Drag of wall-bounded flows

Pim A. Bullee

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Pim Adriaan Bullee

Thesis committee members:

Prof. dr. J. L. Herek (chairman)	Universiteit Twente
Prof. dr. ir. R. G. H. Lammertink (promotor)	Universiteit Twente
Prof. dr. ret. nat. D. Lohse (promotor)	Universiteit Twente
Dr. S. G. Huisman (co-supervisor)	Universiteit Twente
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Prof. dr. C. Sun	Tsinghua University
Prof. dr. ir. C. G. P. H. Schroën	Wageningen University & Research, Universiteit Twente
Prof. dr. ir. C. H. Venner	Universiteit Twente

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Microscopy image of silica wafer in which double re-entrant cavities are etched (see chapter 1), with a natural diffraction pattern of the incident light at the surface. Image taken by Jan van Nieuwkasteele.

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Pim Bullee,

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University of Twente,

P.O. Box 217, 7500 AE Enschede, The Netherlands

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DRAG OF WALL-BOUNDED FLOWS

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by

Pim Adriaan Bullee
Born on the 2nd of March 1989
in Gorinchem, The Netherlands

This dissertation has been approved by the promotor:

Prof. dr. ir. R.G.H. Lammertink

Prof dr. ret. nat. D. Lohse

and the co-supervisor:

Asst. prof. dr. Sander G. Huisman

Voor Henri & Renske

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Preface

This thesis is about boundary conditions. In Dutch this would translate as “randvoorwaarden”, literally “edge conditions”. To me, as a Dutch person, has this translation always been a bit confusing. It appears very similar to “randzaken”, the non-essential issues that are best not to pay too much attention to. The boundary conditions are, however, very essential to fluid dynamics, and actually to any system that can be captured by equations¹. For the flow systems I have studied in this thesis, it is the conditions at the walls, bounding the flow, that determine the flow field further away from the wall. Is it smooth or rough, or maybe slippery and water repellent? In a wide range of flow conditions, I have studied what the influence of these different wall conditions is on the flow itself.

¹In all fairness, in its widest sense also to things that can not directly be described by equations, but that is better left untouched here.

“If you can’t solve a problem, then there is an easier problem you can solve: find it.”

C.H. (Kees) Venner

from: George Pólya *How to Solve It* (1945)

Introduction

Motivations for this work

“Work costs energy”. A very free expression of the first law of thermodynamics, but in the context of this introduction also a very practical one. Because, as energy comes at a cost, most easily (but not most importantly) expressed as money, a reduction in energy required to do the same amount of work is something everyone is interested in. The maritime industry is no exception to this, and also driven by the wish (and regulations) to reduce and change the composition of exhaust gas emissions, a research collaboration under the name of GasDrive was established. In this project the universities of Twente, Wageningen and Delft work together with maritime companies on the concept of a new ship. This ship is powered by an LNG engine together with a solid oxide fuel cell, and uses an underwater exhaust to release CO₂ bubbles under its hydrophobic coated hull as a drag reducing mechanism. This thesis has its focus on the drag reducing part of this project and studies the influence that surface properties such as roughness and hydrophobic chemistry have on flows.

Certain surface properties, for instance those related to the ‘slipperiness’ of the surface, can best be studied on a local sub-millimetre scale. The effects of surface roughness on the drag force however, are studied in flow conditions that are representative of real applications. Knowledge of a multitude of topics is therefore needed to appreciate the results of these studies. The remainder of this chapter will introduce the most important topics.

Slip versus no-slip

The no-slip boundary condition is probably known by any student that has at some point taken an introductory course in fluid dynamics. It is the assumption that the velocity of the fluid element in contact with the wall, is zero with respect to the velocity of the wall itself. It is probably the most

used constraint to find the particular solution, from the generic solution of the (simplified) differential equations that describe the flow (Navier–Stokes). As it is so commonly used, it is therefore tempting to take this condition for granted (as probably most students do), but this was not always the case. In the 19th century, this was definitely not the case, and its nature was highly debated by many scientists such as Bernoulli, Maxwell, and Prandtl, to name a few of the heavy-weights involved (Lauga et al., 2016).

In reality, the boundary condition at the wall is not no-slip, but allows for small fluid velocity. This is expressed as a slip length b , that is determined by the distance between the actual wall and the point, at which the linearly extrapolated velocity profile reaches zero velocity, as is shown schematically in figure 1. This defines the slip velocity as $u_b = b \frac{\partial u}{\partial y}$. For liquids on smooth surfaces, the typical slip length is in the nanometre range (Neto et al., 2005; Bocquet and Charlaix, 2010; Rothstein, 2010; Lee et al., 2014). Because this slip length is so small, the no-slip boundary condition works very well for most cases. Only when the typical length scale of the systems that are studied also becomes very small $\mathcal{O}(10 \mu\text{m})$, will the slip become relevant. The renewed interest in fluid slip and the slip boundary condition of the past fifty years, is therefore explained with the emergence of microfabrication methods and micro-fluidics in the 70s and 80s of the twentieth century.

Figure 1: Graphical representation of a channel flow with a no-slip boundary condition at the top wall, and a slip boundary condition at the bottom wall. Shown is the parabolic velocity profile $u(y)$ that is affected by the boundary conditions.

The amount of slip on the interface between a liquid and a solid is therefore negligible, and the no-slip boundary condition holds for most (practical) cases. For the gas-liquid interface however, this is very different. For

instance, if there would be a lot of friction on the gas-liquid interface, a gas bubble would rise slower in water under the influence of buoyancy. Because the dynamic viscosity of a gas is $\mathcal{O}(10^3)$ times lower than that of water, the gas-liquid interface is commonly regarded a shear-free, or a perfect slippery interface. So if one would succeed in creating a surface, that can act as a bounding wall to a liquid, and is composed of alternating solid-liquid and gas-liquid interfaces, surely some interesting (and useful) physics would be observed.

Superhydrophobicity

Superhydrophobicity occurs on surfaces that feature some sort of micropattern, and are capable of trapping air between the elements of this pattern and the liquid on top of the surface. The elements of this pattern can be pillars, holes, or have some complicated geometry. The typical length scale for such structures is $\mathcal{O}(\mu\text{m})$. To support the gas-liquid interface, the surface chemistry should be of the water repellant (hydrophobic) kind. This non-wetted, so-called Cassie–Baxter state, of a surface is meta-stable. Under influence of an external (pressure) trigger, or even spontaneously, can the surface transit to the wetted Wenzel state (Sbragaglia et al., 2007; Quéré, 2008). Both states of the liquid on the surface are illustrated in figure 2. Such a wetting event starts locally, with a failure of the gas-liquid interface between two adjacent pillars, but from there the wetted liquid will start flowing between pillars, which results in a complete failure of the surface. By using ‘holes’ instead of pillars, the individual gas-liquid interfaces are separated from one another, and local failure does not necessarily trigger global failure (Hensel et al., 2014; Domingues et al., 2017).

In the non-wetted state, the solid-liquid interface is smaller compared to a smooth surface. Instead is the surface composed of solid-liquid interfaces and gas-liquid interfaces: no-slip interfaces and no-shear interfaces. As a result, a liquid on top of such a surface experiences less friction. At solid-liquid interface it still has a zero velocity at the wall, whereas it can accelerate over the gas-liquid interface. The result is an (averaged) apparent slip length for the surface (Lauga et al., 2016). To increase the slip length, the area fraction of the gas-liquid interface $\phi_g = A_{gl}/A_0$, with A_0 the projected area, should be increased (Ybert et al., 2007; Ng and Wang, 2010; Lee et al., 2016). When the gas-liquid interfaces become larger however, the surface becomes more prone to (spontaneously) transit to the Wenzel state. This is explained with the Laplace pressure, the extra pressure a curved gas-liquid interface can exert on the fluid, because of its interaction with the surface. In its most

Figure 2: Illustration of a liquid on top of a surface of pillars in the non-wetted Cassie–Baxter state and the wetted Wenzel state. In the Cassie–Baxter state the solid-liquid contact area is smaller than the projected area, the area it would have on a smooth surface. Hence, there is less friction between the surface and the liquid. For the Wenzel state the opposite is true: the solid-liquid contact area is larger than the projected area, as a result will the friction also be larger compared a smooth interface. The typical superhydrophobic surface that is found in literature has a pillar diameter d and pitch p of $\mathcal{O}(\mu\text{m})$, a typical value for the pillar height h is $\mathcal{O}(10\text{--}100\ \mu\text{m})$.

simplified form it is expressed as

$$\Delta P = \frac{4\gamma}{D}, \quad (1)$$

with γ [N/m] the surface tension, that represents the intermolecular attraction of molecules inside the liquid that strive to minimize the surface area of the liquid. The diameter of the curved gas-liquid interface is denoted here by D , and hence it is clear that for larger diameters, the Laplace pressure goes down. As a result the gas-liquid interface is more prone to failure. For a gas-liquid interface with diameter $1\ \mu\text{m}$, equation 1, gives a Laplace pressure of 2.88 bar in water ($\gamma = 72\ \text{mN/m}$). For larger gas-liquid interfaces of $50\ \mu\text{m}$ this reduces to 0.06 bar. This explains why some surfaces with larger gas-liquid interfaces to study slip, only work with additional pressure support (Karatay et al., 2013). A solution by scaling down the gas-liquid interface diameter D together with the pillar diameter, to maintain a large ϕ_g , will not give desired results. The shear-free interfaces are in that case not long enough to provide the liquid with enough space to accelerate along the interface, to reach a velocity large enough to sufficiently contribute to an apparent slip length (Peaudecerf et al., 2017). Hence, it is a delicate balance between a surface that is very slippery (has long slip lengths), and a surface that can remain in a non-wetted state for a long time under various conditions.

Turbulent flows

One of the more elegant aspects of fluid dynamics is the use of dimensionless numbers. This can be the normalization of a length scale, but also a ratio between two physical quantities. Often these represent counteracting effects, and hence these dimensionless numbers are characteristic for the flow phenomena observed. This is also why engineers study scaled down models of aircraft in a wind tunnel: as long as the dimensionless numbers are the same for the model and its bigger brother, so will be the results for the scaled down model and the full-size aircraft.

