Numerical investigation on the effect of homogenous roughness due to biofouling on ship friction resistance

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Abstract. Ships are subject to increased surface roughness due to the attachment of biofoulings on their hull. When the surface of a ship’s hull is rough, increased frictional resistance can be expected. A ship’s frictional resistance make up almost 80 – 85% of its total resistance. Therefore, it is crucial to maintain the ship’s frictional resistance value to a minimum. In this study, the effects of roughness length scale due to biofouling on friction resistance are investigated. To achieve reliable results, this study used the 3D DTMB 5415 model that was established as a benchmark study by ITTC. Roughness length scales representing biofoulings are applied to the model and analyzed by using the CFD software at a service speed, reaching a Froude Number of 0.28. Results of the simulation are compared and analysed to gain an understanding of the increased friction resistance value due to biofouling. For the smooth case, the results are in agreement with the towing test conducted by ITTC. In addition, friction resistance is found to be increasing along with the rise of the roughness length scale.

Keywords: CFD, DTMB 5415, ITTC, skin friction, surface roughness

1. Introduction

In accordance with the Global Maritime Axis (GMA) vision declared by the President of Indonesia, the country has been strengthening its navy force by purchasing various new vessels to answer the specific needs or roles required in the field. In the year 2021, the country has a growing list of navy ships, consisting of 7 frigates, 24 corvettes, five submarines, 179 patrol vessels, and ten mine warfare [1]. A recent purchase of two novel offshore patrol vessels (OPVs), ordered by the Ministry of Defense of Indonesia, has just begun its first production stages in August 2021 [2]. While this decision helps strengthen the navy, another essential thing to do is keep the vessels in prime condition. After a few years of operating, a vessel is bound to be interrupted in terms of resistance by the attachment of biofouling to its hull [3]. The term biofouling refers to an accumulation of unwanted aquatic growth that appears on a ship hull by utilizing their bioadhesion ability [4], [5]. Since the navy utilizes vessels with high-speed criterion, these vessels require higher power to tow the ship against its total resistance force at such velocity [6]. However, when a vessel is covered with biofouling, its resistance will increase. For instance, CFD prediction found the type of fouling imposed a 49% increase in ship frictional resistance of a KCS model when travelling at 24 knots [7]. This leads to even higher fuel consumption to make up for the required power lost to the effects of biofouling on its resistance. This intrusive phenomenon also causes a decline in the vessel’s traveling speed, increases the fuel consumption, and thus resulting in...
higher CO₂ emissions [8]. For instance, a ship’s frictional resistance increased for up to 80% due to the adhesion of a biofouling with only 1 mm height, causing a loss of speed by 15% [9]. In another study, it was found that with 5% increase of biofouling, the ship’s fuel consumption also increased by 17%, indirectly causing a 14% increase in greenhouse gases emissions [10]. Low speeds can become a trouble for navy vessels, specifically when the vessel needs to chase down foreign and unauthorized fishing boats that trespass the Indonesian waters, such as when KN Pulau Nipah 321, a patrol vessel with a maximum speed of 22 knots, chased down an unauthorized fishing boat in 2020 [11]. Therefore, it’s important to investigate the effect of biofouling on the resistance of warships.

A navy combatant model, David Taylor Model Basin (DTMB) 5415, was used in an international collaborative research project to provide database for surface-ship resistance and propulsion model-scale CFD prediction [12]. This preliminary US navy combatant vessel design had been tested by International Towing Tanks Convention (ITTC) in various towing tanks across the world at three Froude Number variations, including 0.1, 0.28 and 0.41. A report was later published to serve as a recommended procedure and guidelines on validating a numerical computation of a ship’s resistance, officially referring the DTMB 5415 model towing experiments as validation for CFD simulation [13]. The validation allows the researcher to evaluate the process used in a numerical investigation of ship resistance, by using the DTMB 5415 model first before continuing the study of another vessel in focus of the project. If the results conform well with ITTC, the simulation is valued as reliable and thus, the next simulation can be carried out. The effect of fouled hull would be easier to analyse when it can be compared to a smooth hull condition, as This also gives advantage when investigating the biofouling effect on ship resistance, as it’s much easier to measure the effect of fouled hull when compared to a verified smooth results of a benchmark model.

