Validation of Ship Scale CFD Self-Propulsion Simulation by the Direct Comparison with Sea Trials Results

Dr. Dmitriy Ponkratov¹, Constantinos Zegos²
¹, ²Technical Investigation Department (TID), Lloyd’s Register, London, UK.

ABSTRACT
Even though Computational Fluid Dynamics (CFD) codes have been validated extensively by developers and researchers, most of this is against model scale experiments. Due to the difference in certain aspects of flow characteristics between ships and models it is important to validate against full scale measurements, however, these are scarce and even when available they are usually incomplete. Lloyd’s Register Technical Investigation Department (LR TID) has the capability to collect specific data during ship operations which even under normal circumstances pose numerous challenges and are costly to acquire. As part of the research work reported here, all these obstacles were overcome to culminate in the successful ship scale validation of the code and methods. This paper is a further development of the work reported by Ponkratov et al. (2014).

KEYWORDS
CFD, Self-Propulsion simulation. Thrust and Torque measurements.

1. INTRODUCTION
As a result of recent advances in numerical methods and a reduction in cost of computational power, complex hydrodynamic modelling challenges, such as ship scale self-propulsion, can now be investigated by researchers using CFD.

The investigation presented in this paper required a careful implementation of different numerical approaches, such as the Volume of Fluid (VOF) method to resolve the water free surface and sliding mesh approach for propeller rotation. Each of these approaches has to be validated in order to obtain an accurate performance prediction of a ship using CFD. Such validations are routinely carried out at model scale, where it is possible to control the experimental setup to a high degree and variables can be monitored relatively easily. As a result, researchers have a wealth of information regarding the comparisons between model tests and CFD, especially for resistance tests of bare hulls and propeller open water tests. The next logical step is CFD calculation of a self-propelled vessel at model scale. Until recently, these cases were quite scarce, due to the computational power required to carry out simulations. Researchers were therefore forced to implement various simplifications.

One such simplification is to use an actuator disk method instead of the actual propeller. This approach avoids complexities associated with moving meshes resulting in shorter run times, whilst providing reasonable results, as shown by Turnock et al. (2008), Pacuraru et al. (2011).

A second simplification to the computational setup is to disregard the free surface, consequently avoiding uncertainties of the VOF model, as presented by Krasilnikov (2013).

The third simplification is to calculate the free surface but not to calculate the dynamic sink and trim of the vessel, as reported by Turnock et al. (2008), Bugalski et al. (2011). This approach helps to avoid the uncertainties caused by the DFBI model; hence the simulation convergence is faster.

With the increase in computational power it became possible to avoid simplifications and to calculate the self-propulsion using the actual towing tank set up (Dhinesh et al. 2010, Carrica et al. 2011, Shen 2012, Lin et al. 2012) and directly compare the results with model scale data. These calculations are very important from a research point of view, as they introduce a new numerical towing tank concept, allowing for model tests to be carried out numerically.

Unfortunately, both model scale calculations and model tests have a recognised disadvantage, known as the scale effect. In order to avoid this issue the simulation of self-
propulsion at full scale is necessary. The same simplifications mentioned previously, valid for the model scale calculations, can be applied to ship scale simulations to the same effect. For example, the concept of actuator disk for the full scale simulations has been successfully used by Tzabiras (2001), Kim et al. (2005), Tzabiras et al. (2009).

Undoubtedly, the most advanced scenario is to simulate the self-propulsion at full scale without any assumptions, i.e. with the free surface, with the real propeller and free sink and trim. At the time of writing this paper the authors were aware of one published simulation performed in these conditions and reported by Castro et al. (2011). As these researchers did not have data from sea trials, they had to compare their results with scaled quantities obtained empirically according to the ITTC procedures.

Lloyd’s Register (LR), as the oldest classification society and one of the leaders in hydrodynamic research, had a unique opportunity to collect and analyse data from model scale tests, as well as full scale trials for the purpose of validating CFD predictions carried out in-house. For the current study a medium range tanker has been selected and all required information, such as model test results, hull, propeller, rudder and appendages drawings have been collected. The main particulars of the vessel are presented in Table 1.