Perhaps the most important dimensionless number is the Reynolds number. It gives the ratio between inertial and viscous forces as

$$\text{Re} = \frac{\rho U L}{\mu}. \quad (2)$$

Here ρ [kg/m³] is the fluid density, U [m/s] its velocity and L [m] the dominant length scale, e.g. diameter for pipe flows, and the length in stream-wise direction for external flows. The dynamic viscosity of the fluid is denoted by μ . For low Reynolds numbers ($\text{Re} \lesssim 3 \times 10^3$ for internal flows and $\text{Re} \lesssim 3 \times 10^5$ for external flows), the viscosity is strong enough to dampen out perturbations from the mean flow conditions, and is dominant over the fluids inertia. This we call laminar flow conditions. The perturbations are caused by (unwanted) small changes in the initial- and boundary conditions. For increasing Reynolds numbers, inertial effects grow stronger, and perturbations and instabilities to the mean flow will eventually be omnipresent. This we call a developed turbulent flow. Because of the dominating inertial forces, the flow is extremely sensitive to changes in initial and boundary conditions, and hence of an unpredictable and chaotic nature, in which a large range of time- and length scales are present (Romano et al., 2016). It turns out that it is not so easy to come up with a unambiguous definition of turbulence, and many scientists, engineers, and textbooks have come up with different descriptions, one maybe even more poetic than the other². The most striking quote I found, that catches the difficulty of finding a proper definition of turbulence, comes from the book of Tropea et al. (2016): “*What is turbulence? Turbulence is like pornography. It is hard to define, but if you see it, you recognize it immediately.*” (Romano et al., 2016).

Because the viscous forces are actually not that strong for liquids commonly found in nature and technology, the vast majority of flows that we

² “*Big whorls have little whorls, which feed on their velocity; And little whorls have lesser whorls, And so on to viscosity (in the molecular sense).*” — L.F. Richardson

Figure 3: Different regions in the boundary layer versus the distance from the wall, normalized with the thickness of the boundary layer y/δ and in viscous wall units $y^+ = y/\delta_v$, for a Reynolds number of $Re = 10^4$. Figure is adapted from (Pope, 2000).

encounter are turbulent. Examples include flows around ships and aircraft, but also in air ducts, and around wind turbines. Even inside the human body, turbulence is found in the blood recirculation system (Romano et al., 2016). Turbulence can occur in the absence of shear boundaries, as a *free shear flow*, for instance the exhaust gasses of a jet engine, the fumes of an erupting volcano and the air stream you create with a strong exhale. Examples of turbulence where boundaries are involved, so called *wall-bound turbulence*, include the flow over an aircraft wing, around a ship hull and through air ducts.

Very close to the wall in a turbulent wall-bounded flow, the no-slip boundary condition causes the velocity to be much lower as compared to the free stream conditions further away from the wall. Hence, whereas in the free-stream the flow conditions are dominated by inertia, very close to the wall viscous effects are dominant. Because this is a gradual transition, different characteristic regions can be defined near the wall as indicated in figure 3.

In the viscous sublayer directly at the wall, all stresses are the result of viscosity, and hence the (wall) shear stress between the liquid and the wall can be defined from the wall-normal mean velocity gradient as (Pope, 2000)

$$\tau_w = \mu \left(\frac{d\langle U \rangle}{dy} \right) \Big|_{y=0}. \quad (3)$$

The wall shear stress τ_w [N/m²] can be used as a generalized parameter to compare different wall-bound turbulent flows by defining a characteristic (wall friction) velocity and length scale as (Schlichting and Gersten, 2000)

$$u_\tau = \sqrt{\frac{\tau_w}{\rho}}, \quad (4)$$

$$\delta_v = \frac{\nu}{u_\tau}. \quad (5)$$

This results in the normalization of the velocity and wall-normal distance as

$$u^+ = \frac{u}{u_\tau}, \quad (6)$$

$$y^+ = \frac{y}{\delta_v}. \quad (7)$$

Almost all turbulent wall-bound flow have (in the limit of large Reynolds numbers) the same velocity distribution $u^+(y^+)$, which is therefore referred to as the *universal law of the wall* (Schlichting and Gersten, 2000). The velocity profile over a smooth wall in the log-law region is given by the Prandtl–von Kármán log-law

$$u^+ = \frac{1}{\kappa} \ln y^+ + B, \quad (8)$$

with $\kappa = 0.41$ the von Kármán constant and $B = 5.2$ the intercept for a smooth wall. Although the values shown in literature have some variations, typically they are within 5% of what is stated here (Pope, 2000).

Taylor–Couette turbulence

Classical systems that are used to study turbulence are pipe flow, channel flow, and Rayleigh–Bénard convection, with each system having its advantages and disadvantages. In this thesis the Taylor–Couette geometry is used to study turbulent flows, which consists of two concentric cylinders of equal height. A graphical representation is given in figure 4. The fluid in the annulus between the cylinders is set in motion by rotating either, or both cylinders. It is a closed system, meaning that the balance between the energy input and energy dissipation is exactly defined (Grossmann et al., 2016).

For Taylor–Couette we define two Reynolds numbers based on the width of the gap (annulus) between the cylinders $r_o - r_i$ as

$$Re_{i,o} = \frac{r_{i,o} \omega_{i,o} (r_o - r_i)}{\nu}. \quad (9)$$

Figure 4: Graphical representation of a Taylor–Couette setup, consisting of two co-axial cylinders of different radii. The gap between the cylinders is filled with a working liquid, typically water. The dimensions of the specific setup used in this thesis, the Twente Turbulent Taylor–Couette facility (T³C), are shown here as well (van Gils et al., 2011).

The radii of the inner and outer cylinder are denoted by $r_{i,o}$. The subscripts i and o denote the inner and outer cylinder, respectively, for all variables introduced. The rotational velocity is given by ω [rad/s], and $\nu = \mu/\rho$ [m²/s] is the kinematic viscosity. When both inner and outer cylinder are rotating, it might be more convenient to define a ‘combined’ shear-Reynolds number

$$Re_s = \frac{r_i(\omega_o - \omega_i)(r_o - r_i)}{\nu}. \quad (10)$$

To highlight the analogy between Taylor–Couette flow and Rayleigh–Bénard convection, the so-called *driving* of the system, the amount of energy that we put into it, is expressed by a Taylor number, rather than by a Reynolds number (Eckhardt et al., 2007a):

$$Ta = \frac{(1 + \eta)^4}{64\eta^2} \frac{(r_o - r_i)^2 (r_i + r_o)^2 (\omega_i - \omega_o)^2}{\nu^2}. \quad (11)$$

The Taylor number is reminiscent of the Reynolds number squared, but also includes the additional geometric information of the system $\eta = r_i/r_o$, the radius ratio of the cylinder.

In Taylor–Couette flow, the flux of angular momentum from the inner to the outer cylinder is conserved radially in the gap. For the pure laminar non-vortical flow case, the flux of angular momentum is solved analytically

as

$$J_{\text{lam}}^\omega = 2\nu r_i^2 r_o^2 (\omega_i - \omega_o) / (r_o^2 - r_i^2). \quad (12)$$

For turbulent flow, this flux is given by

$$J^\omega = r^3 (\langle u_r \omega \rangle_{A,t} - \nu \frac{\partial}{\partial r} \langle \omega \rangle_{A,t}). \quad (13)$$

Here u_r is the fluid velocity in radial direction, and $\omega = u_\theta/r$ is the angular velocity of the fluid. Quantities inside brackets $\langle \dots \rangle_{A,t}$ are averaged over the area A at constant distance from the centre (giving the surface of a cylinder in the gap) and time t . The ratio between the angular momentum flux for turbulent and laminar Taylor–Couette flow is the angular velocity Nusselt number (Grossmann et al., 2016)

$$Nu_\omega = \frac{J^\omega}{J_{\text{lam}}^\omega}. \quad (14)$$

The flux of this angular momentum is linearly related to the torque $\mathcal{T}$, which is needed to keep the cylinders spinning at constant angular velocity, and hence to the driving of the system:

$$J^\omega = \frac{\mathcal{T}}{2\pi L \rho}. \quad (15)$$

The torque $\mathcal{T}$ can be measured with relative ease and high accuracy, making the TC system very well suited to not only study the $Nu_\omega(Ta)$ scalings, but also skin friction drag. To this end a skin friction coefficient can be defined as

$$C_f = \frac{\mathcal{T}}{L \rho \nu^2 Re_s^2}. \quad (16)$$

With this coefficient, a fair comparison can be made of the influence that flow and bounding wall conditions have on the drag.

Rough walls in turbulent flows

Just as most flows that we encounter in nature and technology are turbulent, the majority of the surfaces that bound them (the walls) are rough. A rough surface has a larger contact area with the liquid, as is clear from figure 2, and hence there will be more friction. Depending on the geometrical characteristics of the roughness and the fluid velocity, a low-pressure wake region will form downstream (behind) the individual roughness elements. Hence, the drag force at the surface is a combination of viscous effects related

to the contact area and this pressure contribution. For larger flow velocities, the magnitude of viscous effects will stay the same, while the contribution of the pressure drag increases.

As there are many different types of roughness geometries, there is a desire to express the surface roughness k by an equivalent sand grain roughness $k_{s,eq}$ that can be related to the work of Nikuradse (1933), who studied flow through pipes that have a well defined roughness k_s created by sand grains of specific sizes. His velocity measurements revealed that the log-law relation of equation 8 remained intact, but shifted by a ΔU^+ over the full height of the profile.

$$u^+ = \frac{1}{\kappa} \ln y^+ + B - \Delta U^+ \quad (17)$$

The superscript $+$ denotes normalization with inner wall units, similar to equations 6 and 7. For a smooth wall obviously $\Delta U^+ = 0$, but for rough walls its value depends on the Reynolds number, until a certain maximum Reynolds number. Then it will reach a constant, $\Delta U^+ = 8.5$, and does not further change with increasing Reynolds numbers (Nikuradse, 1933; Pope, 2000). By comparing the velocity in the log-layer from a surface with roughness k , to the results of Nikuradse (1933) for a specific k_s and at a sufficiently large Reynolds number, we can determine the equivalent sand grain roughness $k_{s,eq}$. The equivalent sand grain roughness $k_{s,eq}$ is then used as a fitting parameter. This can be derived by rewriting equation 17, and plugging in $\Delta U^+ = 8.5$ (therefore we need a sufficiently large Reynolds number), which gives:

$$k_{s,eq} = \exp \left[\frac{1}{\kappa} \left(8.5 + \frac{1}{\kappa} \ln y - u^+(y) \right) \right], \quad (18)$$

with $u^+(y)$ the measured velocity profile in the log-law region (Schlichting and Gersten, 2000; Flack and Schultz, 2010).

