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Effect of ship hull form on the resistance penalty from biofouling

Dinis Oliveira, Ann I. Larsson and Lena Granhag

ABSTRACT

Hull biofouling is a well-known problem for the shipping industry, leading to increased resistance and fuel consumption. Considering that the effects of hull form on resistance are known to be higher for a less slender hull, it is hypothesised in this paper that the effect of biofouling roughness on resistance is also dependent on the hull form. To test this hypothesis, previously reported full-scale numerical results on a containership are re-analysed. Form effects on roughness penalties, corresponding to $K_{\Delta C_T} = 0.058 \pm 0.025$, are observed at a low speed (19 knots, $Re_c = 2.29 \times 10^5$), which are however cancelled out by traditionally neglected roughness effects on wave-making resistance at a higher speed (24 knots, $Re_c = 2.89 \times 10^5$). It is concluded that hull form effects on biofouling penalties can be significant at low speeds, though not generalisable for higher speeds, namely when wave-making resistance corresponds to $\geq 29\%$ of total resistance.

Introduction

In today’s globalised world, about 90% of world trade relies directly on commercial shipping (ICS 2017). Sea transport is at present dependent on fossil fuels as energy sources and contributes to $\sim 3\%$ of global CO$_2$ emissions (IMO 2014), thus raising economic, environmental and societal issues.

A well-recognised way to avoid fuel penalties, and at the same time reduce emissions of greenhouse gases and air pollutants, is by making the hull as smooth as possible, and retaining smoothness during vessel operation. The ideal situation would be a hydraulically smooth hull. The required roughness height $k_t$ for a smooth hull is however prohibitively low, usually $< 20 \mu m$, which is much lower than typical out-of-dock roughness of $\sim 150 \mu m$ (Lackenby 1962; Johansson 1984; Schultz 2007). Therefore, ship operators must find a balance between voyage costs, ie increase in fuel costs or reduced speed, and maintenance costs, ie hull conditioning, cleaning and repainting. In order to achieve this, a practical method to accurately determine increased resistance due to hull roughness is required.

Currently, the most common methods available for determining increased propulsion power due to hull roughness include sea trials and voyage performance monitoring (Andersen et al. 2005; Munk et al. 2009; ISO 2016). However, sea trials are only performed sporadically, due to time and environmental constraints, whereas voyage performance data suffer from high uncertainty due to changes in draft and trim, weather conditions (wind and waves), sea currents, temperature and salinity (Munk et al. 2009). Thus, most of the previous studies on hull penalties have relied on laboratory-scale experimental results and extrapolation methods to the full-scale ship (Leer-Andersen and Larsson 2003; Schultz et al. 2011; Monty et al. 2016).

In this paper, it is hypothesised that the hull form can affect the prediction of full-scale penalties based on laboratory-scale friction results, since results from the laboratory are often scaled up to the full-scale ship using flat plate frictional resistance. Previously published computational fluid dynamics (CFD) results for a container-ship (Demirel et al. 2017) are re-analysed here in order to determine the extent to which the form of the hull can affect the penalty due to hull roughness. The theoretical background is developed in the next sub-section.

Roughness effects at laboratory and full scale

Hull roughness is caused by mechanical processes on the hull surface (eg sand-blasting treatment on steel prior to painting, mechanical damage to the coating, and
occurrence of corrosion) and it is significantly increased by the occurrence of biofouling, ie the growth of marine organisms on the hull surface (Schultz 2007).

From a smooth to a hydraulically rough surface, frictional resistance, ie the tangential force caused by viscous flow over the surface, increases due to protrusion of roughness elements above the laminar viscous sub-layer (Larsson and Raven 2010a). This results in a downward shift in velocities measured at a range of distances from the hull (Lewthwaite et al. 1984). Thus, the following equation applies for the velocity profile within the so-called logarithmic region of the boundary layer (Walker 2014):

\[ u^+ = \frac{1}{k^+} \ln \left[ (y + \epsilon)^+ \right] + C - \Delta U^+ \tag{1} \]

where velocity \( u^+ \), the component tangential to the hull, made dimensionless by dividing by the wall shear velocity \( u_\tau \) \( (u^+ = u/u_\tau) \), is given as a function of the distance from the hull \( y \), with wall origin error \( \epsilon \), also given in viscous units: \( (y + \epsilon)^+ = (y + \epsilon) u_\tau / u \). Both the origin error \( \epsilon \) and the roughness function \( \Delta U^+ \) are variables dependent on the roughness of the wall.

Determination of the wall shear velocity, \( u^+ = (\tau_w / \rho)^{1/2} \), is a fundamental step for determining the effects of roughness on frictional drag in wall-bounded turbulent flows. The value of \( u^+ \) is used in scaling of the velocity profile, as presented in Equation 1. The roughness function \( \Delta U^+ \) (Equation 1) then corresponds to a downward shift in velocity profile in the presence of a rough wall: \( \Delta U^+ \) is zero for a hydraulically smooth surface and > 0 for transitionally and fully rough surfaces. The roughness function \( \Delta U^+ \) has been determined for different types of roughness (Figure 1), including uniform sand roughness (Nikuradse 1933; Cebeci and Bradshaw 1977) and antifouling coatings (Johansson 1984; Schultz 2004; Candries and Atlar 2005). As shown in Figure 1, \( \Delta U^+ \) can be collapsed for disparate roughness geometries in the fully rough regime (ie high Reynolds number) by use of the so-called equivalent sand-roughness height \( k^+ \). However, there is no universal curve for the transitionally rough regime, ie a curve applicable to all roughness geometries at moderate values of Reynolds number \( k^+ \) (Bradshaw 2000). Therefore, \( \Delta U^+ \) must be experimentally determined for each new type of roughness (Grigson 1987). Using \( \Delta U^+ \) curves (Figure 1), extrapolation methods enable determination of ship-scale increased resistance for a given hull roughness condition (Schultz 2007). Alternatively, \( \Delta U^+ \) can be used as an input to CFD codes for calculating ship resistance (Leer-Andersen and Larsson 2003; Demirel et al. 2014, 2017).