### Hull particulars

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length overall to beam ratio</td>
<td>Loa/B</td>
</tr>
<tr>
<td>Waterline length to beam ratio</td>
<td>Lwl/B</td>
</tr>
<tr>
<td>Length between PP to beam ratio</td>
<td>Lpp/B</td>
</tr>
<tr>
<td>Beam to design draught ratio</td>
<td>B/T</td>
</tr>
<tr>
<td>Block coefficient</td>
<td>Cb</td>
</tr>
<tr>
<td>Prismatic coefficient</td>
<td>Cp</td>
</tr>
<tr>
<td>Midship coefficient</td>
<td>Cm</td>
</tr>
<tr>
<td>Waterplane coefficient</td>
<td>Cw</td>
</tr>
</tbody>
</table>

### Propeller particulars

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of blades</td>
<td>Z</td>
</tr>
<tr>
<td>Diameter to design draught ratio</td>
<td>D/T</td>
</tr>
<tr>
<td>Expanded Blade Area ratio</td>
<td>Ae/A</td>
</tr>
<tr>
<td>Pitch coefficient</td>
<td>Pe/D</td>
</tr>
</tbody>
</table>

### Rudder particulars

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area to WL length and draught</td>
<td>A/Lwl*T</td>
</tr>
</tbody>
</table>

**Table 1. Ship main particulars.**

Procedures developed and validated in the laboratory of the LR Technical Investigation Department (LR TID), were applied successfully to obtain the values of propeller thrust and torque of the vessel during sea trials. In addition, the vessel's speed, shaft rotation rate and rudder angle were also recorded during the trials.

**Figure 1. Full scale vessel modelled in CFD.**

Commercial CFD software STAR-CCM+ was used to carry out simulations of the ship in the same conditions as the sea trials. In order to have an accurate representation of the ship, all hull appendages were modelled. The final goal was to simulate ship, free to sink and trim. However, as the work is still in progress, in the current paper all presented CFD results are interim and calculated for a fixed sink and trim condition.

At the first stage, the comparison of CFD results for propeller open water characteristics, ship resistance and propulsion was performed against the model scale tests. As this showed good agreement in all aspects, the work progressed to the second stage, which included full scale simulations of sea trials.

## 2. SEA TRIALS DATA

### 2.1 REASONS FOR COLLECTING DATA

As mentioned earlier, the main technical challenge of any full scale CFD self-propulsion simulation is to perform calculations without any major simplifications; e.g. with free surface, rotating propeller and all appendages. However, the principal difficulty is to ensure that calculated values are in line with the measured ones; hence, the comparison and validation of the results is necessary. In the case of model scale calculation, the validation can be performed by direct comparison, as the major quantities, such as resistance, propeller thrust and torque are measured in a controlled environment during the tests. At ship scale the direct comparison is limited, since usually only a few basic quantities are measured during sea trials. In this situation there are two possible alternatives: either to compare the full scale CFD results
with the predictions based on ITTC scaling procedures or
to measure additional quantities during actual sea trials.
Traditionally, the first alternative is the simplest one as the
ITTC procedures are well known, for example the
comparison with scaled ITTC procedure for KCS
containership has been completed by Castro et al. (2011).
However, it should be noted that these predictions rely on
empirical functions, so may not always be accurate. In
particular, special attention should be paid when the
scaling procedures are used for vessels with energy saving
devices, such as ducts, pre-swirl stators or vortex
generators, as strictly speaking the ITTC scaling
procedures have not been validated for these devices.
The second alternative is less common, as it involves
additional undertaking for the measurement of the
required quantities. During the sea trials conducted by a
shipyard, the practical interest lies in the confirmation of
the contractual requirements, such as ship speed at a
specific engine power. Hence, only the ship speed, engine
power and shaft speed are recorded as a matter of course.
It should be noted that these records are dependent on
environmental factors such as wind, waves and current.
Some ITTC or ISO corrections should be applied in such
cases. Hence, the remaining challenge is the lack of
reliable recorded variables for direct comparison between
CFD calculations and sea trials. Over many years, LR TID
has developed extensive experience in performing
complex measurements as a result of attending sea trials
of different types of vessels.
One of the authors attended the sea trials of the medium
range tanker and conducted the measurements of ship
speed, engine power, shaft speed, propeller thrust and
torque. Before attending the trials all equipment was
carefully checked and calibrated in the LR TID laboratory.
Prior to departure for the sea trials the hull was cleaned.
Figures 2 and 3 show the hull conditions before and after
the cleaning. For comparison, Figure 4 shows the surface
mesh on the bilge keel. The propeller surface was also
polished. As an example, Figures 5 and 6 show the
propeller hub surface before and after polishing.
The speed tests were conducted under various power
conditions at ballast, design and scantling draughts. The
aforementioned quantities were measured for all tests.
In the current study only one case has been selected, as the
weather was exceptionally calm for its duration. As a
result, the ship showed almost identical shaft speed,
power, thrust and torque for the “forth” and “back” runs,
Table 4.
It was therefore considered that the environmental impact, due to waves and wind was minimal for this particular test. The ship speed recorded by GPS was slightly different between reciprocal runs due to the current; therefore the final speed was calculated as an average between the two runs. No other corrections, such as ITTC corrections for weather were applied, as they were not required due to the favourable environmental conditions.