The difference between smooth hull and fouled hull is identified as the difference in surface roughness, which can be indicated by the roughness height. Biofoulings on ship hulls induces an increase of the hull’s roughness. In 1932, Nikuradse conducted a roughness experiment by investigating a pipe of turbulent flows, where the increase of the frictional resistance of the flow was measured by incorporating the pipe with sand grains to represent surface roughness and their effect [14]. This method continues to be used to investigate the effect of surface roughness due to its convenience of using equivalent sand-grain roughness \( (k_s) \) as a representation of roughness length-scale [15]. Due to the arbitrary form of biofouling, experimental studies using flat plates have been conducted to determine the appropriate value of \( k_s \) for specific roughness, ranging from anti-fouling coating to heavy calcarceous fouling [15], [16]. However, a recent study discovered a different \( k_s \) number for a light calcarceous fouling than the previously known \( k_s \) value for the same fouling type [15]. The use of flat plate as an experimental model is based on the assumption that the skin friction value of a flat plate is equivalent to that of a ship of the same length [17]. To simulate the effects of the roughness, we use the roughness function to modify the standard wall function [18].

Since CFD has been undergoing robust development, investigation on 3D ship models can be conducted to limit the spending on experiments. This study explores the change in a warship resistance due to biofouling attachment to a DTMB 5415 hull. It is hoped that this study could serve as a reference for further investigations on Indonesian warship resistance.

2. Methods
The methods that were used include determining the model of the ship, defining the suitable mathematical model consisting of RANS based equation and turbulence model, as well as defining the roughness function. The CFD simulation set-ups were then evaluated by using grid independence study and the results were later compared to empirical equations to gain independence, where the results conform well with the theory and previous experiments regarding this phenomenon. Furthermore, the value of drag are later analysed to obtain understanding on the effects of surface roughness on the ship’s hull, as well as other phenomena related to surface roughness.
2.1 Drag force

The total drag is comprised of two main components, namely the frictional and the pressure drag, depicted by $F_F$ and $F_P$ respectively. Frictional drag is subject to shear stress experienced by the surface of the object while the pressure drag is induced by the pressure distributed along the surface. Each drag force acts differently, where the shear stress acts tangential to the flow around the object, while the distributed pressure acts perpendicular to the flow around the object. The relationship between each drag component is depicted through Equation 1.

$$F_T = F_F + F_P$$  \hspace{1cm} (1)

A method to simplify the calculation and comparison between drag values is to convert them into non-dimensionalized forms. These forms allow comparison to be made between the drag of two different shapes. To convert the drag values to non-dimensional forms, the values are divided by dynamic pressure and wetted surface area of the models. Equation 2 shows the relationship between each non-dimensional component of drag force.

$$C_T = C_F + C_P$$  \hspace{1cm} (2)

Where $C_T$ depicts the drag force, $C_F$ depicts frictional drag, and $C_P$ depicts pressure drag all of which are in coefficient form. Coefficient forms of drag force are obtained by using Equation 3 as stated below.

$$C_T = \frac{F_T}{0.5 \rho V^2 A}$$  \hspace{1cm} (3)

where $\rho$ represents fluid density, $V$ is fluid velocity, and $A$ represents wetted surface area.

2.2 Model and description

This study used DTMB 5415, a displacement ship equipped with a bulbous bow with a purpose to reduce total resistance [17] as depicted in Figure 1. The dimensional characteristics of this model can be seen in Table 1. The ITTC conducted experiments using this model at a range of Froude Numbers as explained in [13]. However, to further understand the skin friction behaviour of the model at a low speed, the model was tested at a Froude Number of 0.28.

Figure 1 Lines plan of DTMB 5415

To account for the behaviour of frictional drag on flat and curved surfaces, this study also considered to investigate the effects of biofouling on flat plates. The results of both models were compared to each other to prove Froude’s assumption that the skin friction of flat plates and hull model are equal to each
other, when both models are of the same length. In this case, this assumption is useful to analyze the accuracy of the rough surface simulation [21].