The effect of roughness on ship resistance has been studied extensively within the past 100 years. Increased frictional resistance is usually pinpointed as the primary effect, corresponding to increased tangential forces acting on the hull (Woods Hole Oceanographic Institute 1952; Schultz 2007). When analysing the components of ship resistance \( R_T \), frictional resistance \( R_F \) is separated from pressure resistance \( R_P \), each representing the area integral of tangential and normal components of force, respectively, projected in the direction opposing the ship’s movement:

\[ R_T = R_F + R_P \] (2)

or, using dimensionless coefficients:

\[ \frac{R_F}{1/2 \rho U^2 S} = \frac{R_F}{1/2 \rho U^2 S} + \frac{R_P}{1/2 \rho U^2 S} \equiv C_T = C_F + C_P \tag{3} \]

where \( \rho \) is fluid density, \( U \) is the ship speed and \( S \) is the wetted surface area of the hull.

Alternatively, \( C_T \) can be decomposed into equivalent flat plate frictional resistance \( C_{F0} \) and residuary resistance \( C_R \):

\[ C_T = C_{F0} + C_R \tag{4} \]

Equation 4 embodies the decomposition of ship resistance as initially proposed by William Froude in 1872–1874 (Larsson and Raven 2010b). Frictional resistance \( C_{F0} \) is traditionally expressed as a function of inertia and viscosity, ie what is known today as the Reynolds number \( Re \), representing purely frictional resistance of an equivalent flat plate, where ‘equivalent’ means a flat plate with the same waterline length and wetted surface area as the ship or ship’s model. The residuary component \( C_R \) is expressed as a function of inertia and gravitational forces, ie Froude number \( Fr \), as it is assumed to be predominantly composed of wave-making resistance. However, \( C_R \) also

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**Figure 1.** Two examples of roughness function \( \Delta U^+ \): Nikuradse’s uniform sand roughness function (Nikuradse 1933; Cebeci and Bradshaw 1977) and Colebrook-type roughness function (Johansson 1984).
contains Re-dependent viscous components, so a third way of decomposing ship resistance was called for, separating wave-making resistance $C_W$ from the viscous component $C_V$ (Larsson and Raven 2010b):

$$C_T = C_V(Re) + C_W(Fr).$$  \hspace{1cm} (5)

Wave-making is assumed to be exclusively dependent on inertia and gravitational forces (Froude number), meaning that $C_W$ can be assumed to have the same value at both model scale and full ship scale, provided that $Fr_m = Fr$, and that William Froude's assumption on similarity of wave patterns holds (Larsson and Raven 2010c).

The viscous resistance coefficient $C_V$ can be further expressed as the sum of flat plate resistance $C_{FP}$, form effect on friction $C_{FF}$, and form effect on pressure $C_{VP}$ (Larsson and Raven 2010b):

$$C_T = (C_{F_0} + C_{FF} + C_{VP}) + C_W$$  \hspace{1cm} (6)

Form effects $C_{FF}$ and $C_{VP}$ arise from differences between the flat plate friction line $C_{F_0}$ and the hull's viscous resistance $C_V$, due to two simultaneous mechanisms: (1) increased friction on the hull (tangential force), denoted by $C_{FF}$, arises from the displacement of streamlines around the hull, resulting in increased speed near the hull as compared to the flow around the flat plate, and (2) increased pressure resistance on the hull (normal force), denoted by $C_{VP}$ and referred to as viscous pressure, arises from the boundary layer development along the length of the hull, which causes streamline displacement from the hull in the aft body, thus decreasing the pressure at the aft body and leading to increased resistance (Lackenby 1962; Larsson and Raven 2010b).

In practice, form effects can be obtained from running a model in the towing tank at sufficiently low speed, since $C_W \rightarrow 0$ as $Fr \rightarrow 0$ (Larsson and Raven 2010c), or running CFD simulations without a free-surface ($C_W = 0$), also known as single-phase or double-body simulations (Kouh et al. 2009). Form factors are then defined as (Nagamatsu 1980):

$$K_F = \frac{C_F}{C_{F_0}} - 1$$  \hspace{1cm} (7)

$$K_p = \frac{C_{VP}}{C_{F_0}}$$  \hspace{1cm} (8)

where $C_F$ is the hull friction coefficient, as in Equation 3. Ship resistance can be finally expressed as a function of the form factors $K_F$ and $K_p$, flat plate friction $C_{F_0}$ and wave-making resistance $C_W$:

$$C_T = (1 + K_F + K_p) C_{F_0} + C_W$$  \hspace{1cm} (9)

or, using a single form factor $K = K_F + K_p$:

$$C_T = (1 + K) C_{F_0} + C_W$$  \hspace{1cm} (10)

Form factors are Reynolds-dependent and scale dependent (Garcia-Gomez 2000; Kouh et al. 2009), as will be further discussed in this paper.

The effects of roughness at ship scale can be expressed in terms of increased resistance $\Delta C_T$, representing the change in force required to tow the ship. Schultz (2007) uses Granville’s similarity law scaling, which makes use of a roughness function $\Delta U^+$ (as in Equation 1) in order to determine the change in flat plate frictional resistance $\Delta C_{FP}$. The final formula for total ship resistance is given as (Schultz 2007; modified after Gillmer and Johnson 1982):

$$C_{T,s} = C_{F_{0,s}} + C_R + \Delta C_{F_{0,s}} + C_A$$  \hspace{1cm} (11)

where $C_R$ includes wave-making resistance and other forces acting normal to the hull, as derived directly from model-scale tests at $Fr_m = Fr$. For a smooth hull, the frictional coefficient $C_{F_{0,s}}$ is obtained from friction lines, eg ITTC-57 friction line (ITTC 2011). The ITTC-57 friction line for $C_{F_{0,s}}$ is a not a ‘pure’ flat plate friction line, since it already includes ~12% form factor (Larsson and Raven 2010c). Finally, a constant correlation allowance, $C_A$, accounts for form effects not included in the previous coefficients, among other corrections required for the scaling-up of model results obtained in a towing tank (Gillmer and Johnson 1982; Woo et al. 1983).