2.2 SHIP SPEED
The ship speed was recorded for all double runs by an independent GPS system installed by LR TID on the navigation bridge. Figure 7 shows the track of the runs for speed tests.

2.3 SHAFT SPEED
In order to record the shaft speed, a reflecting tape was glued on the shaft line and an optical sensor was mounted close to the shaft, as shown in Figure 8.

Each time the tape passed the sensor window a voltage signal was transmitted to a recording device. For ease of reference, the tape was fitted to the shaft in line with the sensor window when the key blade was located in the top dead centre, resulting in a simultaneous recording of key blade position.

2.4 TORQUE AND THRUST
The propeller thrust and torque were measured by strain gauges similar to those used in Formula 1 cars for various measurements, including the wheel axis moment.
The installation of the gauges on the shaft was the most difficult task due to gauge sensitivity, which required careful treatment. Before each gauge was fitted, the corresponding area of the shaft was manually polished with sand paper and longitudinal and transversal axes were drawn, Figure 9. Before and after the installation, all strain gauges were tested by the strain indicator and the gauges’ Out of Balances (OOB) were identified.

Figure 9. Strain gauge for the torque measurements attached to the shaft.

The amplifier and battery pack were also mounted to the shaft, as shown in Figure 10. Six batteries allowed the continuous working of the system for the duration of the sea trials.

Figure 10. The installation for the torque measurements.

The measurement of the propeller torque requires only one full-bridged strain gauge and the values are usually attained to a high degree of accuracy. The torsional deformation of the shaft, usually in the order of a hundred micro strains, is generally much higher than equipment tolerances, usually ten micro strains. The propeller thrust on the other hand is the most challenging value to measure, as the longitudinal deformation of the shaft is in the same order of magnitude as the equipment tolerances. Hence, even careful installation of the thrust strain gauges does not guarantee a high accuracy of measurements. In order to build a full bridge, two thrust strain gauges were glued on the opposite sides of the shaft, Figure 12 and a heating compensator was applied. Figure 13 shows the installation for thrust measurements.

Figure 11. The signal pick up mounted close to the aerial.

Figure 12. One of the two strain gauges used for the thrust measurements attached to the shaft.

Before commencement of thrust and torque measurements it is necessary to determine the zero level, when there are no forces or moments acting on the shaft e.g. when the shaft is stationary. Theoretically, this can be done after the installation whilst the ship is alongside. However, the zero level on the “cold” shaft will not necessarily be the same.
as on the “warmed” shaft during the trials, hence the heating impact would be ignored.

Stopping the shaft at sea to determine the zero level of a heated shaft is also impractical, as the ship movement due to the inertia or current, would create forces and moments even on the stopped shaft. The only practical solution, which was applied in the current case, was to record zero thrust and torque after the arrival from sea trials, when the ship was already alongside and the shaft was still “warm”. The recorded strain values were thus post processed, in order to obtain the values for the thrust and torque.

Figure 13. The installation for the thrust measurements.

As several strain gauges were installed on the shaft, it was possible to cross check the measured values between them. As stated earlier, the torque results are usually the most reliable and the cross check confirmed this, revealing the difference was within 1% from two independent torque sensors. For the thrust cross check the difference from two independent installations was within 6%. Hence, the CFD results were compared with two values for the thrust and torque obtained from independent analysis.

