Predicting the Effect of Hull Roughness on Ship Resistance using a Fully Turbulent Flow Channel and CFD

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Abstract: The effects of poor hull surface conditions on fuel consumption and emissions are well-known yet not thoroughly understood. Therefore, the present study investigates the effect of widely adopted fouling control coatings and mimicked biofouling on a full-scale representative ship, the KRISO Containership (KCS). Different surfaces were tested in the Fully turbulent Flow Channel (FTFC) of the University of Strathclyde (including a novel hard foul-release coating, commonly used antifouling, barrier resin, soft foul-release coatings, and sandpaper-like surfaces). Then, the corresponding roughness functions developed for the test surfaces were embedded in Computational Fluid Dynamics (CFD) simulations using the modified wall function approach. Interestingly, the numerical predictions on the KCS hull showed that the novel hard foul-release coating tested had better hydrodynamic performance than the smooth case (maximum 3.6% decrease in the effective power requirements). Eventually, the present study confirmed the practicality of the FTFC used in combination with CFD-based studies to predict the effects of hull roughness on ship resistance and powering.

Keywords: Fully Turbulent Flow Channel; Roughness Functions; Ship Resistance; Marine Coatings; Computational Fluid Dynamics.

1 INTRODUCTION

A ship’s hull surface condition is crucial to its hydrodynamic performance (Schultz, 2007). Hence, choosing the right fouling control coating (FCC) and drydock strategies for a vessel can offer significant economic and environmental advantages. Theoretical and numerical methods based on the turbulent boundary layer similarity law scaling technique, which was proposed by Granville (Granville, 1978, 1958), can accurately predict the hull roughness effect on ship resistance, provided that the roughness function of the surface is known (Demirel, 2015).

The aim of this study is to obtain new roughness functions for commonly used marine coatings and biofouled hull conditions from Fully turbulent Flow Channel (FTFC) experiments and predict their effect on full-scale ship resistance and powering. Also, an important objective was to utilise the FTFC of the UoS, which is a more practical facility than a towing tank. Therefore, various types of FCCs were tested in the FTFC, including antifouling, soft foul-release, barrier resin coatings and the newly developed and patented hard foul-release coating (FR02) by Graphite Innovation & Technologies (GIT, 2021). Similarly, roughness functions were developed from FTFC tests for widely adopted sandpaper-like surfaces mimicking biofouled conditions (medium light slime and medium slime) as similarly done in towing tests (Schultz, 2004; Song et al., 2021c). Furthermore, the roughness functions developed for a sandpaper-like surface (Sand 220) from the FTFC experiments was compared with previous towing tank tests. Finally, the present study also aims to confirm the robustness of CFD-based methods to predict the effect of hull roughness on ship resistance and powering using FTFC-based roughness function models.

The remaining of the paper is structured as follows: Section 2 presents the methodology adopted, including the experimental setup, roughness functions development, CFD simulations, and experimental uncertainty analysis. Section 3 of the paper discusses the results of the current experimental and numerical investigation. Furthermore, the novel roughness functions of the test surfaces are presented and used to predict the variation of resistance coefficients and effective power requirements for the full-scale KRISO Container Ship (KCS) hull. Section 4 presents the conclusions of the study with some concluding remarks and recommendations for future studies.

2 METHODOLOGY

2.1. Approach

Figure 1 shows a schematic illustration of the experimental and numerical methodology adopted to investigate the roughness effects of marine coatings and hull roughness on the well-known KRISO Container Ship (KCS) (“KCS Geometry and Conditions,” 2008). Drag characterisation
of arbitrary rough surfaces on flat plates can be evaluated by the indirect method for pipes (Granville, 1987) that uses the pressure drop \( \Delta p \) which can be measured along the streamwise length of the coatings (i.e., the pressure drop method). The FTFC was used to determine the skin friction coefficients \( c_f \), by measuring the pressure drop \( \Delta p \) on the test surfaces. Eventually, the roughness functions for the test surfaces were obtained (i.e., roughness functions, \( \Delta U^+ \), roughness Reynolds numbers \( k^+ \), roughness length scale, \( k \), etc.), and compared with literature, e.g., previous towing tests (Yeginbayeva et al., 2020).