The physical explanation for the upper bound of the shift $\Delta U^+ = 8.5$, is that the drag on the rough surface is then fully dominated by the pressure drag. This occurs at $k_{s,eq}^+ = 70$, and a further increase in roughness, or Reynolds number, will not further increase the pressure drag. The skin friction coefficient C_f then also becomes independent of the Reynolds number. This is referred to as the *fully rough* regime. A surface is *hydrodynamically smooth* when its roughness does not protrude through the viscous sublayer $k_{s,eq}^+ < 5$, and hence does not influence the flow. For any $k_{s,eq}^+$ in between these extremes, the surface is *transitionally rough*, and $\Delta U^+(k_s^+)$ (Schlichting and Gersten, 2000).

In Taylor–Couette flow the wall shear stress τ_w is directly coupled to the

torque $\mathcal{T}$ via

$$\tau_w = \frac{\mathcal{T}}{2\pi r_i^2 L}, \quad (19)$$

and can therefore be measured with relative easy and accuracy. In the Taylor–Couette facility used in this thesis (van Gils et al., 2011), typical values are $\tau_w = \mathcal{O}(100 \text{ N/m}^2)$. Following equations 6 and 7, this gives a viscous length scale $\delta_v = \mathcal{O}(5 \mu\text{m})$. Hence, any surface roughness greater than $k_{s,eq} \approx 0.4 \text{ mm}$ will be in the fully rough regime. For a ship the viscous length scale is larger, $\delta_v = \mathcal{O}(25 \mu\text{m})$, derived using a flat plate approximation for a 100 m vessel with a velocity of 10 m/s. Hence a hull with a sand grain roughness of $k_{s,eq} \approx 2 \text{ mm}$ would already be considered to be in the fully rough regime. The growth of biofouling such as algae and barnacles on ship hulls therefore has a huge impact on the fuel consumption of the ship. Not only due to the increased contact area between the hull and the water, but there is also a major contribution of pressure drag, as this roughness is in the fully rough regime. The total amount of energy spent in the maritime industry to overcome this additional drag from increased surface roughness is considerable, making this a very relevant topic of study.

Guide through this thesis

In chapter 1 of this thesis, we study the slip lengths of laminar flows over surfaces that feature ‘double re-entrant cavities’. These structures allow for the formation of gas-liquid interfaces with large diameters $\mathcal{O}(100 \mu\text{m})$. The influence that surfactants, adsorbed at the gas-liquid interface, have on the slip length is studied as well. These surfactants can cause a Marangoni stress, that cancels out the shear-free boundary condition of the interface. In chapter 2, results are presented that show how a Taylor–Couette cell can be used to characterise rough surfaces, and determine an equivalent sand grain roughness in turbulent flows. We advocate that this is a very promising method, because of the high precision of the relatively simple torque measurements. We also comment on the influence of the curved, rough walls of the Taylor–Couette geometry on these kind of measurements. In chapter 3, the effects of sandpaper roughness on air bubbly drag reduction are studied in Taylor–Couette turbulence. By changing the axially heterogeneous distribution of the roughness, the structure of the turbulent flow is controlled. We link these varying flow conditions to the distribution of air bubbles in the flow, and to the observed changes in drag reduction that were measured. Finally, in chapter 4 we study drag reduction in turbulent Taylor–Couette flow, and compare results for a smooth, hydrophilic inner cylinder to a rough, hydrophobic inner cylinder. We show that for a low

amount of air bubbles present in the flow, a drag increase related to the surface roughness of the hydrophobic inner cylinder is found, compared to the hydrophilic inner cylinder. For larger amounts of air bubbles, the rough hydrophobic inner cylinder gives more drag reduction compared to the smooth hydrophilic inner cylinder.

Chapter 1

Surfactant slip over double re-entrant cavities¹

We studied flows with a mean velocity of 7 mm/s in microchannels of height 200 μm , of which the bottom silica surface was made superhydrophobic by etching double re-entrant cavities. The surfaces had either 150 μm diameter cavities that cover 52 % of the 1 cm² bottom wall area, or 200 μm diameter cavities with a coverage of 60 %. Micro-particle image velocimetry (PIV) with a spatial resolution of 5 $\mu\text{m} \times 5 \mu\text{m}$ was used to measure the flow field near the bottom of the channel. The maximum slip lengths were found at the centres of the gas-liquid interfaces that separate the air in the cavities from the liquid in the channel. For the 150 μm cavities a maximum slip length of 14 μm was found with a mean value for the surface of 4 μm , which is 25 % of the analytically predicted mean slip length. For the 200 μm cavities the maximum slip length was 28 μm , with a mean value for the surface of 9 μm , which is 36 % of the analytical prediction. The influence of surfactants on the slip length was studied using four different concentrations of sodium dodecyl sulphate (SDS) that were below the critical micelle concentration. No significant difference in the slip length and velocity profiles was found between the four different concentrations and the measurements with pure water. We attribute this to the rapid adsorption and desorption of SDS at the interface. This prevents that a concentration gradient is formed, that would lead to a Marangoni stress and the resulting flow along the interface.

¹Double re-entrant surfaces were designed and produced by Eddy M. Domingues, Sankara Arunachalam, and Himanshu Mishra, of the King Abdullah University of Science and Technology. The microfluidic chips were designed and produced by Jan W. van Nieuwkastele, of the University of Twente. Experiments, analysis and writing are done by Bullee, supervision by Lammertink. Proofread by Huisman, Lohse, and Lammertink.

1.1 Introduction

Ever since Navier introduced his slip boundary condition (Navier, 1823), researchers have been looking for ways to study the amount of slip, as well as the parameters that are of influence on the amount of slip and applications of slippery surfaces. Various reviews have been written on these topics, see for instance Neto et al. (2005); Rothstein (2010); Maali and Hbushan (2012); Lee et al. (2014, 2016). In contrast to the no-slip boundary condition, the Navier slip boundary condition allows for a non-zero fluid velocity at the wall, relative to the velocity of the wall itself. The amount of slip is quantified by the slip length b , which is defined as the distance between the actual wall and a virtual wall, at which the linearly extrapolated velocity profile reaches zero velocity.

For flows over smooth surfaces, slip lengths reported do not exceed 10 nm (Bocquet and Charlaix, 2010). This slip is commonly referred to as true or intrinsic slip, which occurs at the interface between the solid wall and the fluid. When the surface is of a more complex, non-smooth geometry, a second fluid can be trapped between the main fluid and the wall, and slip occurs at the fluid-fluid interface (Neto et al., 2005). Here the ratio between the viscosities of the two liquids is an important parameter that determines the amount of slip. This slip the fluid-fluid interface is referred to as apparent slip. The most common examples are flows over superhydrophobic surfaces, where a gas layer is trapped between the liquid and the surface. The gas-liquid interface is generally regarded as a “shear-free” boundary, and, as a consequence, low drag and a large degree of slip are found (Tsai et al., 2009; Rothstein, 2010; Lee et al., 2014). The slip over such a superhydrophobic surface of alternating no-slip (solid-liquid) and no-shear (gas-liquid) interfaces is typically referred to as apparent or effective slip, and typical slip lengths are in the micrometre range, so two orders of magnitude larger compared to intrinsic slip, and scale with the size of the gas-liquid interfaces.

The two most import parameters that determine the amount of slip that can be found on superhydrophobic surfaces are the gas-liquid interface fraction of the total surface area ϕ_g , and the pitch of the surface microstructures L . Dependent of the geometry of the structures (e.g. grooves, posts, or holes) different analytical expressions can be derived for the slip length (Ybert et al., 2007; Lee et al., 2016). For the analytical derivations it is typically assumed that the gas-liquid interface is flat, and pinned at the edges of the microstructures. Deviations from these assumptions are found in practice, and result in variations in slip length. For instance, Karatay

et al. (2013) and Haase et al. (2016) showed that a slightly curved gas-liquid interface (protruding into the liquid) gives more slip compared to a flat interface. An interface that has a high curvature however will result in less slip (Hyväluoma and Harting, 2008; Karatay et al., 2013; Haase et al., 2016). The effect of the deformation of the gas-liquid interface on the slip length was studied numerically by Hyväluoma and Harting (2008). By varying the Capillary number Ca , it was revealed that for larger shear-rates (i.e. larger deformations of the interface) the slip length decreases (Hyväluoma and Harting, 2008). Contaminations at the gas-liquid interface, for instance from surfactants, that result under flow conditions (shear) in a surface tension gradient along the interface, can generate a Marangoni stress in opposite direction to the slip velocity (Schäffel et al., 2016; Peaudecerf et al., 2017; Song et al., 2018). When this Marangoni stress is strong enough, as was shown experimentally by Song et al. (2018), the direction of the flow near the gas-liquid interface can even be in opposite direction to the flow in the bulk of the fluid (Song et al., 2018). In practice, a smaller slip length will be found in experiments compared to the analytical expression, as the ideal conditions of a flat interface and a completely surfactant free liquid are unlikely to be realized (Peaudecerf et al., 2017).

In a two-dimensional numerical study on the influence of surfactant concentration on slip, Peaudecerf et al. (2017) model the adsorption and desorption kinetics of sodium dodecyl sulphate (SDS) and its transport between the interface and the bulk. The concentration gradient along the gas-liquid interface results in a Marangoni stress, which is coupled to the viscous stresses in the liquid and determines the boundary condition for the (slip) velocity at the gas-liquid interface. For long gas-liquid interfaces ($> 10^3 \mu\text{m}$), this effect becomes less strong, as the surfactant gradient over the length of the interface is smaller. With further increasing the interface length, the amount of *slippiness* at the surface that is lost from the addition of surfactant decreases towards that of a no-shear interface (Peaudecerf et al., 2017). In the limit of short ($g \leq 10 \mu\text{m}$) interfaces, the amount of slip found is very limited for both pure *clean* water and *contaminated* water with $10 \times 10^{-2} \text{ mmol/l}$ SDS. For long ($g \geq 10 \times 10^4 \mu\text{m}$) interfaces, the shear-free asymptote is reached for both clean and contaminated water.