2.5 RUDDER MOVEMENT

Apart from the strain gauges, a draw wire sensor was mounted on the rudder stock, in order to measure the rudder angles. After the installation, shown in Figure 14, and whilst the ship was alongside, the rudder was applied in the range of 20° port to 20° starboard, in 5° steps, in order to calibrate the sensor and to determine the dependence of the wire displacement on the rudder angle. After calibration, the resulting coefficient was applied in the recording software in order to acquire the signal directly in degrees. As it was possible to charge and discharge cargo tanks during the sea trials, measurements were performed for scantling, design and ballast draughts.

Figure 14. Voltage displacement draw wire connected to rudder stock.

The weather conditions were best on the day when the vessel was in design draught; it was therefore decided to perform the CFD study for the same draught, helping to further minimise the uncertainties related to environmental impact.

3. CFD METHODOLOGY

3.1 GEOMETRY

The supplied hull geometry was in the form of an IGES file, however, the tessellation quality was low. Therefore the surface wrapper was used to obtain the geometry definition. Despite this procedure, which involved deriving the geometry from the original, the hull dimensions were compared at numerous frames along the length of the ship and showed a high degree of accuracy. The geometries of all appendages were built by LR TID using the ship’s drawings.

For both model scale and ship scale simulations the geometry used matched the respective case for the measurements. For the resistance test, only the naked hull with the rudder was modelled, whereas for the self-propulsion case, the propeller was added. In ship scale all appendages were included, i.e. the bilge keels and rudder. In addition, the superstructure was included, to more accurately account for air resistance.

3.2 COMPUTATIONAL DOMAIN

The computational domain extended to about one ship length fore of the bow, two ship lengths aft of the stern, one ship length to both port and starboard and one ship length below and above the water surface.

For simulations excluding the propeller, only half a ship was modelled, taking advantage of the symmetry about the centreline. An illustration of the computational domain is
provided in Figure 15. A separate cylindrical domain was created for the propeller to allow for the rigid body rotation to be modelled. Internal interface boundary conditions were implemented on the cylinder faces between the rotating and static domains. A trimmed mesh was used and was aligned to the still water free surface. The Volume of Fluid (VOF) method was used to capture the deformation of the free surface.

![Figure 15. Computational domain around the ship.]

The thickness of the first cell close to the hull and propeller geometries was set such that the $y+$ values were within 30 to 500 for the full scale calculations and in the order of 1 for the model scale ones. The initial computational mesh for the ship scale case was approximately 18 million cells, however, after a careful sensitivity study the number of cells for the final mesh was reduced to approximately 10 million. It has been confirmed that the results obtained on the coarser mesh are in good agreement with ones obtained on the finest mesh. Even with the final mesh, the total computational time took approximately 24 hours per self-propulsion case, using 128 cores of the LR TID cluster.

3.3 MODELLING APPROACH

The system of RANS equations for mass and momentum transfer was closed with the $k$-$\omega$ SST turbulence model. The transport equations of the problem are solved by a segregated solution method with a SIMPLE-type algorithm for pressure-velocity coupling. The unsteady simulation of the rotating propeller was resolved using a time-accurate sliding mesh approach. However, on the first stage of the simulation a Moving Reference Frame (MRF) approach was used, to accelerate the resolution of free surface.

In all cases, the simulations were set up to match the inputs and unknowns, as in the physical world. The model scale simulations of naked hull resistance were set up with the following key parameters, matching those of the towing tank resistance test:
- Trim;
- Draught;
- Towing speed.

In addition, the propeller shaft speed was set up for the self-propulsion simulation. For the resistance tests the hull resistance was monitored and compared against the model scale results. For the self-propulsion simulation the propeller thrust and torque were also monitored.

The calculation strategy for the fixed sink and trim, self-propulsion case in ship scale was the following: at the first stage the calculation was performed with the speed corresponding to the mean speed measured during the trials and propeller rpm modelled by the Moving Reference Frame approach. The trim and sink were taken from the model scale report. The time step was selected to be 0.05 sec. Once this calculation had converged and the free surface was fully resolved the time step was reduced to 0.0032 sec and propeller rigid body rotation activated. Once the convergence was achieved, the imbalance of longitudinal forces between propeller thrust and effective ship resistance was noted and the shaft speed was adjusted in order to minimise this imbalance. The calculated values for the model and ship scale simulations are presented in Tables 3 and 5.

4. RESULTS

The results from this project are the comparisons between the CFD predictions and the measurements taken during the towing tank tests and the sea trials.