The modified wall function CFD simulations were adopted in the present study to predict the effect of the test surfaces on the full-scale KCS hull. The experimental roughness functions were embedded in CFD using the modified wall function approach to predict the effect of such surfaces on ship resistance and powering. The resistance coefficient results of the numerical predictions were then compared and validated across similar studies assessing the KCS resistance in smooth and rough conditions (Ravenna et al., 2022a; Song et al., 2020; Yeginbayeva et al., 2020). Finally, the variations in effective power, \( \Delta P_E \) due to each test surface were estimated to give an immediate understanding of the effects of marine coatings and hull roughness on ship resistance and powering. Comparison and validation of the \( \Delta P_E \) values were conducted across the two numerical methods adopted and among similar studies (Schultz et al., 2011).

The FTFC enables the measurement of much higher flow speeds that would not be otherwise achievable in a typical towing tank with flat friction test plates (Ravenna et al., 2019). Table 1 summarises the main particulars of the FTFC upper limb section. For more information on the FTFC design, operation and calibration, the reader is advised to see (Marino et al., 2019).

![Figure 1: Schematic illustration of the methodology adopted.](image1)

![Figure 2: The Fully Turbulent Flow Channel (FTFC) of the University of Strathclyde.](image2)

**Table 1: Main particulars of FTFC upper limb.**

<table>
<thead>
<tr>
<th>Name</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length (Tolerance)</td>
<td>L</td>
<td>mm</td>
<td>3000 (±0.05)</td>
</tr>
<tr>
<td>Height (Tolerance)</td>
<td>H</td>
<td>mm</td>
<td>22.5 (±0.05)</td>
</tr>
<tr>
<td>Beam (Tolerance)</td>
<td>b</td>
<td>mm</td>
<td>180 (±0.05)</td>
</tr>
<tr>
<td>Mean bulk velocity range</td>
<td>( U_b )</td>
<td>m/s</td>
<td>1.5 – 13.5</td>
</tr>
<tr>
<td>Flow rate</td>
<td>Q</td>
<td>l/s</td>
<td>10 – 60</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>( Re_b )</td>
<td></td>
<td>( \approx 3.0 \times 10^6 )</td>
</tr>
<tr>
<td>Material</td>
<td></td>
<td></td>
<td>Stainless steel (316L)</td>
</tr>
<tr>
<td>Centrifugal Pump power</td>
<td>P</td>
<td>kW</td>
<td>22</td>
</tr>
</tbody>
</table>

### 2.2.2. Test Panels Design and Preparation

In the present experimental campaign, four different types of FCCs were tested in the FTFC, including the newly developed hard foul-release coating (FR02) manufactured by GIT and marine coatings type that are commonly used in the shipping industry manufactured at Dalhousie University (DU), i.e., a self-polishing antifouling coating (AF01), a gelcoat barrier coating (BL01), and a soft foul-release coating (FR01). Furthermore, two sandpaper-like surfaces mimicking slime biofouling, i.e., Sand 220
(medium light slime) and the coarser, Sand 60-80 (medium slime) manufactured at the UoS were tested. The coated panels (Figure 3-a) were tested along with an uncoated “control surface” or the “reference” to represent a hydraulically smooth surface, Figure 3-b. Table 2 describes the dimensions of the test panels, while a breakdown of the type of each marine coating applied and the method of application is provided in Table 3.

![Smooth Reference](image)

**Figure 3:** Surfaces tested in the FTFC.

### Table 2: Dimensions of the FTFC test panels.

<table>
<thead>
<tr>
<th>Dimension</th>
<th>[mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner length</td>
<td>599</td>
</tr>
<tr>
<td>Inner breadth</td>
<td>210</td>
</tr>
<tr>
<td>Inner thickness</td>
<td>14</td>
</tr>
<tr>
<td>Outer length</td>
<td>662</td>
</tr>
<tr>
<td>Outer breadth</td>
<td>282</td>
</tr>
<tr>
<td>Outer thickness</td>
<td>16</td>
</tr>
</tbody>
</table>