The numerical results of Peaudecerf et al. (2017) with a gas-liquid interface of length $100 \mu\text{m}$, and a mean fluid velocity of $33 \mu\text{m/s}$, show that a *strong* surfactant, defined as a surfactant that has a high affinity for the gas-liquid interface, and a low diffusivity, can already result in a fully no-slip boundary condition at a surfactant concentration of 10^{-10} mmol/l . For a *weak* surfactant (low affinity and high diffusivity) under the same conditions,

the transition to fully no-slip occurs at concentrations above 10 mmol/l. In this case, the relatively fast diffusion counteracts the surfactant concentration gradient, and hence the resulting Marangoni stresses. For SDS, which is considered a *mild* surfactant, the transition to fully no-slip occurs under this conditions at a concentration of 10^{-1} mmol/l. About 50 % of the shear-free boundary condition is already lost at a SDS concentration of 10^{-3} mmol/l, where the slip condition of the surface is expressed as the ratio in fluid shear stress of (partly) shear-free versus a no-slip interface. Because low concentrations of surfactants in the liquid can already result in a concentration gradient (and hence Marangoni stresses) along the gas-liquid interface, the exact choice of adsorption kinetics is claimed to be of minor importance (Peaudecerf et al., 2017). For SDS it is known, however, that it very quickly adsorbs and desorbs from the gas-liquid interface, as it accounts for changes in the concentration at the interface relative to the concentration in the bulk (Casandra et al., 2017).

Experiments by Schäffel et al. (2016) with a mean flow velocity of 58 mm/s over a pillar superhydrophobic surface, showed 10 times less slip compared to their numerical predictions for clean water. This was attributed to surface active pollutants, that were unavoidably present during the clean-water experiments (Schäffel et al., 2016). The addition of the surfactant sodium 1-decanesulfonate (abbreviated as S-1DeS) further decreased the slip length, however, not to a no-slip condition. For the concentrations of S-1DeS reported, however, one might expect that based on the numerical results of Peaudecerf et al. (2017), a fully no-slip boundary condition would have been found. As the flow velocities are $\mathcal{O}(10^3)$ larger compared to Peaudecerf et al. (2017), and a different surfactant was used, a direct comparison is however not justified. Because of the larger flow velocity, the surfactant transport in the experiments by Schäffel et al. (2016) is dominated by advection, and the Péclet number $Pe = Lu/D$, with L the characteristic length, and D the diffusion coefficient (hence giving the ratio between advection and diffusion), $Pe = \mathcal{O}(100)$. In the work of Peaudecerf et al. (2017), diffusion will play a larger role in the transport of surfactants, as $Pe = \mathcal{O}(1)$, using $D \approx 5 \times 10^{-10} \text{ m}^2/\text{s}$ for SDS. In this work, we will match the Pe from (Schäffel et al., 2016), and the surfactant type from (Peaudecerf et al., 2017), to allow for a better understanding of the influence of the flow velocity on the slip lengths that are measured for different SDS concentrations.

The non-wetted, or Cassie–Baxter, state of a liquid on a superhydrophobic surface is unstable and can spontaneous transit to a wetted (Wenzel)

stated (Sbragaglia et al., 2007; Qur, 2008). To delay and prevent this transition, the geometrical and chemical properties of the surface must be optimised, as the superhydrophobic character of the surface is the result of a combination of both properties (Li et al., 2007; Lee et al., 2016). Thanks to modern microfabrication techniques, surfaces with microstructures of a high level of detail can be fabricated. This results in a broad range of surfaces of different microstructures that can be found in literature, such as pillars (Peters et al., 2009), pyramids (Qi et al., 2009), mushrooms (Liu and Kim, 2014), and fractal-like structures (Onda et al., 1996). The ideal microstructure for creating a superhydrophobic surface, features an overhang geometry (Cao et al., 2007). For such *re-entrant* structures, the area of the gas-liquid interface will have to increase when the liquid tries to wet the gap between the microstructures. The surface tension will act to minimize this area and hence counteracts the wetting. As a result, the surface can be made non-wetting even for materials that are hydrophilic (Cao et al., 2007; Tuteja et al., 2008; Liu and Kim, 2014; Hensel et al., 2014; Nosonovsky and Bhushan, 2016; Kaufman et al., 2017). A locally initialized wetting event will typically result in complete wetting of the surface by the liquid, as the locally wetted liquid will propagate sideways, and pushes out the gas from the yet un-wetted regions (Sbragaglia et al., 2007; Pirat et al., 2008; Peters et al., 2009; Hensel et al., 2014; Domingues et al., 2017). By isolating the individual air pockets (or cavities) from one-another, this effect can be prevented (Hensel et al., 2014; Domingues et al., 2017). By creating cavities that feature an overhang geometry, or a re-entrant (or even double re-entrant) micro-structure, researchers might possibly have found the holy grail of superhydrophobic surfaces: long-term stable hydrophobicity, even from a hydrophilic base material (Hensel et al., 2014; Kaufman et al., 2017; Domingues et al., 2017)

A superhydrophobic surface with large gas-liquid interfaces, that is also highly non-wettable, is a perfect surface to study slip. We used two different surfaces with circular double re-entrant cavities of diameters 150 μm and 200 μm , and used micro-particle image velocimetry to measure velocities close to the surface with a resolution of 5 $\mu\text{m} \times 5 \mu\text{m}$. This allowed for very local measurements of the fluid velocity while it travels over the gas-liquid interface, and hence also for the derivation of a local slip length at that specific location on the gas-liquid interface. This is used to study the effects that SDS surfactant molecules that are adsorbed at the gas-liquid interface have on the slip length, as a (local) gradient in surfactant concentration will translate to a Marangoni stress that has a direct influence on the (local) slip

length. By making use of flow conditions that result in large Péclet numbers $Pe \approx 400$, we show that surfactants have no influence on the slip length under our experimental conditions. As this is not in line with what is found in literature for smaller $Pe \approx 1$, it is argued that for flows with surfactants, the flow velocity is of influence on the slip length.

This chapter is organised as follows: In §1.2 we describe the experimental methods. The results are presented and discussed in §1.3, where we show velocity profiles above the gas-liquid interface, plot the local slip length and comment on the role of advection in the adsorption of surfactants at the gas-liquid interface. The chapter ends with a summery, conclusions, and an outlook.

1.2 Methods

1.2.1 Sample preparation

The slip surfaces are made of circular cavities of depth $40\text{ }\mu\text{m}$, etched in silica, with a double re-entrant structure integrated in the cavity walls. In figure 1.1, scanning electron microscopy images of the cavities are shown. For details regarding the production of these slip surfaces, we refer to the work of Domingues et al. (2017). Two different cavity diameters were studied: $150\text{ }\mu\text{m}$ and $200\text{ }\mu\text{m}$. One sample consisted of an area of $1\text{ cm} \times 1\text{ cm}$, in which equal-size circular cavities were organised in a triangular packing. On top of this, a glass slide was bonded that features a channel of depth $200\text{ }\mu\text{m}$ that was created by etching. Stream upwards and downwards of the cavity pattern, triangular inflow and outflow regions are located. Inlet and outlet holes were drilled in these regions. See figure 1.2 for a schematic of the microfluidic chip.

1.2.2 Analytical prediction of slip length

Following the derivation of Ybert et al. (2007), an analytical prediction can be used for the slip length of the surfaces with double re-entrant cavities. For a surface of circular holes the slip length scales logarithmically with the gas fraction as

$$\frac{b}{L} = -A \ln(1 - \phi_g) + B, \quad (1.1)$$

with L the (centre to centre) pitch length in flow direction (Ng and Wang, 2010). The pre-factors A and B were determined numerically by Ng and Wang (2010) as $A = 0.134$ and $B = -0.023$ for $0.25 < \phi_g < 0.78$, assuming a flat gas-liquid interface.

Figure 1.1: Scanning electron microscopy images of the surface showing the top (left) and a cross section of one of the channel walls, revealing the double re-entrant structure (right). Images courtesy of Sankara Arunachalam, Eddy M. Domingues and Himanshu Mishra from the King Abdullah University of Science and Technology.

Table 1.1: Surface parameters and analytically predicted slip lengths using equation (1.1).

cavity [μm]	L [μm]	gas fraction [ϕ_g]	slip [b/L]	slip [μm]
150	201	0.52	0.08	16
200	249	0.60	0.10	25

The two surfaces with cavity diameters $150\,\mu\text{m}$ and $200\,\mu\text{m}$ have a gas fraction ϕ_g expressed as the fraction of cavity area of the total area of 0.52 and 0.60, respectively, determined using image analysis of microscopy images of each surface. Plugging in these values in equation (1.1) predicts for the two surfaces slip lengths b/L of 0.08 and 0.10 respectively. Around a single cavity, a unit cell can be drawn, that acts as the repeating unit for this surface. The width of this unit cell is the pitch L and is $201\,\mu\text{m}$ and $249\,\mu\text{m}$, respectively.

1.2.3 Measurements

A pressure controller (Fluigent MFCZ-EZ) was used to pump ultra-pure water from a water system (Milli Q) at a constant pressure drop through the microfluidic channels. The inlet pressure was set at 100 mbar. In front of the microfluidic chip, a thin capillary of inner diameter $0.25\,\text{mm}$ was placed to increase the total pressure drop over the system. The inlet pressure could therefore be controlled and kept constant by the pressure controller with

Figure 1.2: Schematic of the microfluidic chip. The top view (left) shows the $1\text{ cm} \times 1\text{ cm}$ area in which the cavities were etched, and the triangular inflow and outflow regions. The cross section (right) shows the channel, formed by the silica bottom in which the cavities were etched, and the glass slide that is the top of the channel. Assumed is a flat gas-liquid (air-water) interface at height $z = 0$. Indicated is the domain $z \in [-20, +20]$ in which the focal plane of the microscope was translated with steps of $\Delta z = +2.5\text{ }\mu\text{m}$ during the PIV measurements. The blue $+$ and $\times$ symbols correspond to the markers in figure 1.5 and indicate the x -positions of the results that are shown there.

a higher accuracy. The typical flow velocity in the channel was of order $\mathcal{O}(10\text{ mm/s})$. With these velocities the capillary number is of the order $Ca = \mathcal{O}(10^{-2})$. Following the work of Hyvluoma and Harting (2008), we therefore assume that the shape of the gas-liquid interface is not deformed by the flow. Based on the channel height, a Reynolds number can be defined using the mean velocity as $Re = \bar{u}H/\nu = \mathcal{O}(1)$, with ν the kinematic viscosity of water.