4.1 MODEL SCALE SIMULATIONS

The naked hull resistance and propeller open water characteristics were validated separately before proceeding to a self-propulsion simulation. Results in Table 2 show the resistance comparison.

<table>
<thead>
<tr>
<th>Fr</th>
<th>$C_{HRES} \times 1000$ Model test</th>
<th>$C_{HRES} \times 1000$ CFD</th>
<th>$\Delta$, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1332</td>
<td>3.980</td>
<td>4.038</td>
<td>1.5</td>
</tr>
<tr>
<td>0.1574</td>
<td>3.923</td>
<td>3.956</td>
<td>0.8</td>
</tr>
<tr>
<td>0.1817</td>
<td>3.862</td>
<td>3.836</td>
<td>-0.7</td>
</tr>
<tr>
<td>0.2059</td>
<td>4.312</td>
<td>4.210</td>
<td>-2.4</td>
</tr>
</tbody>
</table>

Table 2. Comparison resistance model tests and CFD.

In all tables the Froude number (Fr) is based on the hull waterline length $L_{WL}$. The standard formula for coefficient of total hull resistance ($C_t$), is used where the reference area is the hull wetted surface area. The naked hull resistance results were regarded as acceptable, as the error margin...
was less than 2.5% in the worst case and generally much lower for the majority of cases.

Figure 16 shows the results of propeller open water calculations compared to the measurements performed in the towing tank. It was found that for the propeller with such a low expanded blade area ratio, the standard simulation setup showed a discrepancy with thrust coefficient $K_t$ of approximately 9% and torque coefficient $K_q$ of approximately 6%, for the advance coefficient $J$ of around 0.5. Through a series of tests it was concluded that this was due to the unavoidable overestimation of the turbulence levels due to early transition from laminar to turbulent flow. In order to confirm this assumption, the case was recalculated with the employment of Gamma Re Theta transition model. It presented an improvement of the integral characteristics, so that the discrepancy for $K_t$ became approximately 2% and for $K_q$ about 4% and for $J$ values around 0.5.

The self-propulsion model scale simulation was a final testament of the setup and computational procedure. Table 3 shows the results of these calculations including wake fraction $W_t$, thrust deduction factor $t$ and relative rotative efficiency $\eta$. As model self-propulsion tests were performed in a towing tank for a number of speeds, it was possible to interpolate the data and obtain the results for a particular speed. Hence, for this calculation the ship model speed was selected corresponding to Froude number identical to the full scale case. During the calculations it was found that standard turbulence model under-predicts the ship wake. This is probably due to highly anisotropic turbulent flow behind the hull with such a high block coefficient. As a result, the propeller thrust and torque showed lower values at the shaft speed, similarly to model test values. In order to achieve the self-propulsion point the shaft speed was adjusted, minimising the imbalance between propeller thrust, effective resistance and skin friction correction force. The value for the skin friction correction was taken from the model test report. As stated in Table 3 the shaft speed was increased by approximately 2% to achieve the self-propulsion point. One of the possible solutions to minimise this discrepancy is to improve the ship wake. Further investigations employing the anisotropic Reynolds Stress Model are planned to be implemented.

### Table 3. Comparison self-propulsion model test and CFD.

<table>
<thead>
<tr>
<th></th>
<th>EXP</th>
<th>CFD</th>
<th>$\Delta$, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n/(V_0/D)$</td>
<td>1.3610</td>
<td>1.3908</td>
<td>2.19</td>
</tr>
<tr>
<td>$Fr=V_0/(gLWL)^{0.5}$</td>
<td>0.1933</td>
<td>0.1933</td>
<td>-</td>
</tr>
<tr>
<td>$K_t=T/(\rho n_{ref}^2 D^4)$</td>
<td>0.1964</td>
<td>0.1965</td>
<td>0.05</td>
</tr>
<tr>
<td>$10K_q=10Q/(\rho n_{ref}^2 D^5)$</td>
<td>0.2496</td>
<td>0.2413</td>
<td>-3.32</td>
</tr>
<tr>
<td>$C_{t, SP}=R_{SP}/(0.5 \rho V^2 H_{WET})$</td>
<td>4.8205</td>
<td>4.8133</td>
<td>-0.15</td>
</tr>
<tr>
<td>$1-W_t$</td>
<td>0.5960</td>
<td>0.5895</td>
<td>-1.09</td>
</tr>
<tr>
<td>$1-t$</td>
<td>0.7794</td>
<td>0.7895</td>
<td>1.30</td>
</tr>
<tr>
<td>$\eta$</td>
<td>1.0040</td>
<td>1.0190</td>
<td>1.49</td>
</tr>
</tbody>
</table>