The decomposition of ship resistance in Equation 11 (Schultz 2007; modified after Gillmer and Johnson 1982) differs somewhat from the ITTC – International Towing Tank Conference 1978 procedure, revised in 2014 (ITTC 2014), here deliberately excluding the air resistance coefficient (only hydrodynamic resistance is considered):

$$C_{T,s} = (1 + K) C_{F_{0,s}} + C_W + (\Delta C_{T,ITTC} + C_{A,ITTC})$$  \hspace{1cm} (12)

where $C_W$ is substituted for $C_{F_0}$, the form factor $K$ is introduced, and a different method is used for determining the hull roughness allowance $\Delta C_{T,ITTC}$, in this case the Townsin’s formula (ITTC 2014):

$$\Delta C_{T,ITTC} = 0.044 \left( k \frac{U}{L} \right)^{3/2} - 10 Re^{-0.5} + 0.000125$$  \hspace{1cm} (13)

Additionally, the revised ITTC-78 procedure includes a Reynolds-dependent correlation allowance $C_{A,ITTC}$ (Equation 12), which results in a value of $(\Delta C_{T,ITTC} + C_{A,ITTC})$ approximate to that obtained using Bowden’s formula, as in the original 1978 procedure (ITTC 2014). It should be noted that Equation 13 is only valid for painted hulls with peak-to-valley roughness height $k_i < 230 \mu m$ (over a 50-mm length).
For biofouling, methods based on the hydrodynamic equivalent sand-roughness height, \( k_s \), have been proposed by several authors to give more accurate predictions of resistance penalty, provided that previous experimental data on biofouling roughness types (\( \Delta U' \)) are available. Such scaling-up methods include Granville’s similarity law scaling (Schultz 2007) and CFD approaches (Eça et al. 2010; Eça and Hoekstra 2011; Demirel et al. 2017). Today, where the Granville method takes about 1/10 of a second to compute on a laptop computer, viscous simulations using CFD can take up to several days to run, thus limiting the number of cases that can be evaluated using CFD. Still, the Granville method has the limitation of not taking hull form effects into account, as roughness penalties are calculated on a flat plate rather than the hull shape.

The hypothesis put forward in the current study is that cost-effective flat plate calculations, such as the Granville’s scaling method (Schultz 2007), could be made more accurate by multiplying roughness effects by the form factor \( (1+K) \). This hypothesis is equivalent to adding the roughness effect, \( \Delta C_{F0,s} \), directly to the flat plate friction line, \( C_{F0,s} \):

\[
C_{T,s} = (1 + K)(C_{F0,s} + \Delta C_{F0,s}) + C_W
\]  

(14)

Thus, assuming that the effect of hull roughness on the ship’s wave pattern is negligible (ie no effect on \( C_W \)), Equation 14 would mean a global roughness penalty on ship resistance given by:

\[
\Delta C_{T,s} = (1 + K)\Delta C_{F0,s}
\]  

(15)

In this paper, this hypothesis is analysed in light of recent developments on the validity of the form factor approach, and tested by re-analysing publicly available CFD results on the KRISO container ships hull (KCS hull) and the equivalent flat plate (Demirel et al. 2017), as further detailed below.

Materials and methods

Repository data

CFD results re-analysed in this study are credited to Demirel et al. (2017). Simulation results from Demirel et al. (2017) are publicly available on the University of Strathclyde data repository, corresponding to full-scale towing resistance solutions for the KCS hull, separated into frictional (\( C_p \)) and normal forces (\( C_n \)). The results were obtained by Demirel et al. (2017) using a finite volume solver for the unsteady RANS equations (Reynolds-averaged Navier–Stokes equations). Resistance results for the equivalent flat plate, composed only of the frictional component \( C_p \), were instead directly gathered from the paper (Demirel et al. 2017, tables 9 and 10), as they were not available from the repository. Simulated full-scale cases correspond to vessel speeds of 19 knots (\( Re_s = 2.29 \times 10^8 \)) and 24 knots (\( Re_s = 2.89 \times 10^8 \)). Other vessel specifications for the KCS hull are presented in Table 1. The full scale ship has never been built, but complete experimental model data, as well as model-scale simulations, are available for this hull geometry (Kim et al. 2001; NMRI 2017). For the KCS hull, wave-making resistance corresponds to 16% and 29% of the total resistance, at speeds of 19 and 24 knots, respectively.

In Demirel et al. (2017), the same turbulence model was used for both hull and flat plate, corresponding to the SST (shear stress transport) \( k-\omega \) turbulence model. It should be noted that modelling of the wall roughness requires a tailored near-wall mesh resolution for each hull fouling condition, as to obtain the correct non-dimensional distance between the first node and the wall, which should be higher than 30 viscous units and higher than \( k_s^+ \) (Eça and Hoekstra 2011; Demirel et al. 2014). More details on the grids used, as well as other numerical settings, can be found in the original study (Demirel et al. 2017).

Seven roughness hull conditions were tested by Demirel et al. (2017), representing both coated and fouled hull conditions that can be observed on ships’ hulls (Table 2). The selected \( k_s \) values were obtained from Schultz (2007). It should be borne in mind that these values might not be representative of the diversity of fouled hull conditions for each fouling rating (Schultz 2007). Finally, it is important to note that Demirel et al. (2017) used a modified roughness function (Demirel et al. 2014), and not the original Nikuradse’s roughness function for uniformly packed sand grains (Nikuradse 1933; Cebeci and Bradshaw 1977). The modified roughness function should better capture the transitionally rough behaviour of the flow over hull roughness (Demirel et al. 2014).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Waterline length, L</td>
<td>232.5 m</td>
</tr>
<tr>
<td>Breadth, B</td>
<td>32.2 m</td>
</tr>
<tr>
<td>Design draft, D</td>
<td>10.8 m</td>
</tr>
<tr>
<td>Block coefficient, Cb</td>
<td>0.6505</td>
</tr>
<tr>
<td>Wetted surface area, S</td>
<td>9,498 m²</td>
</tr>
<tr>
<td>Design speed, U</td>
<td>24 knots</td>
</tr>
<tr>
<td>Design Froude number, Fr</td>
<td>0.26</td>
</tr>
</tbody>
</table>

