Assessing the Hydrodynamic and Economic Impacts of Biofouling on the Hull of Surface Vessels Using Numerical Methods

by

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in
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April 20th, 2017

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We the undersigned committee hereby recommend that the attached document be accepted as fulfilling in part the requirements for the degree of Master of Science in Ocean Engineering.

“Assessing the Hydrodynamic and Economic Impacts of Biofouling on the Hull of Surface Vessels Using Numerical Methods”, a thesis by Letchi Evrard Quentin Anoman

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Abstract

TITLE: Assessing the Hydrodynamic and Economic Impacts of Biofouling on the Hull of Surface Vessels Using Numerical Methods

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Marine fouling on the hull of surface vessels is a topic of increasing importance in the maritime field for its environmental and financial impacts. Recent research developments have introduced mathematical models dealing with the frictional resistance associated with fouling, thus enabling to single out their impact on ship hydrodynamics. Herein, two implementations of these models are presented.

The first implementation used Computational Fluid Dynamics to assess the additional drag induced by fouling for a specific ship model. With the software STAR CCM+, Reynolds Averaged Navier-Stokes equations have been used to model turbulence, wall laws parameters have been adapted to suit a Colebrook-type engineering roughness function, and a hybrid wall treatment captures the flow near the wall. Full details of the simulation set-up are given.

The second implementation used turbulent flow similarity scaling laws in a MATLAB code to predict the added resistance of ships due to biofouling and the associated costs. With flexibility in mind, the code has been designed to account for the singularity of each context based on at least one in-situ observation of the fouling condition. Through a hypothetical, yet realistic scenario, it is shown that...
it enables proactive management by indicating when the cumulative penalty of fouling is no longer tolerable from a financial standpoint. The results of both implementations were validated against experimental data found in the literature. Prediction of the additional drag caused by fouling on a frigate showed excellent agreement of both methods.
Nomenclature

Latin Symbols

$1 + k_1$ Form factor
$A_T$ Transom area
$A_{BT}$ Transverse bulb area
$B$ Log-law intercept
$C_A$ Correlation allowance
$C_B$ Block coefficient
$C_c$ Cleaning cost
$C_F$ Frictional resistance coefficient
$C_f$ Local skin friction coefficient
$C_M$ Midship area coefficient
$C_P$ Prismatic coefficient
$C_R$ Residuary resistance coefficient
$C_T$ Total resistance coefficient
$C_V$ Viscous resistance coefficient
$C_W$ Wave resistance coefficient

v
\( C_{fuel} \)  
Cumulative added fuel cost

\( c_{fuel} \)  
Fuel price

\( C_{WP} \)  
Waterplane area coefficient

\( E \)  
Wall function coefficient

\( F_n \)  
Froude number

\( FC \)  
Cumulative extra fuel consumption

\( g \)  
Acceleration due to gravity

\( h_B \)  
Center of bulb area above keel line

\( i_E \)  
half angle of entrance

\( K \)  
Turbulent kinetic energy

\( k \)  
Roughness height

\( K_F \)  
Cumulative added power

\( k_s \)  
Sand grain roughness height

\( L \)  
Length of ship

\( L_R \)  
Length of run

\( L_{WL} \)  
Length at waterline

\( lcb \)  
longitudinal center of buoyancy forward of 0.5\( L \) as a percentage of \( L \)

\( P_B \)  
Brake power

\( P_E \)  
Effective power

\( R_a \)  
Centerline averaged roughness height

\( R_n \)  
Reynolds number
$R_T$  Total resistance coefficient

$R_t$  Maximum peak to trough roughness height

$R_W$  Wave resistance

$R_{t50}$  Maximum peak to trough roughness height over a length of 50mm

$S$  Wetted surface area

$T$  Threshold time

$T_F$  Draft at forward perpendicular

$T_{sea}$  Time spent at sea

$U$  Velocity

$U_e$  Freestream velocity

$U_f$  Friction velocity

**Greek Symbols**

$\Delta U^+$  Roughness function

$\delta$  Boundary layer thickness

$\epsilon$  Turbulent energy dissipation rate

$\eta$  Efficiency of the propulsion line

$\kappa$  Von Karman constant

$\mu$  Dynamic viscosity

$\nabla$  Volumetric displacement

$\nu$  Kinematic viscosity

$\omega$  Turbulent specific energy dissipation rate
\( \rho \)  
Density of fluid

\( \tau \)  
Shear stress

\( \tau_w \)  
Wall shear stress

**Superscripts**

+  
Normalized variable

**Subscripts**

clean  
Clean, newly coated

exp  
experimental

r  
rough

sm  
smooth

**Acronyms**

ABS  
American Bureau of Shipping

AF  
Antifouling

ATTC  
American Towing Tank Conference

BMT  
British Maritime Technology

CFD  
Computational Fluid Dynamics

DNS  
Direct Numerical Simulation

DOF  
Degrees of freedom

EHP  
Estimated Hull Performance

EVM  
Eddy-viscosity Model
<table>
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<th>Description</th>
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<tr>
<td>GHG</td>
<td>Greenhouse Gas</td>
</tr>
<tr>
<td>IFO</td>
<td>Intermediate Fuel Oil</td>
</tr>
<tr>
<td>IMO</td>
<td>International Maritime Organization</td>
</tr>
<tr>
<td>IPPIC</td>
<td>International Paint and Printing Ink Council</td>
</tr>
<tr>
<td>ITTC</td>
<td>International Towing Tank Conference</td>
</tr>
<tr>
<td>LES</td>
<td>Large Eddy Simulation</td>
</tr>
<tr>
<td>MEPC</td>
<td>Marine Environment Protection Committee</td>
</tr>
<tr>
<td>NSTM</td>
<td>Naval Ships’ Technical Manual</td>
</tr>
<tr>
<td>RANS</td>
<td>Reynolds Averaged Navier-Stokes</td>
</tr>
<tr>
<td>ROV</td>
<td>Remotely Operated Vehicle</td>
</tr>
<tr>
<td>SFOC</td>
<td>Specific Fuel Oil Consumption of Engine</td>
</tr>
<tr>
<td>SPC</td>
<td>Self Polishing Copolymer</td>
</tr>
<tr>
<td>SST</td>
<td>Shear Stress Transport</td>
</tr>
<tr>
<td>TBT</td>
<td>Tributyl Tin</td>
</tr>
<tr>
<td>USN</td>
<td>United States Navy</td>
</tr>
<tr>
<td>VOF</td>
<td>Volume of Fluid</td>
</tr>
<tr>
<td>WHOI</td>
<td>Woods Hole Oceanographic Institution</td>
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Dedication

To God, the source of all grace and knowledge,

To my loving wife, for her continuous and unfailing support, despite having to bear the pain of much distance,

And to our baby, growing healthily in her womb, to whom I hope to be a good father.
Chapter 1

Introduction

1.1 Motivation

Marine fouling or biofouling can be defined as the settlement and growth of marine organisms on a submerged hull surface. When these organisms attach to the hull of a ship, they increase the roughness of the hull surface. As a result, the viscous resistance and the required propulsive power of the ship increase causing an environmental as well as a financial impact.

The International Paint and Printing Ink Council (IPPIC) reported to the International Maritime Organization (IMO) that fouling free vessels can offer a significant contribution to the reduction of greenhouse gas (GHG) emissions from ships (IPPIC, 2010). The report estimates that if the global fleet were to have only a thin layer of slime, 134 million tonnes of extra $CO_2$ would be emitted into the atmosphere by 2020. Moreover, marine fouling poses a threat to the environment through the transfer of invasive species. For this reason, several countries have adopted regulations and the IMO published guidelines to
address the issue (IMO, 2011).

Haslbeck and Bohlander (1992) asserted that nearly 20% of the United States Navy (USN) Fleet propulsive fuel bill is spent annually to overcome the additional drag due to fouling. Schultz et al. (2011) estimated that the overall cost for the USN associated with fouling for the entire DDG-51 class of destroyer is nearly $1b over a 15-year period. Kattan et al. (2015) asserted that fuel consumption may account for 50% or more of a vessel operating costs. Clearly, this ratio will further increase if the resistance of the vessel is affected by biofouling. Considering these facts, the marine industry devotes much efforts to effectively manage biofouling through prevention, monitoring and cleaning.

It is of interest to ship researchers to be able to simulate the effects of biofouling on the hull of surface vessels. A logical extension of this capability is, among others, the ability to predict when it is desirable to undertake hull cleaning from an economic perspective. Therefore, the current study presents a Computational Fluid Dynamics (CFD) implementation of existing mathematical models to capture the drag induced by biofouling. A numerical code is also presented which provides an alternative way of calculating the added power and associated costs of biofouling on a vessel under calm water conditions. This latter tool is flexible enough to give an estimate of when the cumulative penalty of fouling overcomes the cleaning cost.

1.2 The Biofouling Phenomenon

Biofouling occurs within minutes after a body is submerged with the formation of an organic film. This is followed by the colonization of bacterial slimes and
diatoms. Within weeks, larger organisms such as barnacles and macro algae develop. These have a greater impact on a ship’s resistance.

The fouling community is composed of animals (barnacles, hydroids, mollusks, etc.), and plants (brown algae, green algae, etc.). It is well known that the extent and severity of biofouling depends on several factors such as temperature, water salinity, light, nutrient availability and other environmental conditions (WHOI and USN, 1952; Hunsucker et al., 2014). Vessels operating at low speeds or being at berth for extended periods are more likely to become fouled.

A classification of major fouling species is presented in Table 1.1.

---

Figure 1.1: Example of slime fouling on a ship's hull, ABS (2013)
Table 1.1: Classification of major fouling species, Koka (2014)

<table>
<thead>
<tr>
<th>Group</th>
<th>Description</th>
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<tbody>
<tr>
<td>Slime</td>
<td>Biofilms containing bacteria, fungi, unicellular organisms, diatoms, initial algal germination, and low form algae</td>
</tr>
<tr>
<td>Weed</td>
<td>Green or red mats of filamentous plants (e.g. Ulva spp., Ectocarpus)</td>
</tr>
<tr>
<td>Soft fouling</td>
<td>Low form plants and animals that form soft growths up to 150 mm in diameter</td>
</tr>
<tr>
<td>Shell fouling</td>
<td>Hard shelled crustacean worms and mollusks including barnacles, tubeworms and oysters</td>
</tr>
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1.3 Biofouling Management

Typically, biofouling management is addressed through prevention, monitoring and cleaning.

Prevention of marine fouling on the underwater hull of a vessel is made by antifouling coating systems. Since the ban by the IMO of organotin-based systems (notably the well-known tributyl-tin or TBT), the maritime industry has been striving to find an effective replacement. Modern coating systems can be classified into biocide-based and biocide free systems. It should be noted that they are also commonly divided into the following groups which reflect more the technology used: control depletion polymers (CDP), self-polishing co-polymers (SPC), foul release coatings (FRC), and combination technologies.