Figure 17. Convergence of thrust and torque coefficients for ship scale self-propulsion are presented in Figure 17. It is worth noting that the calculation until 300 sec of the physical time was completed using MRF approach, and thereafter the sliding mesh approach was used. The natural four peaks per revolution oscillations of thrust and torque coefficients, due to rotation of four-bladed propeller, are shown in the magnified part of Figure 17. The averaged results of this validation are presented in Table 5. The power in Tables 4 and 5 is presented in a form of the Power Number $N_P = P/(\rho n_{ref}^3 D^5)$, where $P$ is the propulsion power (W), $\rho$ is the water density (kg/m$^3$), $n_{ref}$
is the shaft rotational speed recorded during the sea trials (1/s) and D is the propeller diameter (m).

<table>
<thead>
<tr>
<th></th>
<th>Forth run</th>
<th>Back run</th>
<th>Mean</th>
</tr>
</thead>
<tbody>
<tr>
<td>n/(V_{mean}/D)</td>
<td>1.4048</td>
<td>1.4048</td>
<td>1.4048</td>
</tr>
<tr>
<td>Fr by GPS</td>
<td>0.2010</td>
<td>0.1856</td>
<td>0.1933</td>
</tr>
<tr>
<td>Kt 1st gauge</td>
<td>0.1681</td>
<td>0.1689</td>
<td>0.1685</td>
</tr>
<tr>
<td>Kt 2nd gauge</td>
<td>0.1784</td>
<td>0.1791</td>
<td>0.1787</td>
</tr>
<tr>
<td>10 Kq 1st gauge</td>
<td>0.1895</td>
<td>0.1901</td>
<td>0.1898</td>
</tr>
<tr>
<td>10 Kq 2nd gauge</td>
<td>0.1893</td>
<td>0.1901</td>
<td>0.1897</td>
</tr>
<tr>
<td>Power 1st gauge</td>
<td>0.1191</td>
<td>0.1194</td>
<td>0.1193</td>
</tr>
<tr>
<td>Power 2nd gauge</td>
<td>0.1189</td>
<td>0.1194</td>
<td>0.1192</td>
</tr>
</tbody>
</table>

Table 4. Sea trials results.

It can be seen from the results in Table 5 that the torque coefficient (Kq) predicted by CFD agreed well with both coefficients measured during the sea trials by the independent gauges. The thrust coefficient (Kt) however, agreed well with the coefficient obtained from the first gauge. The second gauge showed a discrepancy of approximately 7% when the result was compared with the thrust coefficient predicted by CFD.

<table>
<thead>
<tr>
<th></th>
<th>Trials</th>
<th>ITTC</th>
<th>Δ, %</th>
<th>CFD</th>
<th>Δ, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>n/(V_{mean}/D)</td>
<td>1.4048</td>
<td>1.3428</td>
<td>-4.41</td>
<td>1.4183</td>
<td>0.96</td>
</tr>
<tr>
<td>Fr</td>
<td>0.1933</td>
<td>0.1933</td>
<td>-</td>
<td>0.1933</td>
<td>-</td>
</tr>
<tr>
<td>Kt 1st gauge</td>
<td>0.1685</td>
<td>0.1632</td>
<td>-3.14</td>
<td>0.1660</td>
<td>-1.47</td>
</tr>
<tr>
<td>Kt 2nd gauge</td>
<td>0.1787</td>
<td>-8.70</td>
<td>-</td>
<td>0.1660</td>
<td>-7.12</td>
</tr>
<tr>
<td>10 Kq 1st gauge</td>
<td>0.1898</td>
<td>0.2064</td>
<td>8.75</td>
<td>0.1932</td>
<td>1.78</td>
</tr>
<tr>
<td>10 Kq 2nd gauge</td>
<td>0.1897</td>
<td>8.82</td>
<td>-</td>
<td>0.184</td>
<td></td>
</tr>
<tr>
<td>Power 1st gauge</td>
<td>0.1193</td>
<td>0.1240</td>
<td>3.96</td>
<td>0.1225</td>
<td>2.75</td>
</tr>
<tr>
<td>Power 2nd gauge</td>
<td>0.1192</td>
<td>4.02</td>
<td>-</td>
<td>2.82</td>
<td></td>
</tr>
</tbody>
</table>

Table 5. Comparison sea trials, ITTC78 and CFD.