### Table 1 Principal dimensions of DTMB 5415

| Parameter                          | Value | Units |
|-----------------------------------|-------|-------|
| Length Between Perependicular (LPP) | 5.720 | m     |
| Draught (T)                       | 0.248 | m     |
| Volume Displacement (V)           | 0.549 | m³    |
| Wetted Surface Area (WSA)         | 4.786 | m²    |

#### 2.3 Mathematical model

Fluid flow modelling was conducted using an incompressible unsteady RANS model, alongside the turbulence $k – \omega$ SST model. The finite volume approach was used to build the spatial discretization of the transport equations. To construct non-overlapping control volumes, the face-based method employed unstructured three-dimensional meshes with an arbitrary number of constitutive faces. Momentum conservation equations and pressure equation were utilised to calculate velocity field and pressure field subsequently [22]. Additional transport equations were discretized and solved using the same techniques as momentum equations in turbulent flows. In the mass and momentum equations illustrated in Equations 4 and 5, $v$ represents flow velocity, $\nabla$ represents mass transport, $\rho$ demonstrates fluid density, $\mu$ represents viscous stress, $f$ shows external force, and $t$ represents time.

$$\frac{\partial \rho}{\partial t} + \nabla (\rho u) = 0$$  \hspace{1cm} (4)

$$\rho \left( \frac{\partial V}{\partial t} + v \cdot \nabla v \right) = \nabla p + \mu \nabla^2 v + f$$  \hspace{1cm} (5)

#### 2.3.1. $k – \omega$ SST turbulence model

A RANS-based equation developed by Wilcox using the $k – \omega$ turbulence model was used to simulate the turbulence effects in near-war flow. The model shows more numerical stability, notably in the viscous sublayer near the wall, when compared to $k – \varepsilon$ turbulence model. High values of $\omega$ around the wall regions allow the model to use non-explicit wall-damping functions, unlike the $k – \varepsilon$ and other two-equation models. In the numerical wall boundary conditions, the distance from the wall to the nearest point must be determined. The transport model equations for both $k$ and $\omega$ scalar turbulence scales are specified by Equations 6 and 7.

$$\frac{\partial \rho K}{\partial t} + \frac{\partial}{\partial x_j} \left( \rho u_j K + \sigma^* \mu_t \frac{\partial K}{\partial x_j} \right) = \tau_{ij} S_{ij} - \beta^* \rho \omega K$$  \hspace{1cm} (6)

$$\frac{\partial \rho \omega}{\partial t} + \frac{\partial}{\partial x_j} \left( \rho u_j \omega - \sigma \mu_t \frac{\partial \omega}{\partial x_j} \right) = a \frac{\omega}{K} \tau_{ij} S_{ij} + \beta \rho \omega^2$$  \hspace{1cm} (7)

Where $\alpha = \frac{5}{9}, \beta = \frac{3}{40}, \beta^* = \frac{9}{100}, \sigma = 0.5, \sigma^* = 0.5$ [23].

#### 2.3.2. Roughness function

Wall function or the logarithmic law of the wall shows that at a given point, the average velocity of a turbulent flow is proportional to the logarithmic measure of the distance between that point and the surface of the wall. The law of the wall is expressed through Equation 8.

$$u^+ = \frac{1}{k} \ln(y^+) + B$$  \hspace{1cm} (8)
Where the non-dimensional wall distance \( y^+ = y u_T / \nu \) (\( y \) represents the wall distance depicted in meters, \( u_T \) represents the shear velocity, and \( \nu \) represents fluid kinematic viscosity), von Karman constant \( \kappa = 0.41 \), and \( B \) depicts a constant. When modelling surface roughness, the wall function is modified due to the downward shift in the logarithmic region of the velocity profile induced by the roughness. Therefore, the roughness function is depicted by Equation 9.

\[
B = \left[ C + \frac{1}{\kappa} \ln \left( k_s^+ \right) \right] \left( 1 - \sin \left( \frac{\pi g}{2} \right) \right) + 8.5 \sin \left( \frac{\pi g}{2} \right)
\]  

(9)

Where \( C \) is 5.1, the roughness Reynolds number \( k_s^+ = \frac{k_s u_T}{\nu} \) (\( k_s \) depicts the equivalent surface roughness height in meters) and \( g \) represents the three roughness flow regime in Equation 10.

\[
g = \begin{cases} 
\ln \left( \frac{k_s^+}{k_{s,smooth}^+} \right) & k_{s,smooth}^+ < k_s^+ < k_{s,rough}^+ \\
\ln \left( \frac{k_{s,rough}^+}{k_{s,smooth}^+} \right) & k_s^+ > k_{s,rough}^+ \\
1 & k_s^+ < k_{s,smooth}^+ \\
0 & 
\end{cases}
\]  