Monitoring the state of an antifouling coating system can be achieved by measuring fuel consumption on regularly spaced time intervals. Yet, such a task is not an easy one as many factors influence data collection: weather (wind
and waves), sea currents, temperature and salinity, ship’s loading, etc. Munk et al. (2009) gives a more thorough analysis of in-service vessel performance monitoring and notes that so far there is no standardized method to measure vessel performance. A simpler approach to assess the fouling condition of a hull is by a visual inspection using a diver or Remotely Operated Vehicle (ROV) with a waterproof camera. Notwithstanding the advantages of this method, it is regarded as subjective.

If the hull is judged as excessively fouled, several cleaning options are available. In the first option, waterborne cleaning can be carried out. It has the advantage of not affecting much the ship’s availability. As noted by the USN Naval Ships’ Technical Manual (NSTM, 2006), “total ship performance and fleet capability can be enhanced by waterborne cleaning and maintenance”. The manual specifies when cleaning should occur and defines several types of waterborne cleaning depending on the sections to be cleaned: full cleaning, interim cleaning and partial cleaning. The US Navy is also investigating a proactive hull maintenance procedure known as grooming (Tribou and Swain, 2010). This is defined as cleaning with soft tools before fouling becomes established (for example, hull grooming of a slimed hull). In the second option, cleaning may take place during dry-docking. In this case, a full or partial blast of the hull plus recoating are possible.

More detailed reviews of biofouling management are developed by INTER-TANKO (2016) and Kattan et al. (2015).
Chapter 2

Theoretical Background

2.1 Computational Fluid Dynamics

Computational Fluid Dynamics (CFD) is the field of study that uses computers and numerical methods to solve problems involving fluid flow. Three main governing equations are solved (Cebeci et al., 2005):

- Continuity equation (conservation of mass)

\[
\frac{\partial \rho}{\partial t} + \nabla (\rho \mathbf{u}) = 0 \tag{2.1}
\]

- Momentum equation (Navier-Stokes)

\[
\frac{\partial \mathbf{u}}{\partial t} + (\mathbf{u} \cdot \nabla) \mathbf{u} = -\frac{1}{\rho} \nabla p + \frac{1}{\rho} \nabla \tau + \frac{1}{\rho} F \tag{2.2}
\]

- Energy equation (first law of thermodynamics)

\[
\frac{\partial E}{\partial t} + \nabla (E \cdot \mathbf{u}) = -\nabla q + \nabla (\Pi_{ij} \cdot \mathbf{u}) + F \cdot \mathbf{u} + S_E \tag{2.3}
\]
Here, \( \mathbf{u} \) is the velocity vector, \( p \) is the pressure, \( \tau \) is the shear stress tensor, \( F \) is the resultant of external body forces, \( E \) is the total energy per unit volume, \( q \) is the heat flux, \( S_E \) is an energy source per unit volume. \( \Pi_{ij} \) is the stress tensor given as the sum of normal and shear stresses, \( \Pi_{ij} = -p\delta_{ij} + \tau_{ij} \) where \( \delta_{ij} \) is the Kronecker delta function (\( \delta_{ij} = 1 \) if \( i = j \), \( \delta_{ij} = 0 \) if \( i \neq j \)) for \( i, j = 1, 2, 3 \).

Typically, a CFD simulation involves three stages:

- **Pre-processing:** at this stage, the problem is formulated by creating the geometry, selecting the applicable equations, defining the boundary conditions and constructing the mesh;

- **Solving:** the CFD package discretizes and solves the equations over the elementary volumes of the mesh;

- **Post-processing:** the results are visualized and analyzed.

An overview of the simulation workflow in STAR CCM+ is presented in Figure 2.1.

Turbulence is one of the many complex engineering fluid flow problems that CFD endeavors to tackle. As in most engineering applications, the flow along the hull of a ship (even more a fouled hull) is turbulent. Turbulence can be modelled by Direct Numerical Simulation (DNS), Large Eddy Simulation (LES) or Reynolds Averaged Navier-Stokes (RANS) equations (Figures 2.2 to 2.4). While DNS solves for the exact governing equations with an extremely fine grid spacing, RANS equations provide closure relations by an averaging process called Reynolds averaging. This process, which is computationally more convenient, considers any instantaneous flow variable as the sum of a mean and a fluctuating variable. For example, the instantaneous velocity would be \( \mathbf{u} = \overline{\mathbf{u}} + \mathbf{u}' \), where \( \overline{\mathbf{u}} \)
Figure 2.1: General sequence of operations in a STAR CCM+ analysis (CD-ADAPCO, 2017)

and $u'$ denote respectively the mean velocity and the fluctuating velocity. LES is a combination of both DNS and RANS with an attempt to solve only for the largest eddies.
Figure 2.2: Direct Numerical Simulation (Cengel and Cimbala, 2006)

Figure 2.3: Large Eddy Simulation (Cengel and Cimbala, 2006)

Figure 2.4: Reynolds Averaged Navier-Stokes Simulation (Cengel and Cimbala, 2006)
The Reynolds averaging process gives rise to Reynolds stresses (or turbulent stresses) such that the shear stress becomes the sum of viscous and turbulent components i.e.:

$$\tau_{ij} = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \tau_{ij,turbulent} \tag{2.4}$$

Where $\tau_{ij,turbulent} = -\rho \bar{u}_i \bar{u}_j$.

As per the eddy-viscosity model (EVM) assumption,

$$\tau_{ij,turbulent} = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \rho K \delta_{ij} \tag{2.5}$$

Here, $\mu_t$ is the eddy viscosity (or turbulent viscosity) which may be modelled in terms of the turbulent kinetic energy $K$ and the turbulent energy dissipation rate $\epsilon$ ($K-\epsilon$ model) or the turbulent kinetic energy $K$ and the turbulent specific energy dissipation rate $\omega$ ($K-\omega$ model).

### 2.2 Prediction of the Required Propulsive Power of Ships

The motion of a ship in seaway is altered by fluid forces opposing the motion. These forces constitute the resistance of the ship. They can be divided into two main components: the frictional resistance, due to tangential shear stresses, and the residuary resistance, the major component of which is wave resistance. William Froude postulated that the wave resistance of a ship could be inferred from a geometrically similar model whereas frictional resistance is a function of the wetted surface area and speed.

Under the assumption that the frictional resistance of a ship is equal to that of an
equivalent flat plate (i.e. ship and plates having the same wetted surface area),
the American Towing Tank Conference (ATTC) of 1947 adopted the Schoenherr
friction line (Equation 2.6) to determine the frictional resistance coefficient $C_F$
of a ship. In 1957, the International Towing Tank Conference (ITTC) adopted
the ITTC-1957 model-ship correlation line (Equation 2.7) which was found to
be a more accurate prediction of $C_F$ for ships.

\[
\log(R_n \cdot C_F) = \frac{0.242}{\sqrt{C_F}}
\]  
(2.6)

\[
C_F = \frac{0.075}{(\log R_n - 2)^2}
\]  
(2.7)

Here, $R_n$ is the Reynolds number of the ship given as $R_n = \frac{UL}{\nu}$ where $U$ and
$L$ are respectively the velocity and length of the ship, and $\nu$ is the kinematic
viscosity of the fluid (typically seawater). There is a good agreement between
both friction lines for $R_n > 10^8$.

According to the 2D extrapolation procedure recommended by the ITTC-1957,
the effective power required to move a ship at speed $U$ is:

\[
P_E = R_T \cdot U
\]  
(2.8)

$R_T$ is the total resistance of the ship given as $R_T = \frac{1}{2} \rho C_T S U^2$, where $\rho$ is the
density of the fluid, $S$ is the wetted surface area of the ship and $C_T$ is the coefficient
of total resistance. The latter is expressed as $C_T = C_F + C_R + C_A$ where the
residuary resistance coefficient $C_R$ is typically determined from model testing
or empirical formulations, and the correlation allowance $C_A$ is an incremental
resistance coefficient used to account for the paint/surface roughness of the new
ship as built.

It should be noted that the procedure was further improved by the adoption of a 3D extrapolation procedure known as the 1978-ITTC performance prediction method (Oosterveld, 1978). The major improvement is the introduction of a form factor $(1 + k_1)$ to take into account the three-dimensional shape of a ship such that $C_V = (1 + k_1)C_F$ and $C_T = C_V + C_R + C_A$. The 3D extrapolation procedure also suggests methods for the calculation of appendage resistance.

Finally, the engine power or brake power $P_B$, i.e. the power delivered at engine coupling can be expressed as:

$$P_B = \frac{P_E}{\eta}$$

(2.9)

Where $\eta$ is the efficiency of the entire propulsion system (combining quasi-propulsive efficiency, shaft efficiency, and gear efficiency).

Holtrop and Mennen (1982) and Holtrop (1984) proposed regression equations for the calculation of the wave resistance of ships (see Appendix A). The recommended ship types and parameter ranges are listed in Table 2.1. Here, $F_n$, $C_P$ and $B$ are respectively the Froude number ($F_n = \frac{U}{\sqrt{gL}}$), the prismatic coefficient and the maximum beam of the vessel.

Table 2.1: Ship types and parameters range (Holtrop and Mennen, 1982; Molland et al., 2011)

<table>
<thead>
<tr>
<th>Ship type</th>
<th>$F_{n,max}$</th>
<th>$C_P$</th>
<th>$L/B$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tankers and bulk carriers</td>
<td>0.24</td>
<td>0.73-0.85</td>
<td>5.1-7.1</td>
</tr>
<tr>
<td>General cargo</td>
<td>0.30</td>
<td>0.58-0.72</td>
<td>5.3-8.0</td>
</tr>
<tr>
<td>Fishing vessels, tugs</td>
<td>0.38</td>
<td>0.55-0.65</td>
<td>3.9-6.3</td>
</tr>
<tr>
<td>Container ships, frigates</td>
<td>0.45</td>
<td>0.55-0.67</td>
<td>6.0-9.5</td>
</tr>
</tbody>
</table>
2.3 Boundary Layer Theory

The boundary layer can be defined as “the area closest to the hull in which the fluid is impeded as a result of its viscosity” (Schultz and Swain, 2000). Along the hull surface of a ship, the boundary layer is mostly turbulent, i.e. the velocity and pressure distribution have random fluctuations in space and time. A turbulent boundary layer can be divided into the following (Figure 2.5):

- the inner region composed of:
  - the viscous (or linear) sublayer where \( U^+ = y^+ \)
  - the log-law layer, Schlichting (1968), where
    \[
    U^+ = \frac{1}{\kappa} \ln(y^+) + B - \Delta U^+ \tag{2.10}
    \]
  - the buffer layer which is a transitional layer between the viscous sublayer and the log-law layer

- the outer region where the local mean velocity is governed by the velocity-defect law \( U_e^+ - U^+ = g(\frac{y}{\delta}) \)

\( U^+, U_e^+, y^+ \) and the roughness Reynolds number \( k^+ \) are normalized quantities given as \( U^+ = \frac{U}{U_e} \), \( U_e^+ = \frac{U_e}{U_e} \), \( y^+ = \frac{y U_e}{\nu} \), and \( k^+ = \frac{k U_e}{\nu} \) where \( U_e \) is the velocity at the edge of the boundary layer, \( U_e = \sqrt{\frac{2w}{\rho}} \) is the friction velocity, \( y \) is the offset distance from the boundary i.e. the wall, and \( k \) is the roughness height (note that \( k \) is an artificial parameter used to assess roughness effects, it is not a physical measurement). \( \kappa \) is the Von Karman constant, \( B \) is the log-law intercept for smooth walls, \( \delta \) is the boundary layer thickness, and \( g \) denotes the
velocity-defect function which is thought to be universal. \( \Delta U^+ \) is the roughness function which accounts for the effects of roughness in the flow field (Figure 2.6).