Velocity measurements

The velocities in the channel were measured using micro particle image velocimetry (μPIV). An inverted microscope (Zeiss Axio Observer.Z1) was used in combination with a $20\times f/0.4$ objective (Zeiss LD Plan-Neofluar) that has a depth of focus of about $5.8\text{ }\mu\text{m}$. Connected to the microscope was a CCD camera with a resolution of 1376×1040 pixel (LaVision Imager Intense). A dual cavity laser (Litron LDY301 Nd:YLF 527 nm) was used to illuminate the $0.45\text{ }\mu\text{m}$ diameter fluorescent seeding particles (microparticles GmbH PS-FluoRot-0.5).

To determine the flow field, the flow was sampled at 17 different heights moving the focal plane from $z = -20\text{ }\mu\text{m}$ to $z = 20\text{ }\mu\text{m}$ in steps of $\Delta z =$

2.5 μm , with respect to the bottom wall at $z = 0$. The location of the bottom wall was determined visually, by varying the focal plane and making use of small imperfections and spots that were present in the silicon. Based on the depth of focus (5.8 μm), we estimate that we determined the bottom of the channel with an accuracy of $\pm 3 \mu\text{m}$. The velocity of tracer particles varies depending on their height within the focal plane. Assuming a linear velocity profile inside the focal plane and an equal distribution of particles, this should not be an issue. Only near the wall, where the very slow particles will be more difficult to resolve and the normalized velocity difference inside the focal plane is large, the results will be biased. To minimize this, 2000 image pairs were captured for each height, with the vertical position of the microscope focal plane controlled by the **Davis 8** PIV software, which was also used for the data analysis. A sum of correlation PIV algorithm was used, with a interrogation window size decreasing from 128 pixel $\times$ 128 pixel to 32 pixel $\times$ 32 pixel, with 50 % overlap and double passes for all window sizes.

Wall position, slip length, and shape of the interface

The silicon wafer material that is the bottom wall of the channel, acts as a mirror and reflects the light scattered by the PIV tracer particles. When the plane of focus is inside the bottom wall ($z < 0$), the reflections of the particles are also solved by the PIV algorithm. This results in a reflection of the velocity profile that can be used to accurately determine the position of the wall (Bolognesi et al., 2013). We used this method to confirm the position of the wall that we determined prior to the experiments.

The slip length was determined at all discrete positions inside the unit cell, by extrapolating the velocity profile ($u(z)$) to the point where the velocity reaches a value of zero, where a minimal slip length of 0 μm inside the unit cell was enforced. To circumvent the velocity bias near the wall, only velocities between $z = 10 \mu\text{m}$ and $z = 17.5 \mu\text{m}$ were taken into account for the extrapolation. The average slip length for the surface is determined by taking the mean of all slip lengths in the unit cell.

The shape of the gas-liquid interface was not determined directly from our experiments. Hence it was unclear whether it is flat, convex, or concave. From bright-field microscope images it is clear that a gas-liquid interface is present between the fluid and the cavities. Based on the observations of the diffraction pattern of the reflected light on the gas-liquid interface, that forms dark rings on the interface that change with the height of the focal plane, we expect however a slightly concave (deformed upwards, into the fluid) interface. This is in line with preliminary results from our mathemat-

Table 1.2: Surfactant concentrations c and corresponding surface tension σ for the different SDS concentrations (Hernández et al., 2006)

label	c [mmol/l]	σ [mN/m]
SDS 0	0.00	72.8
SDS 1	0.08	68.8
SDS 2	0.14	66.9
SDS 3	0.62	59.0
SDS 4	1.00	52.8

ical model that combines the evaporation and condensation of vapor from the liquid into the cavity, diffusion of air from the cavity into the liquid and the static pressure to predict the shape of the gas-liquid interface. As these are preliminary results, we assume in our analysis a completely flat interface. Domingues et al. (2018) found for the same double re-entrant cavities a concave gas-liquid interface with confocal scanning microscopy measurements, under static (no flow) conditions.

Surfactant solutions

Different surfactant concentration solutions were made using sodium 1-decanesulfonate (SDS) (Sigma-Aldrich, $\geq 99.0\%$). We used the work of Hernández et al. (2006) to choose four different SDS concentrations, see table 1.2. The critical micelle concentration (cmc) of SDS is 5 mmol/l (Hernández et al., 2006).

Figure 1.3: Plot of surface tension versus SDS concentration from the data of (Hernández et al., 2006). The red dots correspond to SDS solutions 1–4 from table 1.2.

1.3 Results

1.3.1 Velocity profiles

In figure 1.4 some typical results of the PIV measurements are shown for the 150 μm (left column) and 200 μm (right column) surfaces. The velocities were measured in the horizontal x, y -plane, with a resolution of $5 \mu\text{m} \times 5 \mu\text{m}$, at different heights with separation $\Delta z = 2.5 \mu\text{m}$ in the range $z \in [-20, 20] \mu\text{m}$, indicated in figure 1.2. The top row of figure 1.4 shows the bright field microscope images, with the same field of view as the PIV results shown in the middle row of the figure. The contours correspond to the length of the velocity vectors in the x, y -plane, interpolated in all directions between the discrete measurement positions. The velocity vectors are also plotted in the figures of the middle row. Upstream of the cavities in the low velocity regions, the vectors are slightly oriented towards the high velocity regions at the centre of the cavities. Downstream of the cavities, the opposite is true to account for the fluid continuity condition, in which the z -component of the velocity will also play a role. The bottom figure shows the contour plot of the cross section of the velocity field, for which the location is indicated by the dash-dotted line in the top figure. For all heights at which the velocity field was measured, the fluid velocities are larger for the 200 μm surface compared to the 150 μm surface. As the gas fraction ϕ_g is also larger compared to the 150 μm surface (0.60 versus 0.52), we attribute this to a combination of larger velocities along the bubble interface and a larger gas fraction.

At the solid-liquid interface, the flow velocities are close to zero. Hence we assume that here the no-slip boundary condition is indeed valid, and our velocity measurements close to the wall are biased at these locations. This bias is the result of a low fluid velocity near the wall, which goes to zero at the wall. As a result the displacement of tracer particles between two PIV images is also very small, and maybe even too small to result in a velocity vector. Since the depth of focus of the objective used is about $5.8 \mu\text{m}$, this creates a bias, as the larger displacements of the tracer particles (diameter $0.45 \mu\text{m}$) in fluid layers further away from the wall will also be detected. In figure 1.5, the velocity in the x, y -plane is plotted versus the height z above the surface, for two different positions in the x, y -plane: of maximum slip at the centre of the cavity (so the middle of the gas-liquid interface), and of minimum slip at the solid-liquid interface, at one of the corners of the unit cell. The locations of the measurements are indicated in the bright field microscope images of figure 1.4. On the borders of the unit cell we expect the no-slip boundary condition to hold. Because of a velocity bias in the measurements near the wall, however, it appears as a slip velocity boundary

Figure 1.4: Typical PIV results for the 150 μm (left column) and 200 μm (right column) surfaces. top: Bright-field microscopy image of the surface showing the field of view that of the PIV measurements. In red the unit cell is drawn that was used for determining the mean slip length. The dash-dotted red line indicates the cross section of the flow field in the bottom figure. The blue + and x symbols correspond to the markers in figure 1.5. middle: Vector field in the x, y -plane at 5 μm above the surface, with the background colour corresponding to the vector length. For clarity only every sixth vector in x -direction and every third vector in y -direction is plotted. bottom: Cross section of the velocity field showing the x, z -plane along the dash-dotted line indicated in the left figure.

Figure 1.5: The speed in the horizontal plane versus the height above the surface for the 150 μm (left) and 200 μm (right) surface. Plotted are the velocities at the centre of the cavity (+ symbols) and the velocities at one of the corners of the unit cell ($\times$ symbols), corresponding to the x, y -positions in figure 1.2 and figure 1.4. The shaded area represents the wall, that reflects the PIV tracer particles and mirrors the velocity profile in the channel. It is not a perfect reflection, which we attribute to distortion effects. Near the wall, the reference velocity profile levels off and is not linear anymore, which is not in line with what is expected, but we attribute this to the velocity bias. The velocities that are measured inside the cavities (+ symbols for $z < 0$), we attribute to non-ideal reflections of the particles that are caused by the shape of the interface.

condition. The PIV tracer particles are reflected by the wall, which results in a mirrored velocity profile that appears in the wall (Bolognesi et al., 2013), indicated by the dashed area in figure 1.5.

The velocities that are found in the cavities for $z < 0$ (so inside the gas-phase), are explained as the result of reflections by a curved gas-liquid interface. One could argue, that the gas-liquid interface is concave, and hence *sags* into the cavity, which would explain why a fluid velocity is measured at those positions. The velocities shown in figure 1.5 are measured at the centre of the gas-liquid interface. At this location a displacement of $-20\text{ }\mu\text{m}$ of the interface due to its concave shape could be possible. However, near the edge of the cavity, the velocity profile looks very similar, albeit with a smaller velocity gradient inside the cavity. A displacement of $-20\text{ }\mu\text{m}$ due to a concave shape of the interface close to the edge is impossible without wetting the cavity. We are however convinced air is trapped in the cavities, because we do see the typical diffraction patterns (dark rings) in our bright-field microscope images, see figure 1.4, that are associated with a (slightly) curved gas-liquid interface. Hence, the explanation of a distorted reflection (fish-eye effect) for the velocities that are measured inside the cavities is most viable.

As we only measured the velocities near the channel bottom wall, we estimate the average flow velocity by assuming a flow between parallel plates and fit a parabolic velocity profile. From this we calculate the mean velocity in the channel to be $\bar{u} = (6.9 \pm 0.5)\text{ mm/s}$. Based on this average velocity, for a channel height of $200\text{ }\mu\text{m}$ we find a Reynolds number of $Re = 14$.

1.3.2 Slip lengths

By extrapolating the velocity profiles in the z -direction, for every grid position in the x, y -plane until it reaches a zero velocity, we can construct a plot of the local slip length b as is shown in figure 1.6. To circumvent the velocity bias that results in a flattening of the velocity profile near the wall (see figure 1.5), we used for the extrapolation the velocities that were measured between $z = 10\text{ }\mu\text{m}$ and $z = 17.5\text{ }\mu\text{m}$, and enforce a no-slip boundary condition at the borders of the unit cell around the cavity. By taking the mean of all slip lengths inside the unit cell, we find a mean slip length of $4\text{ }\mu\text{m}$ for the $150\text{ }\mu\text{m}$ surface, which is 25 % of the analytically predicted slip length, see equation (1.1) and table 1.1. For the $200\text{ }\mu\text{m}$ surface the average slip length is $9\text{ }\mu\text{m}$, which is 36 % of the analytically predicted slip length.