**Form factor and wave-making resistance**

Previous studies have reported dependency of the form factor \( K \) on Reynolds number (Garcia-Gomez 2000; Kouh et al. 2009). However, since in Demirel et al. (2017) the Reynolds number varies only within the full scale (\( Re_s = 2.29-2.89 \times 10^8 \)), present analyses rely on a
constant form factor of $K = 0.1$, as previously reported for the KCS hull (Castro et al. 2011; Demirel et al. 2017). A double-body simulation on the KCS hull geometry would enable the form factor $K$ to be determined more accurately for each speed, directly from the comparison between hull resistance without a free surface ($C_W = 0$) and numerical flat plate resistance (Raven et al. 2008; Kouh et al. 2009). However this was not conducted by Demirel et al. (2017).

From Equation 10, the KCS hull’s wave-making resistance coefficient can be obtained as:

$$C_W = C_T - (1 + K) C_{F0}$$

The above equation differs from Equation 15 of Demirel et al. (2017), in that the flat plate’s frictional resistance without a free surface ($C_{W0}$) and numerical flat plate resistance ($C_T$) (Kouh et al. 2009), were related to $C_{F0}$ by the frictional form factor $K_F$, ie $C_F = (1 + K_F) C_{F0}$ The current approach is therefore more consistent with the definition of $(1 + K)$ as the ratio between the hull’s viscous resistance, $C_V = C_F + C_{FP}$, and flat plate’s $C_{F0}$. This results in higher values of $C_W$, compared to the original values reported in Demirel et al. (2017).

### Comparison of penalty prediction methods

Finally, different penalty calculation methods were ranked according to their accuracy, using full-scale CFD results for the KCS hull as reference (Demirel et al. 2017). These methods include the CFD results for the equivalent flat plate from Demirel et al. (2017) and own calculations using the Granville method, as described in Schultz (2007) using the roughness function given by Demirel et al. (2017). Additionally, each of these results (the CFD flat plate, and the Granville method) are further multiplied by a form factor $(1 + K) = 1.1$ (Castro et al. 2011; Demirel et al. 2017), to determine whether such operation brings about any improvement in terms of accuracy.

### Uncertainty estimates

Numerical uncertainties in the CFD results were reported by Demirel et al. (2017) and the numerical method was verified using the grid convergence index method based on Richardson extrapolation. Discretisation errors corresponded to $\sim 1.6\%$ and $\sim 0.7\%$, for $C_F$ (flat plate) and $C_T$ (KCS hull), respectively. The discretisation errors for these coefficients, generically denoted $E_{C_X}$, are propagated in the current paper. Thus, the final error for $\Delta C_X$ is given by the following expression, where $X = T, F$ or $C0$ is used to denote total hull resistance, frictional hull resistance or frictional flat plate resistance (respectively):

$$E_{\Delta C_X} = \sqrt{(E_{C_X,\text{rough}})^2 + (E_{C_X,\text{smooth}})^2}$$

For form factors, which involve the ratio between resistance coefficients (eg Equations 7 and 8), propagated errors are given by the following equation:

$$E_{K_X} = K_X \sqrt{(\frac{E_{C_X}}{C_1})^2 + (\frac{E_{C_X}}{C_2})^2}$$

where $C_1$ and $C_2$ are the two resistance coefficients (or change in resistance coefficient) used in calculating a generic form factor $K_X$, where $X = F, P, \Delta C_F$ or $\Delta C_T$ is used to denote friction form factor, pressure form factor, form factor on change in frictional resistance or form factor on change in total resistance, respectively.

Additionally, Demirel et al. (2017) validated the numerical method against experimental model test data for the KCS hull (Kim et al. 2001) and the Granville method for the flat plate cases. The errors associated with resistance coefficients were proven to be within $2.3\%$.

### Results and discussion

#### Friction form factor, $K_F$

From the available KCS hull and flat plate CFD results, the friction form factor $K_F$ can be readily obtained using Equation 7, where $C_F$ is obtained from the KCS hull and $C_{F0}$ from the equivalent flat plate (Table 3). An approximately constant value of $K_F \sim 0.05$ is observed, with some dependency on $k_s$ (probably due to changes in the numerical grid for different roughness height, since the tailored near-wall mesh is not currently accounted for in the uncertainty estimates), and also with a general increase with speed. The later increase with speed is in agreement with the findings of Kouh et al. (2009), where a linear increase in $K$ with Reynolds number was observed.

The question that follows is whether a similar form effect on friction can be observed also on the change in frictional resistance due to roughness, ie a form factor on $\Delta C_F = (C_{F,\text{rough}} - C_{F,\text{smooth}})$. Results are presented in Table 4 for $\Delta C_F$ (KCS) and $\Delta C_{F0}$ (flat plate), for each rough hull condition ($k_s = 30–10,000 \mu m$), using the smooth case as a reference. Similar to the total friction (Table 3), the
Table 3. Friction and pressure form factors ($K_f$ and $K_p$, respectively) obtained from published CFD results on the KRISO containership hull (KCS) and equivalent flat plate (Demirel et al. 2017), assuming $K = K_f + K_p = 0.1$ (Castro et al. 2011). Uncertainties correspond to propagated numerical discretisation errors from the original simulations.