Figure 2.5: Law of the wall plot for a turbulent boundary layer. \( K \) is the Von Karman constant, \( E' \) is the ratio of the wall function coefficient by the roughness function (CD-ADAPCO, 2017)

### 2.4 Turbulent Flow Similarity and Scaling Law

Granville (1958) postulated that “the frictional effects of any particular roughness may be considered defined when \( \Delta U^+ \) is experimentally determined as a function of \( k^+ \)”. He further argued that “if a representative sample of the rough surface can be obtained on a test plate, it is then possible to make full-scale predictions for any arbitrary roughness configuration without regard to geometrical characterization”. Then, he developed a scaling procedure based on the
following relationships:

\[ \log R_n = \log R_{n,sm} + \frac{(\Delta U^+ - \Delta U_{sm}^+)}{\ln 10} \]  

(2.11)

\[ \log R_n = \log \left( \frac{L U_\tau}{\nu} \right) - \log \frac{C_F}{2} + \frac{\sqrt{C_F}}{\kappa \cdot \ln(10)} \]  

(2.12)

\[ \log R_{n,2} = \log R_{n,1} + \log \left( \frac{L_2}{L_1} \right) \]  

(2.13)

Equation 2.11, valid at constant \( C_F \), means that the frictional curve of a rough plate is offset by a distance \( \frac{\Delta U^+ \cdot \kappa}{\ln(10)} \) from the smooth curve; Equation 2.12 means
that log $R_n$ can be plotted as a function of $C_F$ for a given viscous length scale \( \frac{U}{\nu} \) and plate length; Equation 2.13 means that log $R_n$ of plate 2 is a distance log $\frac{L_2}{L_1}$ from log $R_n$ of plate 1 for a given roughness height and a given viscous length scale.

Therefore, given two plates of respective lengths $L_1$ and $L_2$, with the same roughness, if the velocity of the second plate is known which means $R_{n,2}$ is known, and the roughness functions obtained experimentally with plate 1 are known, we have (as proposed by Granville (1958)):

(a) $R_{n,1}$ is found from Equation 2.13

(b) The viscous length scale is found by solving for Equations 2.11 and 2.12 iteratively,

(c) The rough frictional resistance coefficient curve $C_{Fr}$ is obtained from Equation 2.11,

(d) The rough frictional resistance coefficient of plate 2 is $C_{F2} = C_{Fr}(\log R_{n,2})$.

A graphical representation of the procedure is presented in Figure 2.7. If the roughness function is defined by an empirical formulation, step (a) can be omitted. It should be noted that a similar iterative procedure has been described by Schultz (2007). More recently, Monty et al. (2016) also described a similarity scaling procedure based on momentum equations.
2.5 Roughness Functions for Antifouling Coatings and Fouled Surfaces

Schultz (2004) tested in a towing tank several coating systems in the unfouled, fouled, and cleaned conditions with plates of length \( L = 1.52m \) (Table 2.2). Based on his experiments, he found that a Colebrook-type engineering roughness function appropriately characterizes the roughness of unfouled and fouled coatings:

\[
\Delta U^+ = \frac{1}{k} \ln(1 + k^+) \tag{2.14}
\]

Where

(2.14a) \( k = 0.17R_a \) for unfouled coating

(2.14b) \( k = 0.11R_t \) for surfaces covered with slime only
\( (2.14c) \ k = 0.059R_t\sqrt{\%\text{barnacle fouling}} \) for fouled surfaces with barnacles.

\( R_a \) and \( R_t \) denote physical measurements representing the centerline average roughness height and the maximum peak to trough roughness height respectively. In the case of fouled surfaces with slime or barnacles, \( R_t \) means the slime thickness or maximum barnacle height respectively. Note that \( k \) is an artificial parameter as it relates to physical measurements as illustrated by Equations 2.14a, 2.14b and 2.14c.

Equation 2.14 exhibited excellent agreement between experimental results and the formulations for unfouled coatings and coatings covered with barnacles. It is of note that fouling occurred under static conditions. Even if it is true that \( \Delta U^+ \) would be different for fouling occurring under dynamic conditions (Schultz et al., 2015; Hunsucker et al., 2016; Zargiel and Swain, 2014), it is a good starting point for practical evaluation.

If \( R_a \) and \( R_t \) are not available - which is generally the case in dock-, Table 2.3 can be used as an alternative reference for the determination of the roughness height \( k \). The NSTM rating is an index based on a visual assessment of the fouling condition (NSTM, 2006). Figure 2.8 shows examples of such characterization. Note that \( k_s \) is the equivalent sand roughness height which conforms to a Nikuradse-type roughness function (Nikuradse, 1933). \( R_{50} \) is \( R_t \) for a given length of 50mm; in the case of fouled surfaces, it implies the likely observable maximum peak to trough roughness height.
Table 2.2: Fouling condition of fouled plates, adapted from Schultz (2004)

<table>
<thead>
<tr>
<th>Test surface</th>
<th>Total fouling coverage (%)</th>
<th>Slime (%)</th>
<th>Hydroids (%)</th>
<th>Barnacles (%)</th>
<th>Height of highest barnacles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Silicone 1</td>
<td>75</td>
<td>10</td>
<td>5</td>
<td>60</td>
<td>≈ 6mm</td>
</tr>
<tr>
<td>Silicone 2</td>
<td>95</td>
<td>15</td>
<td>5</td>
<td>75</td>
<td>≈ 7mm</td>
</tr>
<tr>
<td>Ablative copper</td>
<td>76</td>
<td>75</td>
<td>0</td>
<td>1</td>
<td>≈ 5mm</td>
</tr>
<tr>
<td>SPC copper</td>
<td>73</td>
<td>65</td>
<td>3</td>
<td>4</td>
<td>≈ 5mm</td>
</tr>
<tr>
<td>SPC TBT</td>
<td>70</td>
<td>70</td>
<td>0</td>
<td>0</td>
<td>NA</td>
</tr>
</tbody>
</table>

Table 2.3: Fouling reference table, adapted from Swain and Lund (2016); Schultz (2007); NSTM (2006)

<table>
<thead>
<tr>
<th>Condition</th>
<th>NSTM Fouling rating</th>
<th>$k(\mu m)$</th>
<th>$k_s(\mu m)$</th>
<th>$R_{50}(\mu m)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulically smooth</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Applied AF coating</td>
<td>0</td>
<td>2</td>
<td>30</td>
<td>150</td>
</tr>
<tr>
<td>Deteriorated coating</td>
<td>10-20</td>
<td>6</td>
<td>100</td>
<td>300</td>
</tr>
<tr>
<td>Light slime</td>
<td>10-20</td>
<td>17</td>
<td>150</td>
<td>400</td>
</tr>
<tr>
<td>Moderate slime</td>
<td>20-30</td>
<td>30</td>
<td>210</td>
<td>500</td>
</tr>
<tr>
<td>Heavy slime</td>
<td>30</td>
<td>60</td>
<td>300</td>
<td>600</td>
</tr>
<tr>
<td>Small calcareous fouling or weed</td>
<td>40-60</td>
<td>320</td>
<td>1000</td>
<td>1000</td>
</tr>
<tr>
<td>Medium calcareous fouling</td>
<td>70-80</td>
<td>1000</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td>Heavy calcareous fouling</td>
<td>90-100</td>
<td>3500</td>
<td>10000</td>
<td>10000</td>
</tr>
</tbody>
</table>
(a) Fouling rating FR-10 over 100% of area  
(b) Fouling rating FR-20 over 80% of area  
(c) Fouling rating FR-40 over 20% of area  
(d) Fouling rating FR-60 over 90% of area  
(e) Fouling rating FR-70 over 80% of area  
(f) Fouling rating FR-100 over 50% of area

Figure 2.8: Fouling rating (NSTM, 2006)
Chapter 3

CFD Modeling

The CFD package STAR CCM+ was used to simulate the effects of biofouling by modifying surface roughness parameters. The present study makes use of the roughness function defined in Equation 2.14. The roughness height $k$ is taken as per Equations 2.14a to 2.14c or Table 2.3. Here, the advantage is that a direct correlation between a visual assessment of the fouling severity is utilized to determine the drag. The results of the simulation are validated against the experimental data of Schultz (2004). Then, predictions are made for a frigate.

3.1 Simulation Set-up

3.1.1 Geometry

The plate geometry used is in conformity with the experimental setup described by Schultz (2004). It is a flat plate 1.52m long, 0.76 m wide, and 3.2 mm thick. The edges are filleted to a radius of 1.6 mm (Figure 3.1). The submerged height has been set to 0.59 mm.
Figure 3.1: Plate geometry
3.1.2 Computational domain

The domain was setup according the recommended practice of CD ADAPCO for general external flows. The domain extends 2L ahead of the leading edge, 4L behind of the trailing edge, and 2L above the free surface. Its depth and width were taken equal to the depth and width of the real towing tank (Figure 3.2).

3.1.3 Meshing

A mesh is a discretized representation of the computational domain, which the physics solvers use to provide a numerical solution (CD-ADAPCO, 2017). As such, creating the right mesh is an important aspect of the simulation.
The meshing models used in this study are the surface remesher, trimmed cell mesher, and prism layer mesher (refer appendix C for description). Four volumetric refinements were set up around the plate (Figure 3.3). Three volumetric refinements were set up to capture the flow at the free surface (Figures 3.4 to 3.6). Six volumetric refinements were setup to capture the wake (Figures 3.7 and 3.8). The resulting mesh is shown in Figures 3.9 to 3.12.

