation. This could not be mitigated by using more computation cores to run the unsteady simulations, because STAR-CCM+ has practical limits to the number of cells per core. It was typically found that an unsteady simulation (using a time step of 7/2880) required about 17 hours to solve using 60 calculation cores, a rotating arm simulation required about 5 hours to solve using 80 cores, and a steady drift simulation required about 9 hours to solve using 11 cores on a 12-core machine. The 80-core calculations were completed on a 12 node, 132 core cluster. Eleven of the nodes were Dell M610 blade servers each with two Intel Xeon X5660 processors at 2.80 GHz, 48 GB RAM, and a Mellanox Infiniband QDR card. The remaining blade server was a Dell Poweredge 501 NAS, with two Intel Xeon E5520 processors at 2.26 GHz, and 16 GB RAM. The cluster was running 64-bit Centos 6.5 with Rocks 6.1.1 (Sand Boa) Cluster Management. The 12-core machine was a Dell Precision T7500, with two Intel Xeon X5690 processors at 3.46 GHz, with 96 GB RAM, running 64-bit Windows 7 Professional.

The difference in computation time for individual steady versus unsteady simulations is partly offset by the number of simulations required. The work here (exclusive of time and space refinement runs, and manual sinkage and trim iterations) included 26 steady simulations and 12 unsteady simulations. The overall calculation time for the ensemble of steady simulations was roughly 20% less than required for the ensemble of unsteady simulations.

In addition to the trade-off between calculation time and numerical accuracy, a variety of other factors influence the selection of steady or unsteady simulations. A fundamental limitation of steady simulations is that they cannot predict unsteady effects. The best understood of these are the derivatives of manoeuvring forces with respect to vessel accelerations – or added mass effects. Added mass has traditionally been considered to be essentially an inviscid effect, and can be predicted using simpler potential flow techniques.

Other unsteady effects include the more intricate flow history effects, such as gradual growth, decay and movement of vortices as shown earlier. Inclusion of these effects can only be considered to be of benefit to the accuracy of the manoeuvring derivatives if the flow histories in the CFD analyses are reasonably representative of what would be observed in the real manoeuvres to be simulated using the derivatives. For example, a zig-zag manoeuvre may achieve a rotation rate similar to that specified in a given PMM simulation, but the time between rudder executes in the zig-zag manoeuvre may or may not be similar to half the motion period in the PMM simulation. If it is not, then the flow history effects represented in the derivatives would be incorrect. A more concrete example is a turning circle manoeuvre: after the initial phases of turn initiation, unsteady flow-history effects would be small, so the performance would be better predicted on the basis of steady CFD simulations.

One final consideration is that steady simulations are more straightforward to work with. When conducting unsteady simulations, it is necessary to determine what results must be logged during the simulation (otherwise the data is lost). However steady simulations can be run first, then the results interrogated in full detail subsequently. It is also more straightforward to judge the convergence of a steady simulation than to judge the time-accuracy and cyclic repeatability of an unsteady run. At the post-processing stage, it is simpler to interpret how a plot of the steady results and a curve fit of the Taylor series might indicate opportunities for improvement of either the CFD simulations or the form of Taylor series used to represent the results.

Perhaps the most reasonable path forward is to place the emphasis on carrying out steady simulations, and to include a select few unsteady simulations with a coarser mesh. For example, one or two unsteady pure drift simulations and one or two unsteady pure yaw simulations could be done. This would use a regression technique that combines both to develop the manoeuvring derivatives; it is a relatively straightforward extension of the technique applied here. It would allow direct determination of the added mass effects from the unsteady simulations, while benefiting from the finer mesh that can be used practically in steady simulations.

CONCLUSIONS
Verification and validation of CFD simulations of steady and unsteady captive manoeuvring model tests have been completed to investigate the reliability and practicality of CFD for ship manoeuvring prediction.

The spatial discretization error of the steady simulations was found to be small, and less important than the imperfect convergence due to a finite simulation time and unsteady flow features. The validation of steady simulations is very encouraging, with relative differences from physical model test results on the same order as the repeatability uncertainty of the model tests. Accurate resolution of smaller flow features like breaking waves and bubble sweep-down was not found to be important to the manoeuvring forces.

Due to the practicality of solving unsteady simulations in a reasonable time, they were completed using a coarser mesh. The mesh refinement study indicated that some Fourier coefficients of the unsteady results are not mesh-convergent for the set of meshes used. Temporal refinement indicated excellent time-accuracy of the predicted forces. The comparison of the Fourier coefficients of the unsteady force results to physical model test results is not
as encouraging as the steady validation, but differences are always less than 20%.

Manoeuvring derivatives calculated based on the PMM model tests were used to indirectly validate the steady rotating arm simulations. This showed excellent agreement for the lateral force and yaw moment.

Only unsteady simulations can predict derivatives with respect to vessel accelerations (i.e. added mass terms) and flow history effects. However the practicality of using a finer mesh for steady simulations suggests they would be more suitable for predictions of ship manoeuvring in the earlier stages of the ship design process.

As a relatively new application of CFD, there are considerable opportunities for future work. For example, control surfaces and other appendages, other hull forms, roll motions, speed effects, accuracy of simulated manoeuvres, scale effects, further spatial and temporal refinement, and dynamic sinkage and trim effects could all be pursued further.

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REFERENCES