Figure 1.6: Local slip length for the 150 μm (left) and 200 μm (right) surface. The average slip length for the 150 μm surface is 4 μm , which is 25 % of the analytically predicted slip length. For the 200 μm surface the average slip length is 9 μm , which is 36 % of the analytically predicted slip length.

Table 1.3: Surfactant concentrations c and the corresponding average slip length.

label	c [mmol/l]	b [μm]
SDS 0	0.00	9 ± 1
SDS 1	0.08	8 ± 1
SDS 2	0.14	11 ± 1
SDS 3	0.62	9 ± 1
SDS 4	1.00	7 ± 1

1.3.3 Influence of surfactant

In table 1.3 the average slip length of the 200 μm surface is shown for different SDS solutions, including the *SDS 0* reference solution of pure MilliQ water. Apart from a different surfactant concentration are the experimental conditions the same for all experiments. Striking is that for all SDS solutions a slip length of $(9 \pm 2) \mu\text{m}$ was found, 36 % of the analytically predicted slip length for pure water. This is a surprising results, as one would expect a much larger deviation from the analytical prediction for the slip length in the idealised case, in experiments where large concentrations of surfactant are present.

In figure 1.7 the absolute velocity is plotted versus the x -position, for a fixed y -position, and versus the y -position, for a fixed x -position. The fixed positions are at the centre of the gas-liquid interface. No variation of the

Figure 1.7: Local absolute velocity versus y -position (left), and versus x -position (right) over the centre of the gas-liquid interface. Shown are the results for the 4 different SDS concentrations and the reference zero concentration (SDS 0) at three different height above the surface: $z = 0 \mu\text{m}$, $7.5 \mu\text{m}$ ($z/H = 0.0375$) and $15 \mu\text{m}$ ($z/H = 0.075$). The dashed lines indicate the edges of the cavities. No significant trend or difference in velocity and velocity gradient can be observed, not between the different SDS concentrations, and also not compared to the SDS 0 reference concentration.

velocity and its gradient is observed between the different SDS solutions, including the zero concentration reference case, for the 3 different heights above the surface shown here. An explanation for this would be that liquid used for the zero concentration reference contains already enough surfactants and contaminations (from for instance the PIV tracer particle solution), such that a further increase in the surfactant concentration from the addition of SDS does not influence the results anymore. This would be in line with the findings of Peaudecerf et al. (2017), who write that the transition from a regime of no-shear at the gas-liquid interface, to a regime that is dominated by Marangoni stresses at the interface occurs at extremely small concentrations. The Marangoni stress acts as a force opposite in flow direction, and hence reduces the shear-free boundary condition. The lowest concentration of SDS we used in our experiments is $8 \times 10^{-2} \text{ mmol/l}$ (SDS 1), which is according to the numerical data of Peaudecerf et al. (2017), nearly at the no-slip asymptote. In our experiments, however, we do find significant slip. To explain this, we point out the role of advection in the transport of surfactants in our experiments.

Near the gas-liquid interface we find a velocity of $\mathcal{O}(1 \text{ mm/s})$. The residence time above the $200 \mu\text{m}$ cavities is therefore $\approx 0.2 \text{ s}$, which is still larger

than the typical equilibrium time for SDS adsorption which is $\mathcal{O}(10^{-3} \text{ s})$ (Hines, 1996). Because of the relatively large flow velocities, the transport of SDS is in our flow dominated by advection. This is expressed by the Péclet number Pe . For our experiments that use the $200 \mu\text{m}$ cavities $Pe \approx 400$. Since the advection of surfactant molecules in flow (x -) direction dominates the diffusion of surfactant molecules in the wall (y -) direction towards the gas-liquid interfaces, the distribution of SDS concentration in the bulk is therefore presumed uniform. The velocities and Pe in the experiments of Peaudecerf et al. (2017) are two orders of magnitude smaller compared to our experiments. Hence, for those flow velocities, diffusion plays a role, and perhaps the assumption of a uniform bulk concentration might not be valid anymore. The non-uniform bulk concentration allows for an equilibrium between the concentration in the bulk and at the interface at all positions, and also allows for a concentration difference in surfactants that are adsorbed at the interface, along the interface. A sudden variation in SDS adsorption at the interface is rapidly compensated by adsorption or desorption of molecules from, or into the bulk of the liquid (Casandra et al., 2017). The fast adsorption and desorption kinetics of SDS at the interface then prevents a concentration gradient along the gas-liquid interface. This can possibly explain for the differences between our experimental results, and what would be expected based on the work of Peaudecerf et al. (2017).

Another argument is related to the circular geometry of our cavities. Although the rectangular gas-liquid interfaces in the experiments by Peaudecerf et al. (2017) are much longer than ours (2 mm and 30 mm versus $200 \mu\text{m}$), they only have a width of $40 \mu\text{m}$. A long gas-liquid interface has the advantage over a shorter interface, that for the same concentration difference of adsorbed surfactants between the front and the rear of the interface, the concentration gradient (and hence the Marangoni stress) will be smaller. The width of the gas-liquid interface is also of influence on the slip velocity, as there is a velocity gradient of the stream wise velocity component in both the x -direction and the y -direction, as is demonstrated by the symmetrical velocity profiles in the middle plots of figure 1.4. Because our cavities are circular, the gas-liquid interfaces are much wider than those of Peaudecerf et al. (2017). This might be an indication that this second length scale, the lateral extent of the interface, is actually an important parameter. Whereas a rectangular, geometry of small width is a very logical choice to increase the stability of the gas-liquid interfaces, it does not lead to the largest slip lengths.

1.4 Summary, conclusions, and outlook

In this research we have used superhydrophobic surfaces consisting of double re-entrant cavities for their long term stability and long gas-liquid interfaces. Indeed these surfaces have proven to be very stable when submerged and subjected to flow, and have provided an ideal surface for detailed micro-particle image velocimetry measurements of local velocities and slip. From the local velocity measurements in the x, y -plane at different heights z , we were able to construct a velocity field that shows the evolution of the (slip) velocity over the gas-liquid interface. From this, local slip lengths were determined, of which the surface averaged values are within 25 % and 36 % of the analytically predicted slip length, for the two cavity diameters under consideration here. The difference between the analytical prediction and the experimental results might be explained by the presence of surface active pollutants, and is in line with findings in literature (Schäffel et al., 2016; Peaudecerf et al., 2017; Song et al., 2018).

The addition of (additional) surfactants to the fluid was not of influence on the slip lengths measured, which was tested using four different concentrations of sodium dodecyl sulphate (SDS). For the concentrations used, one would expect, that based on the results of Peaudecerf et al. (2017), the gradient in adsorbed surfactants at the gas-liquid interface would generate a Marangoni stress that is strong enough to give a no-slip boundary condition at the interface. This was not the case in our experiments, and also not in the experiments by Schäffel et al. (2016). Both, the experiments by Schäffel et al. (2016) and our experiments, use flow conditions in which advection dominates the transport of the surfactant molecules, as $Pe = \mathcal{O}(100)$. For Peaudecerf et al. (2017), $Pe = \mathcal{O}(1)$, and hence diffusion is also a significant transport mechanism. When advection dominates, the residence time is much shorter and therefore it can be assumed that the bulk concentration is constant. Depletion by diffusion is then not an issue. Hence, for surfactant solutions, the slip length might also depend on the flow velocity.

To better understand the role of advection on the slip length in flows with surfactants, future work should focus on the influence of flow velocity. For this, a large range of velocities should be evaluated, to properly ensure that both regimes of advection and diffusion dominated transport are studied. Future studies on the influence of the dimensions of the gas-liquid interface on the slip length, should also include the width of the interface. Based on the relatively large slip we find for our short gas-liquid interfaces, compared to Peaudecerf et al. (2017), we expect this to be of large influence. The surfaces with circular double re-entrant cavities are highly non-wettable for a long period of time without the need of, for instance, external pressure to

support the gas-liquid interface. We have shown that these surfaces are very well suited for fundamental research on fluid slip. However, such surfaces also have other possible applications in microfluids and high-tech industries. This research demonstrates their reliability using both clean and SDS-polluted water, the latter being more representative for the polluted liquids that are common for applications.

Chapter 2

Characterizing the drag properties of rough surfaces with a Taylor–Couette setup¹

Abstract

Wall-roughness induces extra drag in wall-bounded turbulent flows. Mapping any given roughness geometry to its fluid dynamic behaviour has been hampered by the lack of accurate, direct, and easy to perform measurements of skin-friction drag. Here the Taylor–Couette (TC) system provides an opportunity as it is a closed system and allows to directly and reliably measure the skin-friction. A conceptual complication, however, is the wall-curvature, which potentially complicates to connect the global wall friction with the local wall roughness characteristics. Here, we investigate the effects of a hydrodynamically fully rough surface on highly turbulent, inner cylinder rotating, TC flow. We carried out particle imaging velocimetry (PIV) measurements in the Twente Turbulent Taylor–Couette (T³C) facility with radius ratio $\eta = 0.720$ at Reynolds numbers in the range of $4.6 \times 10^5 < \text{Re}_i < 1.77 \times 10^6$, with water as working fluid. The inner cylinder is covered with P36 grit sandpaper, and the outer cylinder remains smooth and stationary. We find that the effects of a hydrodynamically fully rough surface on TC turbulence,

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where the roughness height k is three orders of magnitude smaller than the Obukhov curvature length L_c (which characterizes the effects of curvature on the turbulent flow, see Berghout *et al.* arXiv: 2003.03294, 2020), are similar to those effects of a fully rough surface on a flat plate turbulent boundary layer. Hence, the value of the equivalent sand grain height k_s , that characterizes the drag properties of a rough surface, is similar to those found for comparable sandpaper surfaces in a flat plate BL. Next, we analytically derive the dependence of the torque (skin-friction drag) on the Reynolds number for given wall roughness, characterized by k_s , and find agreement with the experimental results within 5%. Our findings demonstrate that global torque measurements in the TC facility are very well suited to reliably deduce local wall drag properties for any rough surface.