Figure 3.3: Plate refinements

Figure 3.4: Very thin free surface refinement
Figure 3.5: Thin free surface refinement

Figure 3.6: Thick free surface refinement
Figure 3.7: Wake refinement

Figure 3.8: Wakebox refinement
Figure 3.9: Plate mesh

Figure 3.10: Mesh perspective view
Figure 3.11: Mesh side view

Figure 3.12: Mesh top view
3.1.4 Physical models

In this study, Reynolds-Averaged Navier-Stokes (RANS) equations have been used to model turbulence. The Shear Stress Transport (SST) $K - \omega$ turbulence model was used to close the momentum equations. This model provides the advantages of the $K - \omega$ model near the wall and those of the $K - \epsilon$ model in the far field. It should be noted that a similar study on ships coatings by Demirel et al. (2014) used the same turbulence model.

3.1.5 Boundary conditions

The simulations were set-up for half a model with a symmetry plane which would save computational resources without affecting the end result. The Volume of Fluid (VOF) Damping option has been activated for boundaries that are sensitive to wave reflection as shown in Table 3.1. VOF Damping is an option used to reduce numerical instabilities arising from the reflection of waves.

Table 3.1: Boundary conditions

<table>
<thead>
<tr>
<th>Boundary name</th>
<th>Boundary type</th>
<th>Condition</th>
<th>VOF Damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tank.Inlet</td>
<td>Velocity inlet</td>
<td>NA</td>
<td>Yes</td>
</tr>
<tr>
<td>Tank.Outlet</td>
<td>Pressure outlet</td>
<td>NA</td>
<td>Yes</td>
</tr>
<tr>
<td>Tank.Side</td>
<td>Wall</td>
<td>No slip</td>
<td>Yes</td>
</tr>
<tr>
<td>Tank.Symmetry</td>
<td>Symmetry plane</td>
<td>NA</td>
<td>No</td>
</tr>
<tr>
<td>Tank.Top</td>
<td>Wall</td>
<td>Slip</td>
<td>No</td>
</tr>
<tr>
<td>Tank.Bottom</td>
<td>Wall</td>
<td>No slip</td>
<td>No</td>
</tr>
<tr>
<td>Plate</td>
<td>Wall (smooth/rough)</td>
<td>No slip</td>
<td>No</td>
</tr>
</tbody>
</table>
3.1.6 Wall laws

STAR CCM+ uses wall laws describing closely packed uniform sand grain roughness as given by Cebeci and Bradshaw (1977). In particular, the log-law formulation used in Star CCM+ is:

\[ U^+ = \frac{1}{\kappa} \ln(E^+ y^+) \]  

(3.1)

Here, \( E^+ = \frac{E}{f} \) where \( E \) is the wall function coefficient and \( f \) denotes the STAR CCM+ roughness function given as:

- \( f = 1 \) for the hydrodynamically smooth flow regime \((k^+ < k^+_{sm})\)
- \( f = [A(\frac{k^+-k^+_{sm}}{k^+_{sm}-k^+_r})+Ck^+]^a \) for the intermediate flow regime \((k^+_{sm} < k^+ < k^+_r)\)
- \( f = A + Ck^+ \) for the fully-rough flow regime \((k^+ > k^+_r)\)

Note that \( A \) and \( C \) denote coefficients, and the exponent \( a \) is a function of \( k^+ \),
$k_{sm}^+$ and $k_r^+$.

For the current study, these parameters needed to be adjusted to satisfy the roughness function proposed by Equation 2.14. To begin with, Equation 2.14 is independent of the flow regime. Therefore, $k_{sm}^+$ and $k_r^+$ were made very small so that $f$ would always have the form of the fully rough regime.

The coefficients $E$, $A$ and $C$ are then found as follows:

- Equalizing Equation 2.10 and 3.1 for smooth walls, we have:

\[
U^+ = \frac{1}{\kappa} \ln(Ey^+) = \frac{1}{\kappa} \ln(E) + \frac{1}{\kappa} \ln(y^+) = \frac{1}{\kappa} \ln(y^+) + B
\]  

(3.2)

Eliminating redundant terms, we get

\[
\frac{1}{\kappa} \ln(E) = B
\]  

(3.3)

And for $B = 5.0$, $\kappa = 0.42$

\[
E = e^{\kappa B} = 8.17
\]  

(3.4)

- Keeping the fully-rough expression of the STAR CCM+ roughness function, and equalizing Equations 2.10 and 3.1 for rough flows, we have

\[
U^+ = \frac{1}{\kappa} \ln\left(\frac{E}{f y^+}\right) = \frac{1}{\kappa} \ln( Ey^+ ) - \frac{1}{\kappa} \ln(A + C k^+) = \frac{1}{\kappa} \ln(y^+) + B - \Delta U^+
\]  

(3.5)

Eliminating redundant terms based on equation 3.2, and replacing by the
expression of $\Delta U^+$ in Equation 2.14, we obtain

$$\frac{1}{\kappa} \ln(A + Ck^+) = \frac{1}{\kappa} \ln(1 + k^+) \quad (3.6)$$

Which means that

$$A = 1 \quad (3.7)$$

And

$$C = 1 \quad (3.8)$$

In the end, the default values of the coefficients $E$, $A$, and $C$ were replaced in the simulation parameters by their respective values given in Equations 3.4, 3.7 and 3.8.

### 3.1.7 Solvers

In this study, the solvers used are (refer appendix C for description):

- Implicit unsteady
- 6-DOF solver
- 6-DOF Motion
- Partitioning
- Wall distance
- VOF wave zone distance
- Segregated flow
• Segregated volume fraction

• K-Omega ($K - \omega$) turbulence

• K-Omega ($K - \omega$) turbulent viscosity

### 3.1.8 Stopping Criteria

The simulations were set to run for 50 seconds of physical time. This arbitrary choice was set for three reasons. First, it is a good approximation of the duration of the real experiments; second, it was sufficient to achieve convergence in all scenarios (Figure 3.14); third, it saved computational resources.

The maximum inner iterations were set equal to 5. This parameter defines the number of iterations that the solver executes before moving to the following time step.

![Figure 3.14: Convergence of simulations in fouled condition](image)
3.2 Convergence Study

The convergence study was made on Silicone 1 in the fouled condition at a Reynolds number $R_n = 2.8 \cdot 10^8$. As a reminder, fouled Silicone 1 has a total fouling coverage of 75%, including 10% slime, 5% hydroids and 60% barnacles (Table 2.2). Its roughness height is calculated as per Equation 2.14c:

$$k = 0.059 \times 7\sqrt{60} = 2.742\text{mm} \quad (3.9)$$

In the following tables, the error is defined as:

$$\text{Error}_{exp}(\%) = \frac{C_{F,CFD} - C_{F,exp}}{C_{F,CFD}} \times 100 \quad (3.10)$$

3.2.1 Mesh

The base size of the mesh was varied to assess its impact on the results (Table 3.2). It appears that finer core meshes produced higher discrepancies. This is primarily because they were taking longer to converge in the physical running time of 30 seconds (Figures 3.15 and 3.16). Besides, the transition from prism layers to core mesh was not smooth (Figure 3.17). Also, even though a base size of 0.12 m yielded the most accurate results, the overall topology of the mesh did not look right (Figure 3.18). For this reason, a base size of 0.1 m was chosen to proceed further in the study.
Table 3.2: Mesh convergence study

<table>
<thead>
<tr>
<th>Base size of core mesh (m)</th>
<th>Number of cells ( \times 10^{-6} )</th>
<th>( R_F ) (N)</th>
<th>( C_{F,CFD} \times 10^3 )</th>
<th>( C_{F,exp} \times 10^3 )</th>
<th>Error_{exp}</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.12</td>
<td>0.57</td>
<td>23.642</td>
<td>13.21</td>
<td>12.6</td>
<td>4.87%</td>
</tr>
<tr>
<td>0.1</td>
<td>0.76</td>
<td>23.916</td>
<td>13.37</td>
<td>12.6</td>
<td>6.08%</td>
</tr>
<tr>
<td>0.07</td>
<td>1.5</td>
<td>24.202</td>
<td>13.53</td>
<td>12.6</td>
<td>7.35%</td>
</tr>
<tr>
<td>0.05</td>
<td>3.2</td>
<td>24.312</td>
<td>13.59</td>
<td>12.6</td>
<td>7.84%</td>
</tr>
</tbody>
</table>

Figure 3.15: Shear drag monitor plot
Figure 3.16: Shear drag in the last 12 seconds

Figure 3.17: Improper mesh transition (base size=0.05m)
3.2.2 Time step

After an initial wall convergence study, the time step was changed to assess its effect on convergence (Table 3.3). It is seen that reducing the time brings about faster convergence from the simulated physical time standpoint (Figure 3.19). However, the smaller the time step, the greater the computational time. Considering these facts, and noting that the best results were obtained with a time step of 0.3 second, it was decided to select a time step of 0.3 second to proceed further in the study.

3.2.3 Wall treatment

The result of a simulation analyzing surface roughness is very sensitive to the wall treatment, especially at high roughness levels. STAR CCM+ offers three wall treatment models. The low-$Y^+$ model resolves the viscous sublayer and
Table 3.3: Time convergence study

<table>
<thead>
<tr>
<th>Time step (s)</th>
<th>Number of cells ( \times 10^{-6} )</th>
<th>( R_F ) (N)</th>
<th>( C_{F,CFD} \times 10^3 )</th>
<th>( C_{F,exp} \times 10^3 )</th>
<th>Error_{exp}</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>0.78</td>
<td>22.838</td>
<td>12.76</td>
<td>12.6</td>
<td>1.25%</td>
</tr>
<tr>
<td>0.3</td>
<td>0.78</td>
<td>22.781</td>
<td>12.73</td>
<td>12.6</td>
<td>1.05%</td>
</tr>
<tr>
<td>0.5</td>
<td>0.78</td>
<td>22.883</td>
<td>12.78</td>
<td>12.6</td>
<td>1.5%</td>
</tr>
<tr>
<td>0.7</td>
<td>0.78</td>
<td>23.037</td>
<td>12.87</td>
<td>12.6</td>
<td>2.18%</td>
</tr>
</tbody>
</table>

Figure 3.19: Time convergence plot

is computationally expensive; the high-\( Y^+ \) model does not resolve the viscous sublayer but uses wall functions to derive flow variables; the all-\( Y^+ \) model is a hybrid wall treatment suitable when it is difficult to achieve a pure low-\( Y^+ \) or a pure high-\( Y^+ \) (CD-ADAPCO, 2017).