**Figure 4:** Pressure taps distribution numbered from 1 to 6 on test section of the FTFC.

#### 2.3. Roughness Functions Determination

Roughness Function (or velocity loss function), $\Delta U^*$, is further retardation of flow in the boundary layer over a rough surface due to the physical roughness of that surface, which manifests itself as additional drag relative to a smooth surface. Different surfaces are characterised by different roughness functions to be modelled experimentally (Granville, 1958). The roughness function, $\Delta U^*$ is a function of the roughness Reynolds number, $k^+$, which is defined by Eq (5):

$$ k^+ = \frac{k U_\tau}{v} $$

where, $k$ is the roughness length scale of the surface, and $U_\tau$ is the friction velocity based on wall shear stress defined by Eq (6):

$$ U_\tau = \sqrt{\frac{\tau_w}{\rho}} $$

where, $\tau_w$ is the wall shear stress.

For this study, the indirect method for fully developed pipe flow proposed by (Granville, 1987) is used to calculate the roughness function $\Delta U^*$ and roughness Reynolds number $k^+$ for each coating as follows:

\[ \Delta U^* = \frac{2}{\sqrt{\nu}} - \frac{2}{\sqrt{\nu} \epsilon} \]

\[ k^+ = \frac{1}{\sqrt{\nu}} \frac{\Delta U^*}{\sqrt{\nu} \epsilon} \]

where, $\epsilon_f$ and $\epsilon_f$ are the skin friction factors measured in the smooth and rough pipes, respectively, at the same value of $Re_M \epsilon_f$. Furthermore, the hydraulic diameter, $D_h$ of the channel was calculated by Eq (3).

It is of note that the selection of the roughness length scale, $k$, is critical to define a roughness function model, although $k$ only affects the roughness Reynolds number, $k^+$.

Therefore, $k$ can be selected so that the roughness function models obtained are in agreement with the Nikuradse (Cebeci and Bradshaw, 1977; Nikuradse, 1933) or Colebrook type (Colebrook et al., 1939), provided that the...
observed behaviours are still deemed appropriate relative to each other. Accordingly, the $k$ values were selected (Table 6) to get a good agreement between the present roughness functions and the Nikuradse type reference roughness function model.

2.4. CFD simulations

The present simulations were developed in the Star-CCM+ software package (Version 15.06.007-R8), adopting the Unsteady Reynolds Averaged Navier–Stokes (URANS)-based CFD with the modified wall-function model recently validated by Song et al. (Song et al., 2020c). The governing equations of the present CFD simulations are as in (Ferziger et al., 2020). Furthermore, the $k$-$\omega$ SST (Shear Stress Transport) turbulence model was used with a second-order convection scheme and the Volume of Fluid (VOF) model with Eulerian multiphase was used to simulate surface gravity waves on the interface between air and water. Finally, the free surface effects were modelled using High-Resolution Interface Capturing (HRIC). It is of note that the rationale behind the present CFD modelling choices can be found in (Ravenna et al., 2022).

2.4.1. Geometry and Physical Settings

CFD simulations were carried out on the container ship KCS in full-scale, at a towing speed of 24 knots (12.35 m/s), Froude number $Fn = 0.26$. The Reynolds number based on the ship speed and length was in the range of $Re_L = 2.72 \times 10^6$, which corresponds to the design speed of the full-scale KCS hull. Table 4 presents the particulars of the full-scale and model KCS adapted from Kim et al. (Kim et al., 2001) and (Larsson et al., 2013).

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scale factor</td>
<td>1</td>
</tr>
<tr>
<td>Length between the perpendiculars</td>
<td>$L_{rel}$ (m)</td>
</tr>
<tr>
<td>Length of waterline</td>
<td>$L_{w}$ (m)</td>
</tr>
<tr>
<td>Beam at waterline</td>
<td>$B_{w}$ (m)</td>
</tr>
<tr>
<td>Depth</td>
<td>$D$ (m)</td>
</tr>
<tr>
<td>Design draft</td>
<td>$T$ (m)</td>
</tr>
<tr>
<td>Wetted surface w/o rudder</td>
<td>$W/S_{w} \text{total}$ (m$^2$)</td>
</tr>
<tr>
<td>Displacement</td>
<td>$V$ (m$^3$)</td>
</tr>
<tr>
<td>Block coefficient</td>
<td>$C_s$</td>
</tr>
<tr>
<td>Design speed</td>
<td>$V$ (km; m/s)</td>
</tr>
<tr>
<td>Froude number</td>
<td>$F_n$</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>$Re$</td>
</tr>
<tr>
<td>Centre of gravity</td>
<td>$K_G$ (m)</td>
</tr>
<tr>
<td>Metacentric height</td>
<td>$GM$ (m)</td>
</tr>
</tbody>
</table>