2.1 Introduction

2.1.1 Turbulent boundary layers over fully rough walls

The transport of a fluid over a solid body or the transport of a solid body through a fluid are always hindered by friction forces acting on the interface between the solid and the fluid. Ideally, the solid surface is smooth, and the drag force is a purely viscous force. In nature and engineering applications, however, solid surfaces are nearly always rough. In addition to a modified viscous force, the roughness also results in a pressure contribution to the drag force (‘pressure drag’), and consequently, an increase in the total drag force (the so-called ‘drag penalty’). For most roughness types, the contribution of the pressure drag to the total friction drag at the surface grows with increasing roughness height. Ultimately, when the pressure drag dominates, the surface is called ‘*hydrodynamically fully rough*’.

Due to the obvious interest in reducing the drag penalty, substantial research has been carried out to investigate the effects of rough surfaces on wall-bounded turbulent flows (Jiménez, 2004; Flack and Schultz, 2010; Chung et al., 2021). The key effect thereof is a downward shift (by Δu^+) of the mean streamwise velocity (u^+) in the overlap (or logarithmic) region of the turbulent BL (Hama, 1954). This shift can be seen as a direct measure of the drag penalty. The mean velocity profile for a rough wall in the overlap region is given by the Prandtl–von Kármán profile for smooth walls, minus this shift (Pope, 2000),

$$u^+ = \frac{1}{\kappa} \log y^+ + A - \Delta u^+, \quad (2.1)$$

where y^+ is the wall-normal distance and the von Kármán constant $\kappa \approx 0.40$

and $A \approx 5.0$ are extracted from experimental or numerical data. The superscript ‘+’ as usual indicates a normalization with the viscous velocity scale $u_\tau = \sqrt{\tau_w/\rho}$ and the viscous length scale $\delta_\nu = \nu/u_\tau$, where τ_w is the wall shear stress, ρ the fluid density and ν is the kinematic viscosity. For a fully rough surface, it can be derived from dimensional arguments that the velocity shift Δu^+ depends logarithmically on the roughness height k^+ , see e.g. (M. R. Raupach, 1991; Pope, 2000). The so-called fully rough asymptote of the roughness function is given by

$$\Delta u^+ = \frac{1}{\kappa} \log k_s^+ + A - B, \quad (2.2)$$

where $B \approx 8.5$ is the Nikuradse constant. The equivalent sand grain roughness height k_s^+ is obtained by fitting, such that the velocity shift of any fully rough surface collapses with the velocity shift of sand grains in turbulent pipe flow, that historically grew to be the reference case (Nikuradse, 1933). Hence, the key objective in research of wall bounded turbulent flows over rough surfaces is to relate the statistics of a rough surface to the value of k_s , which characterizes the roughness (Forooghi et al., 2017).

2.1.2 Taylor–Couette flow

TC flow — the flow between two coaxial, independently, rotating cylinders — is a canonical system in turbulence (Taylor, 1923; Grossmann et al., 2016). Since the domain is closed in all directions, global balances can easily be derived and monitored, giving room for extensive comparison between theory, experiments and simulations. Moreover, the torque (skin-friction) can be measured accurately and directly (van Gils et al., 2012; Huisman et al., 2014), in contrast to measurements of skin-friction in open systems.

The forcing strength of the system can be quantified by the ratio between the centrifugal force and the viscous force, i.e. the Taylor number

$$Ta = \frac{1}{4} \left(\frac{1 + \eta}{2\sqrt{\eta}} \right)^4 \frac{(r_o - r_i)^2 (r_i + r_o)^2 (\omega_i - \omega_o)^2}{\nu^2}. \quad (2.3)$$

Here η is the geometric measure of curvature, namely the ratio r_i/r_o of the radii of the cylinders. The subscripts i and o indicate inner cylinder and outer cylinder, respectively. The angular velocity is denoted by ω , and ν is the kinematic viscosity.

The key response of the system can be expressed as Nusselt number Nu_ω , which is the ratio between the angular velocity flux J^ω in radial direction

and its laminar counterpart J_{lam}^ω (Eckhardt et al., 2007b), as

$$Nu_\omega = \frac{J^\omega}{J_{\text{lam}}^\omega} = \frac{r^3(\langle u_r \omega \rangle_{A(r),t} - \nu \partial_r \langle \omega \rangle_{A(r),t})}{2\nu r_i^2 r_o^2 (\omega_i - \omega_o) / (r_o^2 - r_i^2)}. \quad (2.4)$$

Here, $\langle \cdot \rangle_{A(r),t}$ denotes averaging over the cylinder surface $A(r)$ and over time t . The Nusselt number Nu_ω is related to the torque $\mathcal{T}$ required to drive the inner cylinder. In non-dimensional form the torque can be expressed as

$$G = \frac{\mathcal{T}}{2\pi L \rho \nu^2} = Nu_\omega \frac{J_{\text{lam}}^\omega}{\nu^2}. \quad (2.5)$$

From here onwards, we assume inner cylinder rotation only, hence $\omega_o = 0$, as this corresponds to our experiments where we kept the (smooth) outer cylinder stationary at all times.

The torque is directly related to the wall shear stress $\tau_w = \mathcal{T} / (2\pi r_i^2 L)$. As commonly used in other canonical systems (e.g. the flat plate BL), we define the friction factor C_f as (Lathrop et al., 1992):

$$C_f = \frac{2\pi \tau_{w,i}}{\rho d^2 \omega_i^2} = 2\pi Nu_\omega J_{\text{lam}}^\omega (\nu Re_i)^{-2}. \quad (2.6)$$

This relation allows for straightforward comparison with other canonical wall-bounded flows like pipe flow, channel flow, flow over a plate, etc.

The turbulent flow in the TC setup is strongly influenced by the curvature of its bounding walls: the cylinders that drive the flow. This distinguishes turbulent TC flow from turbulent flows in other canonical systems. Bradshaw (1969) realized that the effects of curvature on a turbulent BL are very similar to the effects of buoyancy stratification on a turbulent BL (Obukhov, 1971). In analogy to the Obukhov length (Obukhov, 1971; Monin and Yaglom, 1975), he derived a length scale that separates the curved BL in a region where the effects of shear dominate (i.e. production of turbulence is dominated by shear production), and a region further away from the wall where curvature effects dominate (i.e. the production of turbulence is dominated by curvature). For TC turbulence, this ‘curvature Obukhov length’ is well approximated by (Berghout et al., 2020)

$$L_c = \frac{u_\tau}{\kappa \omega_i}. \quad (2.7)$$

Here we use equation (2.7) as definition of L_c . Using data of the mean velocity profiles from PIV in turbulent TC flow (Huisman et al., 2013b; van der Veen et al., 2016) and direct numerical simulations (DNS) (Ostilla-Mónico et al., 2015), the mean angular logarithmic velocity profile in the

region of the turbulent BL where curvature effects are important was recently derived for smooth wall TC flow (Berghout et al., 2020). By employing a matching argument between the velocity profiles of the turbulent BL and the bulk region, following the work of Cheng et al. (2020), an analytical expression for $Nu(Ta)$ was derived (Berghout et al., 2020).

The effects of irregular boundaries (extended transverse bars in the ‘obstacle regime’, as referred to by Jiménez (2004)) on turbulent TC flow was previously investigated by means of experiments (Cadot et al., 1997; van den Berg et al., 2003; Zhu et al., 2018; Verschoof et al., 2018a) and DNS (Zhu et al., 2017, 2018). Here the ratio k/d between the height of the bars (‘roughness’) and the gap width $d = r_o - r_i$ was as large as $k/d = 0.05$ or even 0.1. Later Berghout et al. (2019) numerically studied the effects of sand grain roughness ($k/d = 0.019$ – 0.087) on the turbulent TC velocity profiles, and found similar transitionally rough behaviour as the sand grain roughness of Nikuradse (1933) in turbulent pipe flow. However, we note that both the experimental and computational studies in TC flow suffered from limited scale separation between the roughness scale k and the gap width d .

In this chapter, we study the effects of a hydrodynamically fully rough inner cylinder on the turbulent wall-bounded flow, with small roughness $k/d = 0.014$, where $k \equiv 6k_\sigma$, and k_σ is the standard deviation of the roughness height. In particular, we keep k much smaller than the curvature Obukov length L_c , namely $k/L_c = 0.007$ – 0.010 . We will demonstrate that in order to study the effects of roughness on a turbulent flow in TC, $k \ll d$ is not enough. Rather, $k \ll L_c$ must also hold, to ensure that effects related to the streamwise curved geometry are not influencing the effects of the roughness.

Hence, we hypothesize that effects of roughness in TC turbulence (where $k \ll L_c$) are similar to the effects of roughness in canonical systems without streamwise curvature. Thus global measurements in the TC facility can be employed to characterize local drag properties of the rough surface. The outer cylinder remains smooth, to allow for optical access of the velocity profiles.

The chapter is organised as follows: In §2.2, we describe the experimental methods. We then (§2.3) discuss the relevant dynamical length scales in the experiment, and elaborate on the different regions in the BL where turbulent production is dominated by shear effects, and where effects related to the streamwise curvature of the setup play a role. We also comment on the scale separation and show that the roughness mainly affects the inertial shear dominated regime, and hence, effects from the streamwise curved geometry of the TC flow do not modify the velocity shift. In §2.4 we use the mean

Figure 2.1: (a) Cross section of the TC geometry. The flow is illuminated by a horizontal laser sheet from the right by a laser. The flow is imaged from the bottom through a mirror. The torque sensor only measured the torque of the middle part of the inner cylinder. (b) 3D visualization of the confocal scan of the used sandpaper. (c) Cross-section of the inner cylinder with the sandpaper attached to the surface.

velocity profiles of the inner cylinder boundary layer, to show that apart from the shift, the velocity profiles for rough and smooth inner cylinders are the same. We use this in §2.5 to calculate the angular velocity shift Δw^+ , from which the equivalent sand grain roughness height is determined in §2.6. In §2.7 we demonstrate that the bulk region of the flow is of constant angular momentum, which is used in §2.8 to analytically derive a relation between the torque (skin-friction drag) and the Reynolds number for given surface roughness k_s , in agreement with our experimental results. The chapter ends with a summary, conclusions, and an outlook (§2.9).