An all-\( Y^+ \) wall treatment has been selected for this study. When using this model, it is recommended that the initial wall cell centroid falls within the buffer layer (Figure 3.20). The sensitivity of the simulation results for fouled or
unfouled plates to the initial wall cell size can be seen in Tables 3.4 and 3.5. In any case, it appeared from the analysis of several other simulations that the best configuration is obtained when the initial wall cell size $y_0$ is comprised between 8% and 15% of an estimate of the boundary layer thickness $\delta_{lim}$ at half-length:

$$0.08\delta_{lim} < y_0 < 0.15\delta_{lim}$$  \hspace{1cm} (3.11)

$$\delta_{lim} = 5.926 \frac{L}{2} \cdot \overline{C_{F,r}}$$  \hspace{1cm} (3.12)

Here, $L$ is the length of the plate or ship, and $\overline{C_{F,r}}$ is an estimated fully-rough frictional resistance coefficient at $\frac{L}{2}$ given as (Faltinsen, 2005):

$$\overline{C_{F,r}} = [1.89 + 1.62 \log(\frac{L}{2k_s})]^{-2.5}$$  \hspace{1cm} (3.13)

It is of note that this estimate of the boundary layer thickness assumes a fully-rough flow regime (Section 3.1.6). Even though it is considered accurate enough for this study, it is recommended to further investigate its validity.

The initial wall cell size can be adjusted by varying the number of prism layers near the wall and their stretching.

Table 3.4: Initial wall cell size study (fouled Silicone 1)

<table>
<thead>
<tr>
<th>$\frac{y_0}{\delta_{lim}}$ (%)</th>
<th>Number of cells $\times 10^{-6}$</th>
<th>$R_F$ (N)</th>
<th>$C_{F,CFD} \times 10^3$</th>
<th>$C_{F,exp} \times 10^3$</th>
<th>$Error_{exp}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.2</td>
<td>0.88</td>
<td>14.314</td>
<td>7.999</td>
<td>12.6</td>
<td>-36.51%</td>
</tr>
<tr>
<td>6.8</td>
<td>0.82</td>
<td>20.913</td>
<td>11.688</td>
<td>12.6</td>
<td>-7.24%</td>
</tr>
<tr>
<td>11.3</td>
<td>0.78</td>
<td>23.41</td>
<td>13.086</td>
<td>12.6</td>
<td>3.86%</td>
</tr>
<tr>
<td>24.9</td>
<td>0.74</td>
<td>26.343</td>
<td>14.723</td>
<td>12.6</td>
<td>16.85%</td>
</tr>
</tbody>
</table>
Figure 3.20: Initial wall cell size requirement for each wall treatment model (CD-ADAPCO, 2017)

Table 3.5: Initial wall cell size study (unfouled Silicone 1)

<table>
<thead>
<tr>
<th>$\frac{y^\infty}{\delta_{lim}}$ (%)</th>
<th>Number of cells $\times 10^{-6}$</th>
<th>$R_F$ (N)</th>
<th>$C_{F,CFD} \times 10^3$</th>
<th>$C_{F,exp} \times 10^3$</th>
<th>$Error_{exp}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.9</td>
<td>0.9</td>
<td>6.281</td>
<td>3.511</td>
<td>3.666</td>
<td>-4.24%</td>
</tr>
<tr>
<td>14.6</td>
<td>0.86</td>
<td>6.499</td>
<td>3.632</td>
<td>3.666</td>
<td>-0.92%</td>
</tr>
<tr>
<td>25.4</td>
<td>0.8</td>
<td>6.641</td>
<td>3.711</td>
<td>3.666</td>
<td>1.25%</td>
</tr>
<tr>
<td>35.3</td>
<td>0.78</td>
<td>6.654</td>
<td>3.719</td>
<td>3.666</td>
<td>1.45%</td>
</tr>
</tbody>
</table>

Note on the derivation of $\delta_{lim}$

A turbulent boundary layer thickness at a distance $x$ from the leading edge of a plate can be approximated as:

$$\delta = \frac{0.16x}{(R_{n,x})^{\frac{1}{5}}}$$

(3.14)

The local skin friction at the location $x$ can also be approximated as (Faltinsen,
Combining Equations 3.14 and 3.15, we have

\[ \delta = 5.926x \cdot C_{f,x} \] (3.16)

By definition of the (total) frictional resistance coefficient \( C_F \), the following relationship holds:

\[ \delta = 5.926x \cdot C_{f,x} < 5.926x \cdot C_F \] (3.17)

Which leads us to an upper estimate of the boundary layer thickness \( \delta_{lim} \) at half-length of the plate - or ship - given in Equation 3.12.

### 3.3 Validation Study

The results of the simulations were validated against the experimental data published by Schultz (2004). Table 3.6 shows that there is excellent agreement between the simulations and the real experiment with the error in all cases not exceeding \( \pm 4\% \).

### 3.4 Results/Prediction

A prediction study was made for the frigate hull form described in Table B.1 and Figure 3.21. The mesh and time step of the simulations were set up with the Estimated Hull Performance add-on of Star CCM+, producing meshes similar to the validation study (note the volumetric refinements in Figures 3.23 and
3.6). The physics and wall treatment were set up identical to the validation study.

The analysis has been restricted to fouling conditions below FR-60 (small calcareous fouling) since ships are rarely let fouled beyond this condition. The detrimental effect of fouling can be seen in Figures 3.25 and 3.26. As expected, the following observations can be made:

- The rougher the surface, the greater the increase in frictional resistance;
- The frictional resistance increases with speeds for a given roughness condition;
- The frictional resistance doubles at 30 kts for the small calcareous fouling
The relative increase in total resistance decreases at high speeds due to the sharp increase in the contribution of wave resistance at $F_n \approx 0.3$.

It is of note that the results obtained here are close to the predictions of Schultz (2007) for a FFG-7 frigate (120 m long).

---

Figure 3.21: Hull form of a frigate (MAXSURF)

Figure 3.22: Domain view for the prediction study
Figure 3.23: Overall mesh for the prediction study

(a) Front view
(b) Side view
(c) Top view

Figure 3.24: Frigate mesh
Figure 3.25: Variation of resistance with speed and fouling condition

(a) $R_F$ against velocity

(b) $R_T$ against velocity

(c) % Increase of $R_F$

(d) % Increase of $R_T$
Figure 3.26: Variation of resistance coefficients with speed and fouling condition

(a) $C_F$ against Froude number

(b) $C_T$ against Froude number

(c) % Increase of $C_F$

(d) % Increase of $C_T$
Chapter 4

MATLAB Code

This section presents the procedure implemented in a MATLAB code to find the additional engine power needed to overcome fouling at a given speed. Then, the calculated rough frictional resistance coefficient, which is key to estimating the hydrodynamic penalty of biofouling, is compared against experimental results. Finally, the procedure to determine the best time to clean a hull has been presented. An analysis of the penalty for various ship types has also been discussed.

4.1 Procedure to Find Added Power Due to Fouling

The procedure to find the required propulsive power of a ship can be adapted to find the extra power required when the hull is fouled. The assumptions are the same, i.e. the frictional resistance is a function of the wetted surface and the speed. The fouling condition needs to be considered since it affects the frictional
resistance through the roughness function.

Consequently, the inputs of the code are:

- the roughness height \( k \) based on Equations 2.14a, 2.14b, 2.14c or Table 2.3,
- the velocity of the ship \( U \) (this could be a range in vector form),
- the length of the ship \( L \), and
- the wetted surface area of the ship \( S \).

From these inputs, the added power due to fouling to keep the ship sailing at \( U \) is calculated as follows:

(e) the Reynolds number of the ship \( R_n = \frac{U \cdot L}{\nu} \);

(f) The corresponding \( C_{F,r} \) is found using steps (b) to (d) as described in the similarity scaling procedure (section 2.4). Step (a) is omitted since the roughness function is defined by Equation 2.14;

(g) \( C_{F, clean} \) for a clean coating is found using \( k_{clean} \) from Equation 2.14a or Table 2.3;

(h) \( \Delta C_F = C_{F,r} - C_{F, clean} \);

(i) \( \Delta P_E = \frac{1}{2} \rho \cdot \Delta C_F \cdot S \cdot U^3 \);

(j) \( \Delta P_B = \frac{\Delta P_E}{\eta} \).
4.2 Calibration and Validation of the Code

As already mentioned, application of Granville similarity scaling law requires the definition of $\Delta U^+$ or reference points. Here, the reference points are chosen from the towing tank experiments of Schultz (2004).

Firstly, an appropriate smooth friction line had to be adopted. It is seen in Figure 4.1 that the Schoenherr friction line (Equation 2.6) correlates more to the measurements than the ITTC-1957 friction line (Equation 2.7). This is in line with the recommendation of Schultz (2004) and is consistent with the idea that the ITTC formula is a model-ship correlation line. Therefore, the Schoenherr friction line was adopted for all predictions. Note that further comparison of the predictions with both friction lines confirmed this choice as the best.

![Figure 4.1: Comparison of ITTC, Schoenherr friction lines and experimental $C_F$ (experimental data from Schultz (2004))](image)

Secondly, the predictions were made for $L_1 = L_2 = 1.52m$ in order to ensure that when no offset of length is made ($\log \left( \frac{L_2}{L_1} \right) = 0$), the reference points are
obtained.
For all predictions of unfouled coatings, the margin of error between predicted and measured frictional coefficients $C_F$ is $\pm 2\%$ (Figure 4.2). For the predictions of barnacle-fouled surfaces, it was found that using $k$ as defined in Equation 2.14c yielded errors of up to $13\%$ in frictional resistance coefficient. For this reason, the roughness length scale was modified as shown in Equation 4.1 which yielded better results:

$$k = 0.042R_t\sqrt{\% \text{Barnacle fouling}}$$  \hspace{1cm} (4.1)

The calculated error in this case was $\pm 3\%$. For the surface covered with slime only, the predicted $C_F$ had a maximum error of $6\%$. No attempt was made to change Equation 2.14b due to the lack of experimental data on slime-only covered surfaces. Figure 4.2 shows the excellent agreement between the predicted and the measured values of $C_F$.

It is interesting to note that the modified roughness length scale (Equation 4.1) agreed well with the classic pontoon experiment of Kempf (1937) where he conducted experiments at the Hamburg Model Basin (HSVA) tank to assess the impact of macrofouling on the resistance of ships (Figure 4.3). The maximum noted error in $C_F$ is $\pm 4\%$ when using Equation 4.1. However, it rises to $9\%$ when using Equation 2.14c.