(10)

The roughness geometry characteristics determines the values of \( k_{s,smooth}^+ \) and \( k_{s,rough}^+ \), which are equivalent to 2.25 and 90.0 respectively. These values were proposed by Ligraini and Moffat [23], where they are implemented in the numerical computation. In this study, the hull was tested with three variations of \( k_s \) values taken from a numerical study [23], namely 81.25 \( \mu m \), 325.00 \( \mu m \), and 568.75 \( \mu m \), representing realistically fouled hull [15]. An experimental study investigated the measurement techniques of hull roughness on a 160 m new class car ferry, where the actual average hull roughness was found to have a value of 81 \( \mu m \), far beneath the hull roughness of a new ship proposed by ITTC at 150 \( \mu m \) [25]. Therefore, the use of 81.25 \( \mu m \) as one of the roughness conditions gained more confidence in representing actual roughness on ships.

2.4 Computational domain and grid generation

The computational domain was generated as fully viscous, to accurately capture the frictional resistance of a ship without being interfered with wave-making resistance. No-slip walls was set for the boundaries at the bottom and the side of the ship, while symmetry was set for the top boundary and half plane of the boundary. While the ship itself was set as a wall function.

![Side View](image1)

![Front View](image2)

**Figure 2** Computational domain size for flat plate and DTMB 5415 CFD simulations.

The overall computational domain size was determined according to the ITTC [25], with \( L \) as the overall length of the ship (see **Figure 2**). The velocity inlet was measured 1.5L from the fore point of
the model, the pressure outlet was set 3L after the aft point of the model, the bottom was set 1.5L from the base of the model and the side wall was measured at 2L from the outer hull of the ship. To capture the effect of surface roughness on the logarithmic layer accurately, the grid around the ship was refined accordingly, by employing an inflation layer near the walls of the model by setting a suitable first layer thickness value determined by the $y^+$ value. This study utilised $y^+$ value which equals to 30 to approximately indicate the initial starting point of the logarithmic layer. As denser grid leads to higher computational cost and time, the grid refinement was only generated where the effects of logarithmic region is the most prominent for roughness condition, thus allowing the far field region of the flow to use normal sized grid as shown in Figure 3.

![Figure 3 Grid generation near the wall with the use of inflation layer as refinement method to better capture the logarithmic layer behaviour.](image)

### 2.4.1. Grid independence study

Ensuring the effectiveness of the grid before starting the simulations is needed to find the appropriate grid configuration. The principle behind the grid independence test is to find which optimum configuration offers the most accurate result while mainting the result’s credibility. This is shown when the results are able to maintain a similar value with only a slight deviation. The grid independence process was conducted at a smooth condition with results as shown in Table 2.

| Total Number of Cells ($\times 10^6$) | $F_D$ (kN) | $\Delta F_D$ (%) | $F_F$ (kN) | $\Delta F_F$ (%) |
|--------------------------------------|------------|------------------|------------|------------------|
| 0.086                                | 43.249     | 30.805           | 0.18       | 38.082           | 0.13567 | 0.793 |
| 0.353                                | 35.685     | 6.718            | 0.374      | 31.654           | 0.671 |
| 1.740                                | 35.184     | 0.010            | 31.691     | 0.001            |
Based on the results, the smallest deviation for both the total drag and frictional drag force, reaching well below 0.1%, was found at a total number of cells of 1.74 million. This result satisfies the requirement of grid independence study, where the deviation between two number of cells should not reach beyond 2%. Therefore, the grid configuration with the total number of cells reaching 1.74 million was used in further analysis of this study.

3. Results and Discussion

Understanding the effect of biofouling or surface roughness on the ship’s frictional resistance in this study is conducted through the results obtained from numerical simulation. These results are conveyed as non-dimensional coefficient to represent the resistance values, and later processed into graphics and images to depict the behaviour of skin friction. Comparison of the results from the two simulated models will give an idea of the flow behavior on a rough surface, especially on the frictional resistance.