<table>
<thead>
<tr>
<th>$U$ [knots]</th>
<th>$k_s$ [μm]</th>
<th>$C_f \times 10^3$ (KCS)</th>
<th>$C_p \times 10^3$ (flat plate)</th>
<th>$K_f = C_f / C_p - 1$</th>
<th>$K_p = K - K_f$</th>
</tr>
</thead>
<tbody>
<tr>
<td>19</td>
<td>0</td>
<td>1.452 ± 0.011</td>
<td>1.386 ± 0.022</td>
<td>0.048 ± 0.001</td>
<td>0.052</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>1.560 ± 0.012</td>
<td>1.485 ± 0.023</td>
<td>0.050 ± 0.001</td>
<td>0.050</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>1.835 ± 0.014</td>
<td>1.750 ± 0.027</td>
<td>0.048 ± 0.001</td>
<td>0.052</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>2.115 ± 0.016</td>
<td>2.022 ± 0.032</td>
<td>0.046 ± 0.001</td>
<td>0.054</td>
</tr>
<tr>
<td></td>
<td>1,000</td>
<td>2.509 ± 0.019</td>
<td>2.401 ± 0.037</td>
<td>0.045 ± 0.001</td>
<td>0.055</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>3.007 ± 0.022</td>
<td>2.886 ± 0.045</td>
<td>0.042 ± 0.001</td>
<td>0.058</td>
</tr>
<tr>
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<td>10,000</td>
<td>3.734 ± 0.028</td>
<td>3.578 ± 0.056</td>
<td>0.044 ± 0.001</td>
<td>0.056</td>
</tr>
<tr>
<td>Average</td>
<td></td>
<td></td>
<td>0.046 ± 0.002</td>
<td></td>
<td>0.054</td>
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<tr>
<td>24</td>
<td>0</td>
<td>1.422 ± 0.011</td>
<td>1.351 ± 0.021</td>
<td>0.052 ± 0.001</td>
<td>0.048</td>
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<tr>
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<td>30</td>
<td>1.577 ± 0.012</td>
<td>1.496 ± 0.023</td>
<td>0.054 ± 0.001</td>
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<tr>
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<td>100</td>
<td>1.840 ± 0.014</td>
<td>1.730 ± 0.027</td>
<td>0.052 ± 0.001</td>
<td>0.048</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>2.121 ± 0.016</td>
<td>2.022 ± 0.032</td>
<td>0.049 ± 0.001</td>
<td>0.051</td>
</tr>
<tr>
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<td>0.048 ± 0.001</td>
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</tr>
<tr>
<td>Average</td>
<td></td>
<td></td>
<td>0.050 ± 0.002</td>
<td></td>
<td>0.050</td>
</tr>
</tbody>
</table>

Table 4. Form factor $K_{ACF}$ on the change in friction due to hull roughness, as obtained from published CFD results on the KRISO containership hull (KCS) and equivalent flat plate (Demirel et al. 2017). Uncertainties correspond to propagated numerical discretisation errors from the original simulations.

<table>
<thead>
<tr>
<th>$U$ [knots]</th>
<th>$k_s$ [μm]</th>
<th>$\Delta C_f \times 10^3$ (KCS)</th>
<th>$\Delta C_p \times 10^3$ (flat plate)</th>
<th>$K_{ACF} = \Delta C_f / \Delta C_p - 1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>19</td>
<td>30</td>
<td>0.108 ± 0.016</td>
<td>0.099 ± 0.032</td>
<td>0.086 ± 0.030</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>0.382 ± 0.017</td>
<td>0.364 ± 0.035</td>
<td>0.051 ± 0.005</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.663 ± 0.019</td>
<td>0.636 ± 0.038</td>
<td>0.042 ± 0.003</td>
</tr>
<tr>
<td></td>
<td>1,000</td>
<td>1.057 ± 0.021</td>
<td>1.015 ± 0.043</td>
<td>0.041 ± 0.002</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>1.555 ± 0.025</td>
<td>1.500 ± 0.050</td>
<td>0.037 ± 0.001</td>
</tr>
<tr>
<td></td>
<td>10,000</td>
<td>2.282 ± 0.030</td>
<td>2.192 ± 0.060</td>
<td>0.041 ± 0.001</td>
</tr>
<tr>
<td>Average</td>
<td></td>
<td>0.050 ± 0.031</td>
<td></td>
<td></td>
</tr>
<tr>
<td>24</td>
<td>30</td>
<td>0.156 ± 0.016</td>
<td>0.145 ± 0.031</td>
<td>0.073 ± 0.018</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>0.419 ± 0.017</td>
<td>0.399 ± 0.034</td>
<td>0.049 ± 0.005</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.699 ± 0.019</td>
<td>0.671 ± 0.038</td>
<td>0.042 ± 0.003</td>
</tr>
<tr>
<td></td>
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<td>1.093 ± 0.021</td>
<td>1.050 ± 0.043</td>
<td>0.041 ± 0.002</td>
</tr>
<tr>
<td></td>
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<td>1.593 ± 0.025</td>
<td>1.535 ± 0.050</td>
<td>0.038 ± 0.001</td>
</tr>
<tr>
<td></td>
<td>10,000</td>
<td>2.320 ± 0.030</td>
<td>2.220 ± 0.060</td>
<td>0.045 ± 0.001</td>
</tr>
<tr>
<td>Average</td>
<td></td>
<td>0.048 ± 0.019</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

change in the KCS hull’s frictional resistance coefficient is always higher than that of the equivalent flat plate, with $K_{ACF}$ values of ~ 0.05 (Table 4). The propagated error is higher for $K_{ACF}$ (Table 4) compared to $K_f$ (Table 3), which is due to larger relative uncertainties in $\Delta C_f$ and $\Delta C_p$ at low $k_s$ values. However, the average $K_{ACF}$ values (Table 4) are not significantly different from those obtained for $K_f$ (Table 3), thus suggesting that the same form effect applies to both frictional resistance and the friction penalty due to hull roughness/fouling, at both speeds (19 and 24 knots).

**Pressure form factor, $K_p$**

The pressure form factor $K_p$ is the difference between the overall form factor $K$ and friction form factor $K_f$ (Equations 9 and 10). In Table 3, a value $K = 0.1$ was assumed (Castro et al. 2011), since the $K$ could not be directly determined from simulation results (double-body simulations would be required; Raven et al. 2008). Consequently, the $K_p$ values in Table 3 should be analysed with caution, as these depend on a $K$ value from a different source and unknown uncertainty. However, these results appear to suggest that form effects on viscous pressure are comparable to form effects on friction.