At this point, it is relevant to note that there is excellent agreement between the predictions of the code which, as a reminder, is based on turbulent flow similarity scaling (Section 2.4), and the CFD predictions as evidenced by Figure 4.4.
Figure 4.2: Comparison of experimental and calculated $C_F$ (experimental data from Schultz (2004))

Figure 4.3: Comparison of experimental and calculated $C_F$ (experimental data from Kempf (1937))
Figure 4.4: Comparison of resistance predictions of MATLAB against CFD for a frigate (wave resistance calculated as per Holtrop and Mennen (1982) and Holtrop (1984) method)
4.3 Biofouling Tolerability Prediction

4.3.1 Method

If the added power due to fouling is known for a ship having a specific operational profile and a specific coating system, the best time to clean a sister ship under the same conditions (or the same ship during her next period of activity) can be inferred.

Firstly, the roughness height for the entire hull after a given period of activity must be assessed. Hulls of ships are almost never covered with uniform fouling. For example, the uneven distribution of light or the difference in flow characteristics along the hull affects fouling species distribution (INTERTANKO, 2016; Swain and Lund, 2016). However, when analyzing the roughness of a hull, it is possible to subdivide the hull in $m$ zones, visually assess the fouling condition of each zone, and determine a weighted average roughness height for the entire hull as

$$k = \frac{\sum_{j=1}^{m} \left(\frac{\text{area}_j}{2}\right) \cdot k_j}{100},$$

where $\text{area}_j$ and $k_j$ are respectively the percent relative area and the roughness height of each zone.

Secondly, the added power due to fouling is determined using the procedure described in section 4.1. To achieve this, the type of function that governs the increase of fouling over time has to be assumed from observation (linear, ascendant sinusoid, square-root, etc.). For example, Aertssen (1966) observed that the added power due to fouling on MV Jordaens followed nearly a square-root curve, which suggests a linear roughness increase over time; after several sea trials in temperate waters, Lewthwaite et al. (1985) observed that the increase in local skin friction $C_f$ due to slime followed an ascendant sinusoid (seasonality effect), which suggests likewise for the roughness increase over time; or a piece-
wise linear function could be adopted to account for an accelerated growth of fouling in ports. It is expected that the governing function would be dependent upon the coating system and the operational profile of the vessel (geographic area of operation, time at berth, speed, etc.).

Finally, the time $T$ at which the cumulative cost of fouling overcomes the cleaning cost is determined using the following steps:

(k) The function $k(t)$ is defined based on the time spent at sea $T_{sea}$, and discretized with $n + 1$ data points $k_i = k(t_i), i = 0, \ldots, n$;

(l) $\Delta P_B(t_i) = f(k_i)$ is calculated using steps (e) to (j) described above (section 4.1);

(m) The sum of all added power due to fouling over time is $K_F(t) = \int \Delta P_B dt$, which by trapezoidal rule of integration can be approximated as $K_F = \sum_{i=1}^{n} \frac{(t_i - t_{i-1}) + \Delta P_B(t_i)}{2}$;

(n) The cumulative extra fuel consumption over time is $FC(t) = SFOC \cdot K_F(t)$, where $SFOC$ is the specific fuel oil consumption of the engine;

(o) The cumulative cost of extra fuel burned over time is $C_{fuel}(t) = c_{fuel} \cdot FC(t)$, where $c_{fuel}$ is the fuel price;

(p) Finally, $T$ is the solution of the equation: $C_{fuel}(t) = C_c$, where $C_c$ is the cleaning cost (equivalently, $T$ is determined by the intersection of the curve of added fuel cost and the cleaning cost line).

A summary of the procedure is presented in Figure 4.5.
Average the roughness height at time $T_{sea}$

$$k = \sum_{j=1}^{m} \left( \frac{\% area_j \cdot k_j}{100} \right)$$

Define $k_i = k(t_i)$

Find $\Delta P_B(t_i) = f(k_i)$

Find the cost of extra fuel burned from $t_0$ to $t$

$$C_{fuel}(t) = c_{fuel} \cdot SFOC \cdot \int_{t_0}^{t} \Delta P_B \, dt$$

Solve for the threshold $T$

Figure 4.5: Summary of the biofouling tolerability prediction method for ships
4.3.2 Results

Prediction Study for a Cruise Ship

Let us consider a hypothetical scenario - the assumptions are summarized in Table 4.1 - involving a Cruise ship (Figure 4.6). After one year at sea without underwater cleaning, her fouling condition is described in Table 4.2 and Figure 4.7 (based on observations by Koka (2014)).

![Cruise ship model (MAXSURF)](image)

Figure 4.6: Cruise ship model (MAXSURF)

With the assumption that fouling rate follows a linear increase, we have $k(t) = a \cdot t + k_{\text{clean}}$, where $t$ denotes time in hours, and $k_{\text{clean}}$ is determined from Equation 2.14a.

We know $k(8760) = 40.4 \mu m$, so $k(t) = 0.004t + 5.1 \mu m$.

By discretizing $n = 100$, the resultant added power and added fuel cost over time are calculated using steps (k) to (p) and plotted in Figure 4.8. It shows that the extra fuel cost exceeds the cleaning cost after nearly 40 days or 1000 hours when the ship is sailing at the cruising speed of 20 knots. If an average operational ship speed of 15 knots is assumed, the threshold is overcome after nearly 60 days at sea or 1500 hours of operation. As expected, higher speeds mean more detrimental effect of fouling.
Table 4.1: Summary of assumptions for the Cruise ship prediction

<table>
<thead>
<tr>
<th>Ship</th>
<th>( L = 314m )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( S = 13347m^2 )</td>
</tr>
<tr>
<td></td>
<td>( U = 20kts )</td>
</tr>
<tr>
<td></td>
<td>( SFOC = 0.171 \text{ kg/kWh} )</td>
</tr>
<tr>
<td></td>
<td>( \eta = 0.75 )</td>
</tr>
<tr>
<td></td>
<td>( R_a = 30\mu m ) (applied fouling release coating)</td>
</tr>
</tbody>
</table>

**Fouling condition**

- In-dock observations: refer Table 4.2
- Fouling rate: follows a linear increase
- Time at sea before docking: 1 year (8760 hours)

**Market**

- Fuel price (IFO 380)=$370/tonne (Source: Ship and Bunker, 2017)
- Cleaning cost=$20,000 per occurrence

---

Table 4.2: Fouling condition

<table>
<thead>
<tr>
<th>Zone</th>
<th>% area relative to total area</th>
<th>Condition</th>
<th>Roughness height (Table 2.3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>17</td>
<td>Light biofilm</td>
<td>( k_1 = 17\mu m )</td>
</tr>
<tr>
<td>2</td>
<td>74</td>
<td>Light to moderate biofilm</td>
<td>( k_2 = 23.5\mu m )</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>Moderate biofilm and / or green weed</td>
<td>( k_3 = 30\mu m )</td>
</tr>
<tr>
<td>4</td>
<td>6</td>
<td>Heavy green weed</td>
<td>( k_4 = 320\mu m )</td>
</tr>
<tr>
<td></td>
<td>Weighted average roughness height</td>
<td></td>
<td>( k = 40.4\mu m )</td>
</tr>
</tbody>
</table>

57
Figure 4.7: Percent distribution of fouling on fouling release coating observed on 5 cruise ships (Koka, 2014)

Figure 4.8: Added power and cumulative extra fuel cost for a cruise ship
The adverse effect of speed for a given fouling condition can be further evaluated in Figure 4.9. Under the earlier assumptions \( (k \approx 40\mu m) \), the added cost of fuel due to fouling in a single day at a speed of 25 knots is over half that of the cleaning cost, i.e. the cleaning cost would be overcome in two days of operations. If the propulsive power is set to a constant value however, there will be an induced reduction in speed which can be quite significant.

![Figure 4.9: Added daily fuel consumption and costs for a cruise ship](image)

**Figure 4.9: Added daily fuel consumption and costs for a cruise ship**

**Prediction of Biofouling Tolerability for Various Ship Types**

In this section, the tolerability of fouling for different ship types is analyzed. Three MAXSURF model bare hull forms have been selected representing respectively a Cruise ship, a Frigate and a Trawler (Figures 4.10 to 4.12). The main particulars of each hull are listed in Table 4.3, and the assumptions for the tolerability prediction are the same as in Table 4.1. The cleaning costs ranges
are typical values given by ABS (2013) and McMillan and Jarabo (2013).

Table 4.3: Ship particulars

<table>
<thead>
<tr>
<th></th>
<th>Cruise ship</th>
<th>Frigate</th>
<th>Trawler</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{WL}(m)$</td>
<td>314.412</td>
<td>80.000</td>
<td>24.028</td>
</tr>
<tr>
<td>$B(m)$</td>
<td>38.595</td>
<td>12.521</td>
<td>5.640</td>
</tr>
<tr>
<td>$T(m)$</td>
<td>8.800</td>
<td>3.000</td>
<td>1.500</td>
</tr>
<tr>
<td>$\Delta(t)$</td>
<td>67396</td>
<td>1378</td>
<td>114.5</td>
</tr>
<tr>
<td>$C_B$</td>
<td>0.616</td>
<td>0.447</td>
<td>0.550</td>
</tr>
<tr>
<td>$C_M$</td>
<td>0.901</td>
<td>0.827</td>
<td>0.858</td>
</tr>
<tr>
<td>$S(m^2)$</td>
<td>12985.395</td>
<td>924.926</td>
<td>144.950</td>
</tr>
</tbody>
</table>

Figure 4.10: Hull form of a cruise ship (MAXSURF)

Figure 4.11: Hull form of a frigate (MAXSURF)
Figure 4.12: Hull form of a trawler (MAXSURF)

Figure 4.13: Fouling tolerability analysis for a cruise ship
Figure 4.14: Fouling tolerability analysis for a frigate

Figure 4.15: Fouling tolerability analysis for a trawler
4.4 Percent Fouling Penalty Calculation

4.4.1 Method

In this section, the fouling penalty is analyzed with respect to the total power requirement. Indeed, it is often convenient to analyze the hydrodynamic impact of biofouling in terms of percentage, especially to facilitate comparisons among different ships. Though noted $\%(\Delta P_E)$, it should be borne in mind that the percent penalty is the same whatever the standpoint in the propulsion line i.e.:

\[
\%(\Delta P_B) = \frac{P_{B,r} - P_{B,s}}{P_{B,s}} \cdot 100 = \frac{\eta P_{B,r} - \eta P_{B,s}}{\eta P_{B,s}} \cdot 100 = \frac{P_{E,r} - P_{E,s}}{P_{E,s}} \cdot 100
\]

\[
\%(\Delta P_B) = \%(\Delta P_E) \tag{4.2}
\]

Now combining with Equation 2.8, we have

\[
\%(\Delta P_E) = \frac{R_{T,r} \cdot U - R_{T,s} \cdot U}{R_{T,s} \cdot U} \cdot 100
\]

\[
\%(\Delta P_E) = \frac{\frac{1}{2} \rho (C_{T,s} + \Delta C_F) SU^3 - \frac{1}{2} \rho C_{T,s} SU^3}{\frac{1}{2} \rho C_{T,s} SU^3} \cdot 100
\]

\[
\%(\Delta P_E) = \frac{\Delta C_F}{C_{T,s}} \cdot 100 \tag{4.3}
\]

In other words,

\[
\%(\Delta P_E) = \frac{\Delta C_F}{C_{T,s}} \cdot 100
\]

(4.4)

Considering that wave resistance is the major component of the residuary resistance, we shall derive from ITTC-1978 that

\[
C_{T,s} = (1 + k_1)C_F + C_R + C_A \approx (1 + k_1)C_F + C_W + C_A \tag{4.5}
\]
Rewriting Equation 4.4,

\[
\% (\Delta P_E) = \frac{\Delta C_F}{(1 + k_1)C_F + C_W + C_A} \cdot 100 \tag{4.6}
\]

The MATLAB code makes use of Holtrop (1984) and Holtrop and Mennen (1982) formulations for form factor, wave resistance and correlation allowance (respectively Equations A.18, A.4 and A.19) to assess the percent penalty.