The computational domain of the present simulations is a virtual towing tank (Figure 5), and the size of the domain was chosen following the International Towing Tank Committee (ITTC) recommendations (ITTC, 2011b) and similar studies (Song et al., 2021a, 2021b, 2020b). For clean hull case, the smooth type of wall-function was used, whereas the rough type of wall-functions, containing the roughness functions of the test surfaces, were used for the rough surfaces of the hull. Finally, the model ship was free to sink and trim, as no constraints were given. Figure 6 shows the volume mesh of the present CFD analysis. The built-in automated mesher of Star-CCM+ software was used to generate the trimmed hexahedral-dominant finite element mesh. Further near-wall mesh refinements were applied using prism layer meshes on the critical regions such as the free surface, the bulbous bow, and the stern. It is of note that for the present simulations, the wall $y^+$ values were kept between 30 and 300 and higher than $k^+$ values, as recommended by (Siemens, 2020), Figure 7. Furthermore, the average wall $y^+$ value is 190 and the number of cells is in the range of 1.4 million, and these values are in close agreement with (Dogrul et al., 2020). Finally, all the simulations used the same mesh regardless of the hull roughness scenarios.

![Figure 5: Computational domain and boundary conditions of the full-scale KCS simulations.](image1)

![Figure 6: Volume mesh used for the KCS full-scale simulations.](image2)

![Figure 7: Non-dimensional wall distance $y^+$ of the full-scale KCS with homogenous hull roughness (Sand 60-80) towed at 24 knots ($Fn = 0.26$).](image3)

2.4.2. Modified Wall Function Approach

Eq (9) shows the roughness function model employed in the CFD software to represent the roughness conditions examined and obtain the variance in frictional resistance coefficients.

$$\Delta U' = \left\{ \begin{array}{ll} \frac{A}{k_s} & \text{for } k^+ < k_{smooth}^+ \\ \frac{1}{k_s} \ln C_s k^+ & \text{for } k_{smooth}^+ \leq k^+ < k_{rough}^+ \\ \frac{1}{k_s} \ln C_s k_{rough}^+ & \text{for } k_{rough}^+ \leq k^+ \end{array} \right.$$  \hspace{1cm} (9)

(Cebeci and Bradshaw, 1977) recommended the following constants: $k_{smooth}^+ = 2.25$, $k_{rough}^+ = 90$, $A = 0$ and $C_s = 0.253$ for traditional Nikuradse roughness function and $C_s = 0.5$ for other roughness types. (Demirel et al., 2017) proposed $k_{smooth}^+ = 3$, $k_{rough}^+ = 15$ and $C_s = 0.26$ when
fitting the roughness function proposed by (Schultz and Flack, 2007). In the results section of the present study (Section 3), different constants to develop the roughness function models for the surfaces tested will be introduced.

2.4.3. Verification and Validation

The verification procedure of the present CFD study was carried out to assess the spatial uncertainty of the simulations. Richardson’s Grid Convergence Index (GCI) method (Richardson, 1911) was adopted as below. According to (Celik et al., 2008), the final expression for the fine-grid convergence index is defined as in equation (10):

\[ GCI_{fin} = \frac{1.25e^{2}_{21}}{e_{21}^{ext} - 1} \] (10)

where, \( e_{a}^{21} \) is the approximate relative error of the key variables, \( \phi_{a} \), obtained by equation (11), i.e., total resistance coefficient, \( C_{f} \), as in equation (12):

\[ e_{a}^{21} = \frac{\phi_{1} - \phi_{2}}{\phi_{1}} \] (11)