**Form effect on $\Delta C_T$**

The hypothesis that the hull form can affect the resistance penalty due to biofouling is summarised by Equation 15: the change in total resistance due to hull roughness at the ship scale ($\Delta C_{T,p}$) is proportional to the change in frictional resistance of the equivalent flat plate ($\Delta C_{p,0}$) by an overall form factor (1-$K$). This hypothesis is tested by comparing $\Delta C_T$ (KCS hull) to $\Delta C_{p,0}$ (flat plate), as presented in Table 5 and Figure 2. The corresponding form factor, $K_{ACF}$, is presented in Table 5.

It is observed that, for the lower speed of 19 knots, $\Delta C_T$ (KCS hull) is always larger than $\Delta C_{p,0}$ (flat plate), with a form factor of 0.058 ± 0.025 (average value ± propagated discretisation error). This is in good agreement with the hypothesis of an overall form effect on hull roughness/biofouling penalties. Additionally, $K_{ACF} = 0.058 ± 0.025$ appears to agree with an overall form factor of $K = 0.1$ (Castro et al. 2011), suggesting approximately the same form effect on both full-scale resistance and hull roughness penalties. However, the hypothesis is not supported by the results at a higher speed of 24 knots, where $K_{ACF}$ is approximately null (~ 0.012 ± 0.012). The same can be observed in Figure 2, where significant differences between the KCS hull and the flat plate (ie form effects) are only observed for the lower speed of 19 knots. Some other effect seems to cancel out the form effect on friction that was previously described. Therefore, further decomposition of
Significant changes in \( C_W \) as a function of hull roughness were noted: for 19 knots, \( C_W \) is 18% lower for a heavily fouled hull compared to the smooth hull, whereas for 24 knots \( C_W \) was up to 30% lower. It should also be noted that \( C_W \) obtained in this study was higher than previously reported in Demirel et al. (2017), as Equation 16 uses the flat plate \( C_F \) instead of the KCS hull \( C_F \). Thus, whereas previously reported \( C_W \) values corresponded to 0.060–0.208 \( \times 10^{-3} \) for 19 knots, and 0.237–0.534 \( \times 10^{-3} \) for 24 knots (Demirel et al. 2017), these values correspond now to 0.231–0.281 \( \times 10^{-3} \) and 0.425–0.611 \( \times 10^{-3} \), for 19 and 24 knots respectively (Table 6). Also, it is noted that, for each speed, the change in \( C_W \) due to hull roughness is not as strong as previously reported in Demirel et al. (2017), where changes corresponded to –56% and –72%, for 19 and 24 knots respectively.

Wave-making resistance

The wave-making resistance coefficient \( C_W \) was obtained using Equation 16 and the results are presented in Table 6. Significant changes in \( C_W \) as a function of hull roughness \( k_s \) were noted: for 19 knots, \( C_W \) is 18% lower for a heavily fouled hull compared to the smooth hull, whereas for 24 knots \( C_W \) was up to 30% lower. It should also be noted that \( C_W \) obtained in this study was higher than previously reported in Demirel et al. (2017), as Equation 16 uses the flat plate \( C_F \) instead of the KCS hull \( C_F \). Thus, whereas previously reported \( C_W \) values corresponded to 0.060–0.208 \( \times 10^{-3} \) for 19 knots, and 0.237–0.534 \( \times 10^{-3} \) for 24 knots (Demirel et al. 2017), these values correspond now to 0.231–0.281 \( \times 10^{-3} \) and 0.425–0.611 \( \times 10^{-3} \), for 19 and 24 knots respectively (Table 6). Also, it is noted that, for each speed, the change in \( C_W \) due to hull roughness is not as strong as previously reported in Demirel et al. (2017), where changes corresponded to –56% and –72%, for 19 and 24 knots respectively.

### Table 5

<table>
<thead>
<tr>
<th>( U ) [knots]</th>
<th>( k_s ) [μm]</th>
<th>( \Delta C_T \times 10^3 \text{(KCS)} )</th>
<th>( \Delta C_F \times 10^3 \text{(flat plate)} )</th>
<th>( K_{\Delta C_T} = \Delta C_T / \Delta C_F - 1 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>19</td>
<td>30</td>
<td>0.106 ± 0.019</td>
<td>0.099 ± 0.032</td>
<td>0.066 ± 0.024</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>0.381 ± 0.021</td>
<td>0.364 ± 0.035</td>
<td>0.046 ± 0.005</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.665 ± 0.023</td>
<td>0.636 ± 0.038</td>
<td>0.046 ± 0.003</td>
</tr>
<tr>
<td></td>
<td>1,000</td>
<td>1.070 ± 0.025</td>
<td>1.015 ± 0.043</td>
<td>0.054 ± 0.003</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>1.587 ± 0.028</td>
<td>1.500 ± 0.050</td>
<td>0.058 ± 0.002</td>
</tr>
<tr>
<td></td>
<td>10,000</td>
<td>2.362 ± 0.034</td>
<td>2.192 ± 0.060</td>
<td>0.077 ± 0.002</td>
</tr>
<tr>
<td><strong>Average</strong></td>
<td></td>
<td></td>
<td></td>
<td>0.058 ± 0.025</td>
</tr>
<tr>
<td>24</td>
<td>30</td>
<td>0.150 ± 0.023</td>
<td>0.145 ± 0.031</td>
<td>0.037 ± 0.010</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>0.379 ± 0.024</td>
<td>0.399 ± 0.034</td>
<td>–0.051 ± 0.005</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.646 ± 0.026</td>
<td>0.671 ± 0.038</td>
<td>–0.038 ± 0.003</td>
</tr>
<tr>
<td></td>
<td>1,000</td>
<td>1.023 ± 0.028</td>
<td>1.050 ± 0.043</td>
<td>–0.026 ± 0.001</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>1.522 ± 0.031</td>
<td>1.535 ± 0.050</td>
<td>–0.009 ± 0.0003</td>
</tr>
<tr>
<td></td>
<td>10,000</td>
<td>2.256 ± 0.036</td>
<td>2.220 ± 0.060</td>
<td>0.016 ± 0.001</td>
</tr>
</tbody>
</table>
| **Average**     |                |                                 |                                 | –0.012 ± 0.012                    