### 4.4.2 Variability/Similarity of Fouling Penalty with Ship Types

In this section, the variability of the percent fouling penalty with ship types is analyzed. It seems that there is in fact some similarity among ship types when they are scaled with respect to their Froude number.

The percent penalty for all three ship types are plotted against velocity in Figure 4.16. Interestingly, when plotted against \( F_n \), the percent penalty of two different ships seems strongly correlated (especially at \( F_n \) less than 0.3) as shown in Figures 4.17 and 4.18. From this study, the correlation has been found to be:

\[
\% (\Delta P_E)_1 = \% (\Delta P_E)_2 \cdot (1 + \alpha \log \left( \frac{R_{n,1}}{R_{n,2}} \right)) \cdot \frac{(1 + k_1)_2}{(1 + k_1)_1} \tag{4.7}
\]

Where \( \alpha = 2.25e^{-3.7 \frac{R_n}{k}} \).

Equation 4.7 has been found by noting the following:

- The percent penalty as given by Equation 4.6 is a function of \( C_W, C_F \) and \( \Delta C_F \), hence also a function of \( F_n, R_n \) and \( k \) (roughness height) - neglecting the effects of \( C_A \);
The gap in percent penalty plotted against \( F_n \) for the various ship types is inversely proportional to the roughness height \( k \) (Figure 4.17).

At the same velocity, Equation 4.7 can be rewritten as:

\[
\%(\Delta P_E)_1 = \%(\Delta P_E)_2 \cdot \left(1 + \alpha \log \left( \frac{L_1}{L_2} \right) \right) \cdot \frac{(1 + k_1)_2}{(1 + k_1)_1}
\]  

(4.8)

If the form factors are neglected, the relationship is simplified as follows:

\[
\%(\Delta P_E)_1 = \%(\Delta P_E)_2 \cdot \left(1 + \alpha \log \left( \frac{L_1}{L_2} \right) \right)
\]  

(4.9)
(a) Clean coating \((k = 2\mu m)\)

(b) Light slime \((k = 17\mu m)\)

(c) Heavy slime \((k = 60\mu m)\)

(d) Small calcareous fouling or weed \((k = 320\mu m)\)

(e) Medium calcareous fouling \((k = 1mm)\)

(f) Heavy calcareous \((k = 3.5mm)\)

Figure 4.16: Percent penalty for various fouling conditions and ship types
(a) Clean coating \((k = 2\mu m)\)

(b) Light slime \((k = 17\mu m)\)

(c) Heavy slime \((k = 60\mu m)\)

(d) Small calcareous fouling or weed \((k = 320\mu m)\)

(e) Medium calcareous fouling \((k = 1mm)\)

(f) Heavy calcareous \((k = 3.5mm)\)

Figure 4.17: Percent penalty vs \(F_n\) for various fouling conditions and ship types
Figure 4.18: Scaled percent penalty vs $F_n$ for various fouling conditions and ship types.
4.5 Discussion

An effective numerical modelling of biofouling has the potential to improve current biofouling management practices. Two applications of such attempt with a MATLAB code have been presented that respectively assess the tolerability of biofouling and its hydrodynamic penalty.

The first application shows that it is possible to evaluate when biofouling is no longer tolerable in a given context considering its cumulative economic penalty. This information would be useful in a bigger maintenance strategy to complement other fouling management tools since none of them is perfect. Antifouling coating systems cannot achieve 100% immunity; monitoring hull performance is prone to error due to the influence of weather, loading condition, currents, etc. (Munk et al., 2009; Townsin, 2003); hull cleaning is unproductive if it does not occur at a proper timing.

Today, it is very common to determine when hull cleaning should occur by monitoring hull performance or visual observation. However, this reactive approach may not be suitable in all circumstances. The prediction of fouling tolerability fits well into a proactive fouling management strategy that some studies advocated (Tribou and Swain, 2010). A proactive approach allows for effective planning. For example, hull cleaning may take one or two days and ship inactivity may be inconvenient if not planned ahead of time. It further helps ship operators ensure that the money spent into fouling precautions is optimally invested. A coating system, for instance, will be selected based upon its resistance to fouling against its ability to sustain multiple cleaning. Recoating costs may also be put into balance.

An interesting aspect of the current numerical code is its flexibility as it en-
deavors to account for the specifics of each case. For example, fouling rate is
determined based on at least one observation of the fouling condition of the ship
considered. If such an observation is made at the end of an operational cycle -
and in between, if many observations are possible - an appropriate knowledge of
the marine growth can be inferred. The fouling rate governing function can also
be adapted to consider the duration of the ship at berth. Moreover, the increase
in resistance for a given fouling condition is dependent on vessel type and speed
which, in the proposed method, are considered as input. As demonstrated by
the analysis of three different ship types, a universal drag factor does not reflect
the specifics of each case.

The second application uses empirical formulations of the wave resistance of
ships to assess the percent penalty incurred by biofouling. This traditional way
of expressing the penalty is useful to compare the vulnerability of different ships
to fouling. The analysis shows that for the same fouling condition and speed,
bigger ships have a greater penalty. This is not surprising as the contribution of
frictional resistance is proportional to the wetted surface area. One would tend
to clean bigger ships more often than smaller ones. However, since cleaning
costs are relatively smaller for bigger ships, it is best to heed the tolerability
prediction method in order to make thoughtful decisions.

It is also interesting to note the correlation of the penalty among ship types.
It is seen that a good approximation of the fouling penalty for any ship can be
obtained from Equation 4.7. Certainly, this correlation needs further investiga-
tion.
Chapter 5

Conclusion

The present work has shown the implementation of two numerical methods to assess the impact of biofouling on the hydrodynamics of ships. Both methods are validated against experimental data available in the literature.

The first method is a CFD assessment of the additional drag incurred by biofouling. The merit of this approach lies in the use of a fully-defined ship model to simulate the hydrodynamic impact of fouling. It has the potential to reduce the need for costly experiments on ships. If appendages and superstructure are modelled, an accurate prediction of the residuary resistance can be made, leading also to an accurate prediction of the percent penalty. The validity of the model presented could be further investigated and compared against experimental results.

The second method developed in MATLAB is a flexible, much faster instrument that could be used to predict when the penalty due to biofouling on a ship overcomes the cleaning cost. The tool aims to account for the specifics of each situation through at least one observation of the fouling condition of a given ship.
A direct measure of the economic penalty in dollar amount is given as well as a relative measure in percentage. With further improvements, the code might be able to estimate the penalty from a pictorial representation of a fouled hull. This would certainly ease in-dock assessments and improve proactive management of ship operations.

It goes without saying that the reliability of the methods herein presented is tied to the reliability of the mathematical models used. Several studies have pointed out the complexity of characterizing and modelling marine fouling, especially biofilms. In particular, Table 2.3 and equation 2.14 need much supportive work. In the end, if it is true that the present work presents two distinct methods that can serve as good approximation tools, research still needs to be done for more accurate predictions.
Chapter 6

Recommendations for future research

The study presented in this thesis is an attempt to model biofouling and estimate its penalty. Both numerical methods discussed could be further improved. It is also possible to push the analysis further into areas not explored here. The CFD analysis has been limited by the time and the computational resources available. The precision of the predictions could be improved by creating finer meshes. With that said, finer meshes will only produce meaningful results if the transition from prism layer cells to core mesh cells is smooth, and if the simulations are run for enough time to achieve convergence. From the author’s point of view, no less than 1.5 million cells should be used.

Another area of improvement in CFD is the wall treatment. From the author’s analysis, the first cell size has to relate to the boundary layer thickness. An alternative equation to Equation 3.12 could be used to assess the boundary layer thickness. Other recent studies have reported that $y^+$ should be greater
than $k^+$ for accurate results (Demirel et al., 2017). This statement could also be further investigated.

Areas not explored in this study that are worthy of interest are the use of CFD to assess the effectiveness of partial hull cleaning. Through the input of various roughness heights along the hull, it is possible to analyze the benefit of having some portions of the hull cleaned instead of others (bow, shoulder, stern, etc.). Also, it is necessary to ascertain the mathematical formulations used (roughness function, fouling reference table) through experiments. In particular, biofilms could be analyzed as viscoelastic fluids instead of solid bodies (sand roughness).

The code implemented in MATLAB, which is also available as a MATLAB application, could be refined to incorporate more parameters. Such a flexible tool, as demonstrated by its use in the tolerability prediction method, is capable of producing prompt results. From the author’s point of view, it holds a lot of potential. However, it definitely needs experts’ advice to produce useful and more realistic results.
References


IMO (2011). Guidelines for the control and management of ships' biofouling to minimize the transfer of invasive aquatic species.

INTERTANKO (2016). INTERTANKO guide to modern antifouling systems and biofouling management.