\( r_{21} \) is the refinement factor given by \( r_{21} = \sqrt[2]{N_{1}/N_{2}} \), where \( N_{1} \) and \( N_{2} \) are the fine and medium cell numbers, respectively. Also, the apparent order of the method, \( p_{a} \), is determined by solving equations (12) and (13) iteratively:

\[ p_{a} = \frac{1}{\ln(r_{21})}\left[\ln\left(\frac{e_{21}}{e_{21}^{ext}}\right) + q(p_{a})\right] \] (12)

\[ q(p_{a}) = \ln\left(\frac{r_{21} - s}{r_{21} - 1}\right) \] (13)

where \( s = sign\left(\frac{e_{21}}{e_{21}^{ext}}\right) \), \( e_{21} = \phi_{1} - \phi_{2} \), \( e_{21}^{ext} = \phi_{1} - \phi_{3} \) and \( r_{32} \) is the refinement factor given by \( r_{32} = \sqrt[3]{N_{3}/N_{2}} \), where \( N_{3} \) is the coarse cell number.

The extrapolated value of the key variables is calculated by equation (14):

\[ \phi_{21}^{ext} = \frac{r_{21} \phi_{1} - \phi_{3}}{r_{21} - 1} \] (14)

The extrapolated relative error, \( e_{21}^{ext} \), is obtained by equation (15):

\[ e_{21}^{ext} = \frac{\phi_{21}^{ext} - \phi_{1}}{\phi_{21}^{ext}} \] (15)

Table 5: Parameters used for the discretisation error for the spatial convergence study, key variable: \( C_{f} \).

<table>
<thead>
<tr>
<th>Full-scale KCS simulation</th>
<th>( N_{1} )</th>
<th>( N_{2} )</th>
<th>( r_{21} )</th>
<th>( \phi_{1} )</th>
<th>( \phi_{2} )</th>
<th>( \phi_{3} )</th>
<th>( e_{21}^{ext} )</th>
<th>( e_{a}^{21} )</th>
<th>( q )</th>
</tr>
</thead>
<tbody>
<tr>
<td>729,083</td>
<td>1,413,800</td>
<td>2,307,581</td>
<td>1.17</td>
<td>1.25</td>
<td>1.988 · 10^{-2}</td>
<td>1.966 · 10^{-3}</td>
<td>3.07 · 10^{-5}</td>
<td>7.18 · 10^{-6}</td>
<td>0.36%</td>
</tr>
<tr>
<td>7,004</td>
<td>985.1 · 10^{-2}</td>
<td>0.17%</td>
<td>0.53%</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 5 depicts the required parameters for the calculation of the spatial uncertainty of the simulation. A grid convergence index, \( GCI_{line} \), of 0.53% was estimated for the fine-grid simulations conducted in the smooth surface condition with the inlet speed of 24 km (\( R_{m} = 2.72 \cdot 10^{5} \)), when using ten iterations every time step of 0.1 s. It is of note that the time step was selected following the recommendations of (ITTC, 2011b), for which \( \Delta t = 0.005 - 0.01 L_{pl}/V \), where \( L_{pl} \) is the ship length at waterline and \( V \) is the ship speed. In comparison to the simulations in (Song et al., 2020c), the number of cells in the present study is considerably less, guaranteeing a reduced computational cost without compromising the accuracy of the results. In fact, the estimated GCI value of 0.53% indicates the great accuracy of the present CFD resistance prediction. Furthermore, the resistance coefficient results of the smooth case agree with the results found in the literature. In fact, the discretisation errors for the spatial convergence study, GCI, found by (Dogrul et al., 2020; Song et al., 2020b) for the KCS model scale hull were 0.40% and 0.10%, respectively. Therefore, the present CFD simulations to predict the effect of hull roughness on ship resistance and powering are further validated.

2.5. Experimental Uncertainty Analysis

The uncertainties of the measurements in the CTF tests were assessed following the ITTC-recommended procedures (ITTC, 2014). The standard errors for the coefficient of friction were calculated based on four to six replicate runs of the \( FR01 \) panel at the minimum and maximum flow velocities, respectively. The precision uncertainty in the skin friction coefficient, \( c_{f} \), was calculated at a 95% confidence interval by multiplying the standard error by the two-tailed t values (\( t = 3.182, 2.571 \)) for three to five degrees of freedom, according to (Coleman and Steele, 2012).