Figure 2. Hull roughness effect on resistance, \( \Delta C_T \), for the KRISO containership (KCS). (a) 19 knots, \( R_e = 2.29 \times 10^9 \), (b) 24 knots, \( R_e = 2.89 \times 10^9 \). Error bars correspond to propagated discretisation error of resistance coefficients \( C_T \) (KCS hull) and \( C_F \) (flat plate). Original data from Demirel et al. (2017). *Significant difference, based on numerical discretisation error.
In Table 6, the expected viscous form effects on the flat plate roughness penalty, \(K \times \Delta C_F\), is also presented for comparison with \(\Delta C_W\). Interestingly, at a speed of 19 knots, the negative change in \(C_W\) is not sufficient to overcome the positive form effect, where the sum \(K \times \Delta C_F + \Delta C_W\) is positive and increases with hull roughness (Table 6, 19 knots), explaining the form factor \(K_{ACT} = 0.058 \pm 0.012\) at 19 knots (Table 5). For 24 knots, the negative change in \(C_W\) is much higher than at 19 knots and cancels out the positive form effect, so that the sum \(K \times \Delta C_F + \Delta C_W\) is close to null (Table 6, 24 knots), explaining the approximately null form factor \(K_{ACT}\) at 24 knots (Table 5). Thus, the hypothesis of a form factor affecting hull penalties due to roughness (Equation 15) cannot be generalised for all speeds, since the effect of hull roughness on wave pattern cannot be neglected.

### Table 6. Wave-making resistance \(C_W\) and form effect on flat plate hull roughness penalty \(K \times \Delta C_F\), as obtained from published CFD results on the KRiSo containership hull (KCS) and equivalent flat plate (Demirel et al. 2017), using \(K = 0.1\) (Castro et al. 2011).

<table>
<thead>
<tr>
<th>(U) [knots]</th>
<th>(k_s) [µm]</th>
<th>(C_W \times 10^3)</th>
<th>(\Delta C_W \times 10^3)</th>
<th>(K \times \Delta C_F \times 10^3) (flat plate)</th>
<th>(K \times \Delta C_F + \Delta C_W \times 10^3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>19</td>
<td>0</td>
<td>0.281</td>
<td>-0.003 (-1%)</td>
<td>0.007</td>
<td>0.007</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>0.278</td>
<td>-0.003 (-1%)</td>
<td>0.010</td>
<td>0.007</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>0.261</td>
<td>-0.002 (-7%)</td>
<td>0.036</td>
<td>0.017</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.247</td>
<td>-0.034 (-12%)</td>
<td>0.064</td>
<td>0.029</td>
</tr>
<tr>
<td></td>
<td>1,000</td>
<td>0.234</td>
<td>-0.047 (-17%)</td>
<td>0.102</td>
<td>0.055</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>0.218</td>
<td>-0.063 (-22%)</td>
<td>0.150</td>
<td>0.087</td>
</tr>
<tr>
<td></td>
<td>10,000</td>
<td>0.231</td>
<td>-0.050 (-18%)</td>
<td>0.219</td>
<td>0.170</td>
</tr>
<tr>
<td>24</td>
<td>0</td>
<td>0.611</td>
<td>-0.009 (-1%)</td>
<td>0.015</td>
<td>0.005</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>0.602</td>
<td>-0.009 (-1%)</td>
<td>0.015</td>
<td>0.005</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>0.551</td>
<td>-0.060 (-10%)</td>
<td>0.040</td>
<td>0.020</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.519</td>
<td>-0.092 (-15%)</td>
<td>0.067</td>
<td>0.025</td>
</tr>
<tr>
<td></td>
<td>1,000</td>
<td>0.479</td>
<td>-0.132 (-22%)</td>
<td>0.105</td>
<td>0.027</td>
</tr>
<tr>
<td></td>
<td>3,000</td>
<td>0.445</td>
<td>-0.167 (-27%)</td>
<td>0.154</td>
<td>-0.013</td>
</tr>
<tr>
<td></td>
<td>10,000</td>
<td>0.425</td>
<td>-0.186 (-30%)</td>
<td>0.222</td>
<td>0.036</td>
</tr>
</tbody>
</table>

### Comparison of penalty prediction methods

Using as a reference the full-scale CFD results for the KCS hull, alternative methods for predicting the rough hull penalty are compared here. These methods are: (1) CFD results for the equivalent flat plate, and (2) the Granville method for the equivalent flat plate. Each of these flat-plate methods is also multiplied by the form factor \((1+K)\) for comparison. The results are presented in Figure 3.

For 19 knots \((Re_s = 2.29 \times 10^9)\), the CFD flat plate results (Figure 3a, ‘flat CFD’) underestimate the effect of roughness on the KCS hull by as much as 4–7%, as expected from previous analyses (Table 5, 19 knots: \(K_{ACT} = 0.058 \pm 0.012\)). The results from the Granville method underestimate \(\Delta C_T\) by 17% for \(k_s = 30\) µm, but are within –4% to +0.2% of the KCS hull reference for all other hull conditions. Multiplication of CFD flat plate results by \((1+K)\) leads to a smaller error, with 2–5% overestimation, while applying \((1+K)\) to the results of the Granville method generally leads to worse results, with –8% to +10% differences relative to the KCS hull. Overall, for 19 knots, the methods are ranked as follows, in order of decreasing agreement with KCS hull results: (1) CFD flat plate results multiplied by \((1+K)\), (2) the CFD flat plate results, (3) the Granville method, and (4) the Granville method multiplied by \((1+K)\).