3.1. Total and frictional drag

3.1.1. Total drag. The total resistance of the ship model and the flat plate has been obtained. The comparison between the two models is explored in this section. Table 3 shows the total resistance of the plate and the ship in all surface roughness condition, as well as the smooth hull condition.

| Roughness height (μm) | 𝐶_𝑻(10⁻³) | Δ𝐶_𝑻(%) |
|-----------------------|------------|---------|
|                       | DTMB | Flat Plate | DTMB | Flat Plate |
| 0                     | 3.347 | 3.322     |       |             |
| 81.25                 | 3.429 | 3.388     | 13.759| 14.523      |
| 325                   | 4.312 | 4.212     | 43.041| 42.349      |
| 568.75                | 4.935 | 4.807     | 63.728| 62.477      |

From Table 3, the results between each surface roughness condition for both flat plate and ship hull experienced a gradual increase due to increasing surface roughness. The increasing rate of both models show similarity with each other. For example, at the highest roughness height, the increase of total resistance coefficient for DTMB 5415 model reached 63.73% while the flat plate experienced 62.5% increase, which only differs at approximately 1%, and similar in all cases of roughness height.

3.1.2. Frictional drag. Since the frictional drag takes up at about 80 – 85% of the total resistance component for low speed vessels [19], a comparison between the frictional resistance and the total resistance values emphasized the importance of frictional resistance. According to the CFD simulation results in Table 4, the average frictional resistance contributes 90.39% for the DTMB model and 89.19% for the flat plate.

| Roughness height (μm) | 𝐶_𝑭(10⁻³) | Δ𝐶_𝑭(%) |
|-----------------------|------------|---------|
|                       | CFD | Granville | CFD | Granville |
|                       | DTMB | Flat Plate | DTMB | Flat Plate | DTMB | Flat Plate |
| 0                     | 3.014 | 2.959     | 2.972 | 2.972     |       |             |
| 81.25                 | 3.087 | 3.005     | 2.980 | 2.979     | 2.426 | 1.580     | 0.263 | 0.207     |
| 325                   | 3.905 | 3.762     | 3.840 | 3.819     | 29.564| 27.135    | 29.179| 28.498    |
| 568.75                | 4.485 | 4.312     | 4.691 | 4.659     | 48.786| 45.726    | 57.818| 56.743    |

Table 4 Comparison of friction resistance coefficient 𝐶_𝑭 and the increase of friction resistance coefficient 𝐶_𝑭 in percent due to the increase in roughness height between CFD simulation and Granville similarity law method for smooth and rough cases.
The results of this study can also be compared with a study where they developed an in-house code to predict the added ship resistance due to biofouling. The study investigated the effects of roughness on a number of ships at 4-26 m/s, including the DTMB 5415. Their results showed that the DTMB 5415 experienced a 4.2% increase of friction resistance at a speed of 10.5 m/s (Fn 0.28) due to light slime condition (30 – 100 μm) [27]. Whereas Table 4 shows that the increase of frictional resistance on DTMB 5415 due to 81.25 μm reached 2.4%, about 1.8% lower than the previous study.

Because frictional drag accounts for roughly 90% of total drag, the increase in frictional drag value due to increased surface roughness follows the same pattern as the increase in total drag. The frictional drag value obtained by CFD simulation of the two models under smooth surface conditions has a good agreement with the ITTC empirical equation [25], with a difference of less than 2%. Furthermore, utilizing the empirical approach of the Granville similarity [26] to estimate \( C_F \) on rough surfaces provides values that are consistent with the CFD simulation results. If we look carefully at Table 4, it can be seen that the average difference in frictional drag values from all roughness variations with CFD simulations and Granville similarity produces a difference in average value of 1.37% as illustrated in Figure 4.

### 3.2. Turbulent Boundary Layer

Another phenomenon that can be observed from the increase of surface roughness is the development of boundary layer that occurs on the wall along the object in the direction of the flow.
Figure 5 shows the differences of the boundary layer’s thickness formed on the ship's hull obtained from the CFD simulation results. The thickness of the boundary layer of the wall will increase following the increase of surface roughness, which was also observed by Demirel, et al. [7] on the hull of the KCS model. This thickening process occurs due to an increase in wall shear stress which causes a decrease in flow velocity in the log region [27].