Appendix A

Calculation of wave resistance

In this study, the wave resistance and the wave resistance coefficient were calculated as proposed by Holtrop and Mennen (1982) and Holtrop (1984) regression equations as follows:

- $F_n \leq 0.4$
  \[
  R_W = c_1 c_2 c_5 \rho g \nabla e^{m_1 F_n^d + m_4 \cos (\lambda F_n^{-2})} \quad (A.1)
  \]

- $F_n \geq 0.55$
  \[
  R_W = c_{17} c_2 c_5 \rho g \nabla e^{m_3 F_n^d + m_4 \cos (\lambda F_n^{-2})} \quad (A.2)
  \]

- $0.4 < F_n < 0.55$
  \[
  R_W = R_{W(F_n=0.4)} + (10F_n - 4) \left[ R_{W(F_n=0.55)} - R_{W(F_n=0.4)} \right] \frac{1.5}{1.5} \quad (A.3)
  \]

And finally, $C_W$ is given as:

\[
C_W = \frac{R_W}{\frac{1}{2} \rho S U^2} \quad (A.4)
\]
The coefficients are given as:

\[
c_1 = 222310 c_7^{3.78613} \left( \frac{T}{B} \right)^{1.07961} (90 - i_E)^{-1.37565} \quad (A.5)
\]

\[
c_2 = e^{-1.89} \sqrt{c_3} \quad (A.6)
\]

\[
c_3 = \frac{0.56 A_{HT}^{1.5}}{BT(0.31 \sqrt{A_{BT}} + T_F - h_B)} \quad (A.7)
\]

\[
c_7 = \begin{cases} 
0.229577 \left( \frac{B}{T} \right)^{0.33333}, & \text{for } \frac{B}{T} < 0.11 \\
\frac{B}{L}, & \text{for } 0.11 \leq \frac{B}{T} \leq 0.25 \\
0.5 - 0.0625 \frac{L}{B}, & \text{for } \frac{B}{T} > 0.25 
\end{cases} \quad (A.9)
\]

\[
c_{15} = \begin{cases} 
-1.69385, & \text{for } \frac{L^3}{\nu} < 512 \\
-1.69385 + (\frac{L}{\nu^{\frac{1}{3}}} - 8)/2.36, & \text{for } 512 \leq \frac{L^3}{\nu} \leq 1726.91 \\
0, & \text{for } \frac{L^3}{\nu} > 1726.91 
\end{cases} \quad (A.10)
\]

\[
c_{16} = \begin{cases} 
8.07981C_P - 13.8673C_P^2 + 6.984388C_P^3, & \text{for } C_P < 0.8 \\
1.73014 - 0.7067C_P, & \text{for } C_P > 0.8 
\end{cases} \quad (A.11)
\]

\[
c_{17} = 6919.3C_M^{1.3346} \left( \frac{\nabla}{L^3} \right)^{2.00977} \left( \frac{L}{B} - 2 \right)^{1.40692} \quad (A.12)
\]

\[
m_1 = \frac{0.0140407L}{T} - \frac{1.75254\nabla^{\frac{1}{3}}}{L} - \frac{4.79323B}{L} - c_{16} \quad (A.13)
\]

\[
m_3 = -7.2035 \left( \frac{B}{L} \right)^{0.326869} \left( \frac{T}{B} \right)^{0.605375} \quad (A.14)
\]

\[
m_4 = 0.4c_{15}e^{-0.034F_a^{-3.29}} \quad (A.15)
\]

\[
d = -0.9 \quad (A.16)
\]
The use of these expressions requires that the total resistance be calculated by adopting the form factor concept (3D extrapolation procedure) which can be calculated as follows:

\[
1 + k_1 = 0.93 + 0.487118 c_{14} \left( \frac{B}{L} \right)^{1.06806} \left( \frac{T}{L} \right)^{0.46106} \left( \frac{L}{L_R} \right)^{0.121563} \cdot \left( \frac{L^3}{V} \right)^{0.36486} (1 - C_P)^{-0.604247}
\]  

\[\text{(A.18)}\]

Where the length of run \(L_R\) is defined as \(L_R = L(1 - C_P + \frac{0.06 C_P - l}{4 C_P - 1})\).

The coefficient \(c_{14}\) accounts for the shape of the stern; assuming a normal stern shape, \(c_{14} = 1\).

The method also suggests formulations for the calculation of \(C_A\) and the wetted surface area \(S\) as follows:

\[
C_A = 0.006(L + 100)^{-0.16} - 0.00205 + 0.003 \sqrt{\frac{L}{7.5}} C_B^4 c_2 (0.04 - c_4)
\]  

\[\text{(A.19)}\]

Where \(c_4 = T/L\) when \(T/L \leq 0.04\) or \(c_4 = 0.04\) otherwise.

\[
S = L(2T + B) \sqrt{C_M} (0.453 + 0.4425 C_B - 0.2862 C_M - \frac{0.003467 B}{T} \frac{A_{BT}}{C_B} + 0.3696 C_{WP}) + 2.38
\]  

\[\text{(A.20)}\]
# Appendix B

## Ship Particulars

Table B.1: Ship particulars

<table>
<thead>
<tr>
<th></th>
<th>Cruise ship</th>
<th>Frigate</th>
<th>Trawler</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{WL}(m)$</td>
<td>314.412</td>
<td>80.000</td>
<td>24.028</td>
</tr>
<tr>
<td>$B(m)$</td>
<td>38.595</td>
<td>12.521</td>
<td>5.640</td>
</tr>
<tr>
<td>$T(m)$</td>
<td>8.800</td>
<td>3.000</td>
<td>1.500</td>
</tr>
<tr>
<td>$\nabla \ (m^3)$</td>
<td>65956.907</td>
<td>1344.657</td>
<td>111.991</td>
</tr>
<tr>
<td>$\Delta(t)$</td>
<td>67396</td>
<td>1378</td>
<td>114.5</td>
</tr>
<tr>
<td>$C_B$</td>
<td>0.616</td>
<td>0.447</td>
<td>0.550</td>
</tr>
<tr>
<td>$C_M$</td>
<td>0.901</td>
<td>0.827</td>
<td>0.858</td>
</tr>
<tr>
<td>$C_P$</td>
<td>0.687</td>
<td>0.551</td>
<td>0.633</td>
</tr>
<tr>
<td>$C_{WP}$</td>
<td>0.802</td>
<td>0.718</td>
<td>0.858</td>
</tr>
<tr>
<td>$S(m^2)$</td>
<td>12985.395</td>
<td>924.926</td>
<td>144.950</td>
</tr>
<tr>
<td>$i_E \ (\text{deg})$</td>
<td>9.9</td>
<td>7.9</td>
<td>26.3</td>
</tr>
<tr>
<td>$A_T(m^2)$</td>
<td>0</td>
<td>1.653</td>
<td>0.3</td>
</tr>
<tr>
<td>$A_{BT}(m^2)$</td>
<td>28.111</td>
<td>0</td>
<td>0.086</td>
</tr>
<tr>
<td>$h_B(m)$</td>
<td>4.718</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>
Appendix C

Description of Simulation Models (CD-ADAPCO, 2017)

Table C.1: Meshers

<table>
<thead>
<tr>
<th>Mesher</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface Remesher</td>
<td>Remeshes the initial surface to provide a quality discretized mesh that is suitable for CFD</td>
</tr>
<tr>
<td>Trimmed Cell Mesher</td>
<td>Generates a volume mesh by cutting a hexahedral template mesh with the geometry surface</td>
</tr>
<tr>
<td>Prism Layer Mesher</td>
<td>Adds prismatic cell layers next to wall boundaries</td>
</tr>
</tbody>
</table>
### Table C.2: Physical models

<table>
<thead>
<tr>
<th>Model</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>All $Y^+$ wall treatment</td>
<td>Hybrid wall treatment using a blending approach suitable for intermediate near-wall cell resolution</td>
</tr>
<tr>
<td>Cell Quality Remediation</td>
<td>Improves the robustness of the solution by identifying and remedying poor-quality cells</td>
</tr>
<tr>
<td>Eulerian Multiphase</td>
<td>Defines the fluids involved in the flow (air, water)</td>
</tr>
<tr>
<td>Exact Wall Distance</td>
<td>Makes an exact projection of the distance of cells near the wall to be used to account for near wall effects in no-slip boundary condition</td>
</tr>
<tr>
<td>Gradients</td>
<td>Defines the method of gradients calculation for scalar values, flux, etc.</td>
</tr>
<tr>
<td>Gravity</td>
<td>Accounts for the effects of gravitational acceleration</td>
</tr>
<tr>
<td>Implicit unsteady</td>
<td>Controls the update at each physical time for the calculation</td>
</tr>
<tr>
<td>K-Omega turbulence</td>
<td>Provides closure of the RANS equations using $K$ and $\omega$</td>
</tr>
<tr>
<td>Multiphase Equation of State</td>
<td>Computes the density and density derivatives of each fluid phase</td>
</tr>
<tr>
<td>Multiphase Interaction</td>
<td>Defines the action of one fluid phase upon another</td>
</tr>
<tr>
<td>Multiphase Mixture</td>
<td>Considers mass, momentum and energy as mixture quantities</td>
</tr>
<tr>
<td>Segregated Volume Flux Based Flow</td>
<td>Solves each of the momentum equations in turn</td>
</tr>
<tr>
<td>SST (Menter) K-Omega</td>
<td>Provides closure of the RANS equations with the SST $K – \omega$ model</td>
</tr>
<tr>
<td>Three Dimensional</td>
<td>Considers a 3-D geometry</td>
</tr>
<tr>
<td>VOF Waves</td>
<td>Typically used for marine applications, it defines surface gravity waves</td>
</tr>
<tr>
<td>Volume of Fluid (VOF)</td>
<td>Simulates flow of several immiscible fluids</td>
</tr>
<tr>
<td>Model</td>
<td>Description</td>
</tr>
<tr>
<td>----------------------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Implicit unsteady</td>
<td>Controls the update at each physical time for the calculation</td>
</tr>
<tr>
<td>6-DOF solver</td>
<td>Computes the forces and moments acting on a 6-DOF body</td>
</tr>
<tr>
<td>6-DOF motion</td>
<td>Solves for the translational and angular motions of a 6-DOF body based on the forces and moments acting on it</td>
</tr>
<tr>
<td>Partitioning</td>
<td>Controls how the solver partitions the domain</td>
</tr>
<tr>
<td>Wall distance</td>
<td>Controls the wall distance computation</td>
</tr>
<tr>
<td>VOF wave zone distance</td>
<td>Controls the wave zone distance computation especially in regions where the damping option is activated</td>
</tr>
<tr>
<td>K-Omega turbulence</td>
<td>Controls the solution of the turbulence transport equations</td>
</tr>
<tr>
<td>K-Omega turbulent viscosity</td>
<td>Controls the computation of the turbulent viscosity</td>
</tr>
</tbody>
</table>
Appendix D

CFD validation study

Herein, the details of the flow near the wall of the fouled Silicone 1 plate are shown as an example. The friction velocity $U_f$, the roughness Reynolds number $k^+$ and the normalized distance from the wall $y^+$ are calculated as described in section 2.3. Wall $Y+$ is calculated by STAR CCM+ as $Y+ = y_0 \frac{U_f}{v}$ where $y_0$ is the size of the initial wall cell.

Figure D.1: Wall shear stress $\tau_w$ over the plate ($Pa = N/m^2$)
Figure D.2: Friction velocity $U_\tau$ over the plate

Figure D.3: Roughness Reynolds number $k^+$ over the plate
Figure D.4: Non-dimensional distance from the wall \( y^+ \)

Figure D.5: Wall Y+ based on first cell size