For 24 knots \((Re_s = 2.89 \times 10^9)\), the CFD flat plate results (Figure 3b, ‘flat CFD’) show errors relative to the KCS hull between –4% and +5%. Results from the Granville method show errors between –19% and +8%. Multiplication of CFD flat plate results by \((1+K)\) leads to an overestimation of as much as 6% to 16%, while applying \((1+K)\) to the Granville method results leads to errors of between –11% and +19%. Overall, for 24 knots, the methods are ranked as follows, in order of decreasing agreement with KCS hull results: (1) the CFD flat plate results, (2) the CFD flat plate

![Figure 3](image-url). Comparison of hull roughness penalty prediction methods for the KRiSo containership (KCS). (a) 19 knots, \(Re_s = 2.29 \times 10^9\), (b) 24 knots, \(Re_s = 2.89 \times 10^9\). Original data from Demirel et al. (2017).
results multiplied by $1 + K$, (3) the Granville method, and (4) the Granville method multiplied by $1 + K$.

From the above results, there is no clear advantage in considering a form factor in rough hull penalty predictions. This is supported by the above observation of a significant change in wave-making resistance resulting in cancelling effects. Overall, the time-saving Granville method seems to yield reasonably accurate results, with no further improvement by considering the form factor.

Finally, it should be stressed that the method used in obtaining the form factor, including which smooth friction line is selected (Garcia-Gomez 2000), has an important impact on the results. There can be large differences in the form factor between model-scale towing tank results and the full scale (Kouh et al. 2009), which might be related to practical problems such as partially immersed bulbous bows, immersed transoms, differences in wave-breaking, Reynolds and Froude number dependency, surface tension effects and occurrence of flow separation at model scale, particularly at the low speed range used for determining the form factor at model scale (ITTC 1990). Thus, a more accurate, full-scale estimation of the form factor $K$ as a function of speed should be considered in the future, eg by performing full-scale double-body CFD simulations (single phase) for the specific hull shape at different speeds of interest (Kouh et al. 2009). Also, the form factor should be determined relative to the same smooth friction line used in roughness penalty predictions (Schoenherr friction line in the Granville method, according to Schultz 2007), to obtain consistent results (Garcia-Gomez 2000). However, the present conclusions obtained with a form factor of $K = 0.1$ (Castro et al. 2011) should not change significantly, since the form factor is not expected to vary widely within the full-scale range of Reynolds number investigated ($2.29 \times 10^9$).

Considering the above, rapid calculation procedures used in economic and environmental assessment of the impact of hull roughness (eg the Granville method) should avoid using the form factor approach as initially hypothesised in this study, at least on vessels with significant wave-making resistance ($\geq 29\%$), as an incorrect form effect might significantly compromise the accuracy of predicted hull roughness penalties on calm-water resistance. Nevertheless, since positive $K_{\Delta C T}$ values could be obtained at lower speed for the KRISO containership (Table 5), future studies should investigate further whether the form factor approach of Equation 15 holds for vessels with a lower percentage of wave-making resistance ($\leq 16\%$), and also for vessels with a higher form factor than the KRISO containership, such as tankers.

**Conclusions**

Form effects on ship resistance arise from flow differences between a ship’s hull and the equivalent flat plate (same length and wetted surface area), leading to higher friction and pressure resistance on the hull. Since some methods for predicting penalties due to hull roughness/biofouling make use of flat plate models, eg the Granville method, it was hypothesised that similar form effects on flat plate resistance would also be observed on hull resistance penalties due to biofouling.

Re-analysis of CFD results from a previous study on the KRISO containership and the equivalent flat plate, at varying hull roughness height (Demirel et al. 2017), show form effects at the lower speed of 19 knots, with $K_{\Delta C T} = 0.058 \pm 0.012$, which however vanish at the higher speed of 24 knots. This is currently attributed to cancelling effects from the traditionally overlooked hull roughness effect on wave-making resistance, which is more significant at a vessel speed of 24 knots, compared to 19 knots. However, additional evidence of roughness effects on wave-making resistance should be gathered in future studies, particularly by experimental fluid dynamics.

Considering the current results, the application of a form effect into flat-plate predictions on hull roughness penalties is discouraged, although it might still be valid at low vessel speeds. Future work should further consider cases with low wave-making resistance ($\leq 16\%$ of total resistance) and also for vessels with a higher form factor. The form factor should preferably be determined for each speed, using full-scale double-body CFD simulations (single phase) and referring to the same smooth friction line used in roughness penalty predictions.

**Nomenclature**

- $B$: ship’s breadth
- $C$: smooth wall log-law intercept
- $C_B$: block coefficient, $C_B = \sqrt{LBT}$
- $C_A$: correlation allowance (constant)
- $C_{A, ITTC}$: Reynolds number-dependent correlation allowance (ITTC 2014)
- $C_F$: resistance coefficient, $C_F = R_F / (\frac{1}{2} \rho U^2 S)$
- $C_{R_F}$: flat plate resistance coefficient, $C_{R_F} = R_{F_0} / (\frac{1}{2} \rho U^2 S)$
- $C_{FF}$: form effect on frictional resistance coefficient, $C_{FF} = R_{FF} / (\frac{1}{2} \rho U^2 S)$
- $C_P$: pressure resistance coefficient, $C_P = R_P / (\frac{1}{2} \rho U^2 S)$
- $C_R$: residuary resistance coefficient, $C_R = R_R / (\frac{1}{2} \rho U^2 S)$
- $C_T$: total resistance coefficient, $C_T = R_T / (\frac{1}{2} \rho U^2 S)$
The authors would like to acknowledge Professor Rickard Bensow (Chalmers University of Technology), for interesting discussions and comments on an earlier version of the manuscript, and Superintendent Lars-Olof Albert, for valuable and encouraging discussions, and for generously allowing access to the full dry-docking process of a merchant vessel. The contribution of three anonymous reviewers to this paper is acknowledged.

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References

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Lena Granhag http://orcid.org/0000-0002-0340-7469

References

Superscript
+ inner-scaling of a variable, using \( \tau_w \) for velocity scale or \( v/\tau_w \) for length scale


ITTC. 2014. Recommended Procedures and Guidelines - 1978 ITTC performance prediction method 7.5-02-03-01.4 (Revision 03).