Notably, the accuracy of the differential pressure sensor is ±0.075%, and the accuracy of the magnetic flow meter was ±0.2%, according to the manufacturer’s specifications. The total uncertainty in the roughness function (\( \Delta U^{+} \)) was ±14.4% or 0.04 (whichever was larger) at the lowest \( Re_{M} \) ±6.5% or 0.04 (whichever was larger) at the highest \( Re_{M} \). For comparison, the high Reynolds number turbulent flow facility at the US Naval Academy achieved a relatively similar level of uncertainty, with their skin friction data being ±1.2% at \( Re_{M} \) between 4.0 · 10^{4} - 3.0 · 10^{5} (Schultz et al., 2015). The total bias limit and precision limit for the skin friction coefficients (\( c_{f} \)) were combined to give a total uncertainty of ±0.74% at the lowest \( Re_{M} \) and ±0.47% at the highest \( Re_{M} \).

3 RESULTS AND DISCUSSION

3.1. Fully Turbulent Flow Channel Experiments

3.1.1. Roughness Function Models

As discussed in the methodology section (Section 2), provided that the roughness functions of the test surfaces are known, the CFD simulations can be used to predict the effect of hull roughness on ship resistance. Once the roughness functions have been calculated, they were directly compared with both Colebrook-type (Grigson,
1992) and Nikuradse-type (Cebeci and Bradshaw, 1977) roughness functions. Furthermore, the roughness functions of the sandpaper-like surfaces were compared for validation purposes with results obtained from other studies. In fact, previous flat plate towing tank experiments conducted for the same surface roughness (Sand 220) were used for comparison to the present results, Figure 8, (Ravenna, 2019). Finally, the new roughness functions have been developed using STAR-CCM+’s built-in features, as in Eq (9).

Figure 8: Experimental roughness function of the Sand 220 surface developed from FTFC pressure drop measurements and from towing tank tests in (Ravenna, 2019).

Figure 9: Experimental roughness functions of the sanded rough test surfaces (Sand 220 and Sand 60-80) developed from FTFC pressure drop measurements.

Figure 10: Experimental roughness functions of the sanded rough test surfaces (FCCs) developed from FTFC pressure drop measurements.

Figure 9, and Figure 10 show the experimental roughness functions, $\Delta U^+$, vs roughness Reynolds numbers, $k^+$ obtained from the FTFC pressure drop measurements following Granville’s approach (Granville, 1987). It is of note that the experimental roughness functions of the FCCs tested were modelled by curve fitting to the roughness function model of Nikuradse. For completeness, in Table 6 are presented the curve fitting coefficients used for all the surfaces tested, where $E$ is the so-called turbulent wall function coefficient. In fact, in StarCCM+, the wall roughness is modelled by moving the logarithmic region of the boundary layer closer to the wall. To decrease roughness, $E$ must be increased to incorporate this effect. Therefore, for the smoother and best performing surfaces (AF01 and FR02) to which corresponded negative roughness function values, $E$ was increased from the standard $E = 9$ to $E = 12$ and $E = 15$, respectively.

Table 6: Curve fitting coefficients of the roughness functions for the test surfaces.

<table>
<thead>
<tr>
<th>Test Surface</th>
<th>Roughness length scale, $k$ [m]</th>
<th>$A$</th>
<th>$C_s$</th>
<th>$E$</th>
<th>$k^+$</th>
<th>$k'_+$</th>
</tr>
</thead>
<tbody>
<tr>
<td>AF01</td>
<td>9.598 $\cdot 10^{-6}$</td>
<td>-1.5</td>
<td>0.2</td>
<td>12</td>
<td>1</td>
<td>15</td>
</tr>
<tr>
<td>BL01</td>
<td>1.822 $\cdot 10^{-5}$</td>
<td>-0.5</td>
<td>0.26</td>
<td>9</td>
<td>2</td>
<td>25</td>
</tr>
<tr>
<td>FR01</td>
<td>1.544 $\cdot 10^{-5}$</td>
<td>-0.5</td>
<td>0.2</td>
<td>15</td>
<td>3</td>
<td>25</td>
</tr>
<tr>
<td>FR02</td>
<td>5.840 $\cdot 10^{-4}$</td>
<td>-1.5</td>
<td>0.26</td>
<td>9</td>
<td>2</td>
<td>15</td>
</tr>
<tr>
<td>Sand 220</td>
<td>1.532 $\cdot 10^{-4}$</td>
<td>0</td>
<td>0.35</td>
<td>9</td>
<td>3</td>
<td>25</td>
</tr>
<tr>
<td>Sand 60-80</td>
<td>3.530 $\cdot 10^{-4}$</td>
<td>0</td>
<td>0.49</td>
<td>9</td>
<td>3</td>
<td>25</td>
</tr>
</tbody>
</table>