The thickening of boundary layer is easier to analyze on a flat surface such as a plate, from which is shown in Figure 6. In general, boundary layers grow thicker in respect to the the flow of the fluid along the object, be it for smooth or rough surfaces. However, objects with the same length and X/L position will generate different boundary layer thickness if the surface is rough. Figure 6 shows the difference in boundary layer from several cross-sections of flat plate with smooth and rough surface conditions. The boundary layer thickness thickens gradually from the smooth surface all the way to the highest roughness height.

| X/L   | 0.872 | 0.698 | 0.523 | 0.349 | 0.174 |
|-------|-------|-------|-------|-------|-------|
| Ux (m/s)               |       |       |       |       |       |
| 0     |       |       |       |       |       |
| -0.25 |       |       |       |       |       |
| -0.5  |       |       |       |       |       |
| -0.75 |       |       |       |       |       |
| -1    |       |       |       |       |       |
| -1.25 |       |       |       |       |       |
| -1.5  |       |       |       |       |       |
| -1.75 |       |       |       |       |       |
| -2    |       |       |       |       |       |

Figure 6: Comparison of boundary layer development along the object for flat plate between smooth and rough surface

3.2.1. Mean velocity profile. The boundary layers can be used to generate mean velocity profile that occurs along the wall of the object, as well as investigate the effects of surface roughness from the derivation of the mean velocity plot in non-dimensional function of $u^+$ and $y^+$. Where $y^+$ represents
the distance between the wall and certain points off the wall, while \( u^+ \) expresses the velocity of the fluid at a \( y^+ \) location. The values of \( u^+ \) and \( y^+ \) are obtained by Equation 11.

\[
\begin{align*}
u^+ &= \frac{U}{\tau}, \quad y^+ = \frac{y \cdot U}{v} \\
\end{align*}
\]

Where \( \tau \) is friction velocity (m/s) which can be derived by the \( C_f \) obtained from CFD simulation \( \left( \tau = U_{\infty} \sqrt{C_f/2} \right) \), where \( U_{\infty} \) is freestream velocity, and \( U \) is dimensional velocity at \( y^+ \) (m/s). A velocity profile graph can be drawn using standard wall function for smooth surface and modified wall function for rough surface based on the shift in logarithmic region.

**Figure 7** (a) Roughness function proposed by Ligraini & Moffat, (b) Mean velocity profile of smooth surface and shift in velocity profile due to range of rough surface condition.

**Figure 7** shows the mean velocity profile which were formed by modifying the standard wall function by using the roughness function proposed by Ligraini & Moffat [23]. The downward shift in the log-law region represents the behaviour of velocity profile on rough surfaces. It can be analysed that on flat plates and ships, the downward shift share an approximately similar values at the same roughness condition. The farther down the downward shift from the smooth case, the slower the velocity of the fluid becomes, leading to the increase of shear stress occurrence on the wall which implicates the increase in frictional drag.

**4. Conclusion**
The CFD shows good accuracy in simulating the surface roughness effect on frictional resistance. Investigation on the effect of surface roughness were carried on two models, namely on flat plate and DTMB 5415. The study utilized RANS equation with \( k - \omega \) turbulence model to predict the drag. Simulating the flow at a roughness surface condition was achieved by implementing the roughness function. The frictional drag results show good agreement with the empirical approach for smooth surface and conform well with another empirical solution proposed by Granville [26]. Based on the results, the frictional drag increases gradually following the increase of the surface roughness. The surface roughness phenomenon also caused the thickening of the boundary layer as compared to the smooth condition of both models. The increase of frictional drag is also apparent in the logarithmic region of the flow, where it was represented by the downward shift on the velocity profile. Overall, the study found that the flat plate and DTMB 5415 models share the same characteristics in terms of frictional drag coefficient and the increase of frictional drag due to surface roughness. Hence, the
frictional drag on both smooth and rough surface conditions on ships can be predicted by using a flat plate with the same length and draft condition determined in preliminary design and in practical conditions.

It should be noted that this study had not covered every aspects of surface roughness and its effects. However, despite the negative effect of surface roughness due to biofouling on frictional resistance, certain types of roughness can instead have a positive impact. For instance, nature-inspired surfaces such as super-hydrophobic surfaces and riblets have been developed to be able to passively reduce the friction drag value [28]. For instance, super-hydrophobic surfaces are able to repel water and reduce 20-30% drag in laminar flow [29], while riblets are able to reduce surface shear stress and momentum losses, resulting in a 10% reduction in drag force [30], [31]. Therefore, the authors encourage further research on these types of surface roughness that remain as a challenging yet interesting topic to be resolved.

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