As discussed earlier, the propeller thrust is the most challenging value to measure, since the longitudinal deformation of the shaft is of the same order of magnitude as the equipment error. Hence, even careful installation of thrust strain gauges does not guarantee a high accuracy of measurements.

Compared to results published by Ponkratov, (2014) those presented in Table 5 are obtained for a slightly finer mesh on the propeller. Also, to achieve the self-propulsion point, the propeller speed was adjusted, as opposed to Ponkratov (2014) where ship speed was used to this end, whilst the shaft speed was kept constant. As discussed earlier, the reason may be an under-prediction of ship wake due to the isotropic turbulence model.

From a practical point of view it is important to compare the sea trials results to estimations completed by ITTC78 method, based on the model test result. It can be seen in Table 5 that ITTC method under-predicted the shaft speed (approximately 4%) and propeller thrust (3% compared to the first sensor and 9% compared to the second one). At the same time it over-predicted the propeller torque by approximately 9% and slightly over-predicted the power by approximately 4%.

Even more important is to compare ship to propeller interaction characteristics, as they play a major role in the propulsion performance. These characteristics cannot be measured in ship scale, as it requires a series of propeller open water and bare hull resistance measurements. However, it can easily be done in CFD for the ship scale propeller and hull. These values were compared with the results obtained from model tests and further scaled by ITTC78 procedure. It can be seen from Table 6 that the discrepancy is within 8-10% for thrust deduction factor, wake fraction coefficient and relative propeller efficiency.

<table>
<thead>
<tr>
<th></th>
<th>ITTC78</th>
<th>CFD</th>
<th>Δ, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-Wt</td>
<td>0.6483</td>
<td>0.7171</td>
<td>10.6</td>
</tr>
<tr>
<td>1-t</td>
<td>0.7667</td>
<td>0.7083</td>
<td>-7.61</td>
</tr>
<tr>
<td>η_p</td>
<td>1.0209</td>
<td>1.1013</td>
<td>7.87</td>
</tr>
</tbody>
</table>

Table 6. Comparison ITTC78 and ship scale CFD.

As the values calculated by CFD shown in Table 5 are in better agreement with ship scale measurements than ITTC78 prediction, it may be deduced that the interaction coefficients predicted by CFD are closer to reality than those scaled from model tests. However, as mentioned earlier it is virtually impossible to validate and confirm such a statement.
5. CONCLUSIONS

Up to the present time the following conclusions can be drawn from the investigation:

- LR TID collected a full complement of documentation and measurement data for a selected medium range tanker, including all model test reports, hull, rudder, propeller and appendages drawings, as well as the IGES file of the hull.
- LR TID conducted the sea trials for the tanker. Prior to the trials the ship hull was cleaned and the propeller surface was polished. The sensors of the type used for measurements on Formula 1 cars were applied successfully to obtain the actual values of propeller thrust and torque.
- Results from two runs of one speed test have been selected for the CFD validation for this particular trial; the environmental influence was minimal and the power, thrust, torque and shaft speed were almost identical, both for “forth” and “back” runs.
- Commercial CFD software STAR-CCM+ was used to carry out simulations in model and ship scales. Hull resistance, propeller open water and self-propulsion tests in model scale were simulated numerically and agreed well with model tests results.
- Full scale self-propulsion simulation of the ship was performed at the same conditions recorded at the sea trial. In order to have an accurate representation, all hull appendages, as well as the superstructure, were modelled.
- ITTC78 prediction method based on the model test results under-predicted the shaft speed and propeller thrust, over-predicted the propeller torque and showed satisfactory agreement in terms of delivered power.
- Agreement of thrust and torque between CFD calculations and the sea trials measurements was very good, confirming the high accuracy of the proposed CFD approach.

6. ACKNOWLEDGEMENT

The current research project was funded under the internal LR R&D program. The authors would like to thank their colleagues D. Radosavljevic, S. Whitworth, K. Kalaskar, A. Caldas and C. Craddock for their significant support.

7. REFERENCES