3.2. Numerical Prediction on full-scale KCS hull

3.2.1. Ship Resistance Coefficients

Numerical predictions were conducted on the benchmark KRISO containership hull at a towing speed of 24 knots ($Fn = 0.26$). The variance of resistance and powering requirements due to different test surfaces were calculated by incorporating the newly developed roughness functions into the Granville similarity law. The total resistance coefficient, $C_T$, is defined in equation (16) as a function of the total drag, $D_T$, the dynamic pressure, $1/2 \rho V^2$, and the hull wetted surface area, $S$:

$$C_T = \frac{D_T}{1/2 \rho V^2 \cdot S}$$  \hspace{1cm} (16)$$

where, $V$ is the towing speed (i.e., the inlet velocity). It is well-known that the total ship resistance coefficient, $C_T$, can be decomposed into the frictional, $C_F$, and the residuary, $C_R$, resistance coefficients, as given by Eq (17):

$$C_T = C_F + C_R$$  \hspace{1cm} (17)$$

The variation of the frictional resistance coefficient $\Delta C_F$ is the difference between the rough, $C_{F, rough}$, and smooth, $C_{F, smooth}$, conditions at the same Froude number can be given by Eq (18):

$$\Delta C_F = C_{F, rough} - C_{F, smooth}$$  \hspace{1cm} (18)$$

Hence, the variation of the frictional resistance due to the presence of roughness can also be expressed in percentage, as in equation (19):

$$\% \Delta C_F = \frac{C_{F, rough} - C_{F, smooth}}{C_{F, smooth}} \cdot 100$$  \hspace{1cm} (19)$$

The total resistance for the rough ship, $C_{T, rough}$, is determined by:

$$C_{T, rough} = C_{T, smooth} + \Delta C_T$$  \hspace{1cm} (20)$$
\[ \Delta C_T = C_{T\text{rough}} - C_{T\text{smooth}} \]  

(21)

Figure 11 presents the resistance coefficients of the test cases obtained from the CFD simulations analysis compared to a hydrodynamically smooth ship hull. It is notable that the total resistance coefficient results are in agreement with other studies found in the literature such as Yeginbayeva and Atlar, 2018). Interestingly, the test cases \( AF01 \) and \( FR02 \) show a negative \( \Delta C_T \) of 2.1% and 3.6%, respectively. As expected, the phenomena of reduced \( \Delta C_T \) values are due to the negative roughness functions, \( \Delta U^+ \) observed from the experimental measurements. On the other hand, the \( BL01 \) and \( FR01 \) cases lead to light \( \Delta C_T \) increases (0.9% for \( BL01 \) and 0.2% for \( FR01 \)) compared to the total added resistance due to mimicked slime (27.7% for \( Sand \ 220 \) and 36.1% \( Sand \ 60-80 \) cases). Above all, it can be noted that the \( FR02 \) is the best performing FCCs tested while the sanded surface, \( Sand \ 60-80 \), leads to a higher increase in the total resistance coefficients.

![Frictional and total resistance coefficients variation in different hull roughness conditions.](image)

Table 7 shows the total resistance coefficients obtained for the test surfaces. It is also notable that the total resistance coefficient results are in agreement with other studies found in the literature such as (Schultz, 2004; Yeginbayeva and Atlar, 2018). Furthermore, the results are reasonably in agreement with other studies found in the literature such as (Schultz, 2004; Yeginbayeva and Atlar, 2018). In fact, (Schultz, 2004; Yeginbayeva and Atlar, 2018) found that a foul-release coating as applied measured an added frictional resistance \( \% \Delta C_T \) equal to 2.6%, and for a 150 m flat plate at 12 knots coated with sand 60-80 calculated \( \% \Delta C_T \) was 59%.

Table 7: Frictional resistance results \( (C_F) \) results on the full-scale KCS hull at 24 knots \( (Fn = 0.26) \).

<table>
<thead>
<tr>
<th>Test Surface</th>
<th>( C_F )</th>
<th>CFD simulations</th>
<th>( % \Delta C_T )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference</td>
<td>1.309 · 10(^{-3})</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>AF01</td>
<td>1.268 · 10(^{-3})</td>
<td>-4.05 · 10(^{-5})</td>
<td>-3.09%</td>
</tr>
<tr>
<td>BL01</td>
<td>1.326 · 10(^{-3})</td>
<td>1.91 · 10(^{-5})</td>
<td>1.46%</td>
</tr>
<tr>
<td>FR01</td>
<td>1.314 · 10(^{-3})</td>
<td>5.44 · 10(^{-6})</td>
<td>0.42%</td>
</tr>
<tr>
<td>FR02</td>
<td>1.238 · 10(^{-3})</td>
<td>-7.08 · 10(^{-5})</td>
<td>-5.41%</td>
</tr>
<tr>
<td>Sand 220</td>
<td>1.835 · 10(^{-3})</td>
<td>5.27 · 10(^{-4})</td>
<td>40.26%</td>
</tr>
</tbody>
</table>

4 CONCLUSIONS AND FUTURE WORK

An experimental and CFD study was carried out to investigate the full ship hydrodynamic performance of different fouling control coatings and mimicked biofouling. The experimental part of the study led to the...
introduction of novel experimental roughness functions for the FCCs tested including GIT’s novel hard foul-release coating (FR02), while the numerical part scaled up the laboratory results to the size of a full ship length. The experimental roughness functions of the test surfaces were developed based on the pressure drop measurements conducted with the Fully Turbulent Flow Channel (FTFC) facility of the University of Strathclyde. The newly developed roughness functions of the fouling control coatings and sanded surfaces were implemented into the modified wall function approach in CFD using the Star-CCM+ software to provide scale-up results to ship length. The benchmark KRISO containership (KCS) hull in full-scale was chosen to calculate the variance of resistance and powering requirements due to different test surfaces at the design speed of 24 knots ($F_n = 0.26, Re = 2.72 \times 10^6$).

Among the four fouling control coatings (FCCs) that were tested in the FTFC, the FR02 coating (hard foul-release) displayed the best hydrodynamic performance across the entire Reynolds number range. In fact, FR02 displayed lower frictional resistance coefficients than if the ship was considered as smooth as the acrylic reference panel (5.57% decrease). Furthermore, FR02 led to a maximum decrease in effective power requirements of 3.6%. The results of the numerical prediction also show that the AF01 (self-polishing antifouling coating) have better hydrodynamic performance than the smooth reference case (maximum decrease in effective power requirements of 2.1%). In contrast, Sand 220 (medium light slime) and Sand 60-80 (medium slime) have, as expected, the highest resistance due to their rougher characteristics. In fact, a ship hull with medium light slime (Sand 220) and medium slime (Sand 60-80) surface roughness characteristics as the test surfaces would experience a maximum increase in effective power requirements of 26.7% and 36.1%, respectively.

Further investigation could be conducted on the prediction of resistance of the coatings at different speeds, on different hulls, and using heterogeneous patch distribution of the roughness. It will also be beneficial to investigate the hydrodynamic performance of the same fouling control coating under the effect of biofouling growth. Exposing surfaces to dynamically grown biofouling will give shipowners and operators a better indication of what powering penalty they should expect from these coatings after a certain time in active service. Finally, applying different mimicked biofouling to the panels before or after the coating application could also serve as a better method to predict the resistance behaviour of the as-applied condition to an existing rough ship hull.

Above all, the present study has provided several important findings, including the procedure to conduct pressure drop measurements with a FTFC, the application of Granville's method for pipes to develop roughness functions, as well as the introduction of roughness functions for novel or widely adopted marine surfaces and mimicked biofouling. The findings presented can help predict the required power, fuel consumption and greenhouse gas emissions of ships with hulls coated with certain fouling control coatings and/or in the fouled condition. As a final remark, the authors would like to emphasise that there is an enormous opportunity for growth around research on FTFCs. Indeed, the present study only represents an infinitesimal fraction of the number of coating products and surface roughness conditions that can be tested.

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ITTC, 2014. Executive Committee Final report and recommendations to the 27th ITTC.