2.2 Experimental setup and methods

2.2.1 Experimental setup

The experiments were done in the Twente Turbulent Taylor–Couette (T³C) facility (van Gils et al., 2011), with water as working liquid. We used a fully rough inner cylinder with an outer radius of $r_i = 201.2$ mm, and a

transparent outer cylinder with an inner radius of $r_o = 279.4$ mm. This gives a radius ratio of $\eta = 0.720$ and a gap width $d = 78.2$ mm. The cylinders have a height of $L = 927$ mm and an aspect ratio of $\Gamma = L/d = 11.9$. For inner cylinder rotation only (the outer cylinder is stationary), the Reynolds number is defined with the velocity of the inner cylinder and the gap width d as

$$Re_i = \frac{\omega_i r_i d}{\nu}. \quad (2.8)$$

Using the viscous velocity u_τ obtained from measurements of the torque, the friction Reynolds number is defined as

$$Re_\tau = \frac{u_\tau (d/2)}{\nu}. \quad (2.9)$$

The roughness used was P36 grit sandpaper (VSM, ceramic industrial-grade), that was fixed to the entire surface of the inner cylinder using double-sided adhesive tape (tesa 51970). We define the characteristic length scale of the roughness as $k \equiv 6k_\sigma \approx 1.07$ mm (corresponding to the 99.8% interval of the height), where k_σ is the standard deviation of the local roughness height $h(x, y)$ (quantified using confocal microscopy (Bakhuis et al., 2020b)), and $k/d = 0.014$.

2.2.2 Experimental procedure

We performed seven experiments with different rotation rates of the inner cylinder, see table 2.1. During these experiments, the torque $\mathcal{T}$ that is required to drive the inner cylinder at fixed rotational velocity was constantly measured. The hollow reaction torque sensor that connects the drive shaft to the middle section of the inner cylinder is indicated in figure 2.1(a). By only measuring the torque on the middle section, possible end-plate effects are eliminated (van Gils et al., 2012). During the torque measurements, PIV was used to obtain the velocity field in the gap. To quantify the reproducibility of our torque measurements, we compared the torque data that were captured during the PIV experiments with three separate torque measurements thereafter. We find a spread in $\mathcal{T}$ smaller than 4% for all cases. These direct and reproducible measurements of the torque (friction) have an accuracy that is comparable to the measurement accuracy of wall shear stress in flat plate BLs, by means of a drag balance (Baars et al., 2016). The advantage of the TC setup is its relative compact size, and straightforward measurements of the torque.

For the PIV measurements, fluorescent polymer tracer particles (Dantec FPP-RhB-10 with diameters from 1 μm to 20 μm) were added to the working fluid. A horizontal laser sheet of ≈ 1 mm in thickness illuminated the

$Ta[\times 10^{12}]$	$Re_i[\times 10^6]$	Nu_w	$C_f[\times 10^{-2}]$	$Re_\tau[\times 10^3]$	$L_c^+[\times 10^3]$	k_σ^+	$\delta^+[\times 10^3]$
0.31	0.46	312	4.62	7.6	3.3	34	1.8
0.57	0.62	403	4.45	10.0	4.3	44	2.3
0.92	0.78	513	4.47	12.7	5.5	57	2.8
1.47	0.99	643	4.43	16.0	6.9	72	3.4
2.18	1.20	784	4.43	19.5	8.4	87	4.1
3.55	1.54	998	4.42	24.9	10.7	111	4.9
4.71	1.77	1137	4.37	28.5	12.2	127	5.4

Table 2.1: Control parameters, global response and relevant length scales, measured during the PIV measurements. Ta or Re_i characterize the driving of the system. Nu_w is the dimensionless angular velocity flux, C_f the friction factor, L_c^+ the curvature Obukhov length as defined in equation (2.7). Finally, k_σ^+ is the standard deviation of the sandpaper roughness, see Bakhuis et al. (2020b), and δ^+ is the BL thickness from equation (2.12).

working liquid at mid-height, through the transparent outer cylinder. The laser sheet was created using a **Quantel EverGreen** 200 mJ laser. The flow was imaged from below, through a window placed in the bottom plate of the apparatus. For this, a mirror was positioned under the bottom plate as drawn schematically in figure 2.1. The camera was a high-resolution sCMOS camera (**LaVision PCO.edge**), with a resolution of $2560 \text{ px} \times 2160 \text{ px}$ and a pixel size of $6.5 \mu\text{m}$. A 100 mm focal length objective with large dynamical range was used, giving an optical magnification of 0.17.

For each rotational velocity of the inner cylinder, 10^4 image pairs were acquired at a recording frequency equal to the rotation rate. The mean velocity distribution in the horizontal plane was computed using single-pixel ensemble correlation (Kähler et al., 2006, 2012). This gave a final resolution of $50 \mu\text{m}$, or about 1600 independent points in radial direction, evenly spread over the entire gap. From the correlation function (obtained for every pixel) one can directly extract the standard deviation of the velocity, by integrating the probability density function $\sigma(u) = \int_{-\infty}^{\infty} (u - \langle u \rangle)^2 \text{PDF}(u) du$, see Scharnowski et al. (2012). This ensures that all turbulent scales are included in the standard deviation, as opposed to regular PIV analysis. The velocity profiles were smoothed using a Gaussian filter with a standard deviation of $\sigma \approx 0.5 \text{ mm}$.

2.3 Curvature effects, the mean velocity profile and scale separation

2.3.1 The relative effects of curvature and shear

To characterize and quantify the relative effects of shear and curvature in TC turbulence, we study the ratio S of turbulence production by shear and curvature related effects (Bradshaw, 1969; Townsend, 1976; Berghout et al., 2020)

$$S = \frac{\overline{u'_\theta u'_r} \frac{d}{dr} U}{\frac{1}{r} \overline{u'_\theta u'_r} U} = \frac{1}{\omega} \frac{dU}{dr}, \quad (2.10)$$

where u'_θ and u'_r are the azimuthal and radial velocity fluctuations, respectively, and $\overline{u'_\theta u'_r}$ is the Reynolds stress. The mean azimuthal velocity is denoted by U , and $\omega = U/r$ is the angular velocity. The curvature Obukhov length L_c defined in equation (2.7) marks the transition from a region where the production of turbulence is dominated by shear ($y < L_c$), to a region where it is dominated by curvature effects ($y > L_c$). Hence, by this definition, for $S = 1$ we have $y^+ = L_c^+$. The definition from equation (2.7)

builds on the existence of a shear logarithmic region, where the gradient of the mean angular velocity is $\frac{d}{dr}U = u_\tau/(\kappa y)$. The angular velocity scale is approximated as $\omega = \omega_i$. Thus the curvature Obukhov length L_c can be defined with the inner cylinder rotation rate ω_i , and the wall-shear stress τ_w only, as done in equation 2.7.

Figure 2.2 is a schematics of a smooth and rough wall velocity profile in TC turbulence, with the relevant length scales and height offsets indicated. It indicates the different regions in the turbulent boundary layer, specifically related to this research are the regions where turbulence production is dominated by shear related effects, and where turbulence production is also influenced by effects that are related to the stream wise curvature of the TC setup.

Figure 2.3(a) presents the gradient of the mean angular velocity profile versus S , calculated from the PIV results. We find fair collapse of the velocity gradients of smooth (grey) and rough (coloured) wall profiles. When the effects of curvature are negligible $S \geq \mathcal{O}(10)$, the gradient of the velocity profile approaches $\kappa^{-1} \approx 2.5$. This occurs in a very small region close to the wall, where we cannot measure due to the presence of the sandpaper roughness. For the rough and smooth wall velocity profiles, we find that the gradient approaches λ^{-1} in the region where curvature affects the flow. For $S \leq 1$, curvature effects dominate the flow, and a constant angular momentum region (i.e. the bulk flow) sets in (Berghout et al., 2020).

2.3.2 The mean angular velocity profile

Figure 2.3(a) shows that the shear & curvature affected region of the BL contains a constant gradient ($= \lambda^{-1}$) of the mean angular velocity. From this observation, Berghout et al. (2020) derived the full equation of the mean angular velocity in the shear and curvature affected region in the BL (in short ‘curvature log’).

The offset of the logarithmic velocity profile (with slope λ^{-1}) in the shear & curvature affected region, as indicated in figure 2.2, is a function of the wall normal location where curvature related effect start to affects the flow. From PIV results, the exact location was found to be $y^+ = 0.20L_c^+$, where L_c^+ is calculated from equation (2.7). Therefore, the offset is $\kappa^{-1} \log 0.2L_c^+ + A$, where $A = 5.0$ is the offset of the logarithmic velocity profile in the shear affected region (Pope, 2000). The transition in the logarithmic velocity profile from the shear affected region to the shear & curvature effected region at $y^+ = 0.20L_c^+$ is not sharp but gradual. To account for this, we introduce a constant C_{bl} that connects the logarithmic velocity profiles of both regions. Berghout et al. (2020) found that $A + C_{bl} + \left(\frac{1}{\kappa} - \frac{1}{\lambda}\right) \log(0.2) = 1.0$, and the

Figure 2.2: Schematics of the various regions in turbulent TC flow. The boundary layer height $y^+ = \delta^+ = 0.65L_c^+$ is defined as the location where the logarithmic profile with slope λ^{-1} ends and the constant angular momentum region of the bulk velocity starts.

mean angular velocity equation as

$$\omega_s^+ = \frac{1}{\kappa} \log 0.2L_c^+ + A + C_{bl} + \frac{1}{\lambda} \log \frac{y^+}{0.2L_c^+} = \frac{1}{\lambda} \log y^+ + \left(\frac{1}{\kappa} - \frac{1}{\lambda} \right) \log L_c^+ + 1.0. \quad (2.11)$$

The transition from the shear & curvature affected region to the constant angular momentum region occurs at $y^+ = 0.65L_c^+$. This height we take as our definition of the BL height. Hence, $\delta^+ = \alpha L_c^+ = 0.65L_c^+$.

2.3.3 Scale separation

Key to the understanding of the effects of roughness in TC turbulence is the concept of scale separation. Therefore, we plot in figure 2.3(b) S versus the wall-normal distance $y^+ = (r - r_i - 2k_\sigma)/\delta_\nu$, where $2k_\sigma^+$ is the wall offset of the sandpaper (Jiménez, 2004). We note that the wall offset $2k_\sigma^+$ of the rough wall is an approximation, however, the results are not sensitive to the exact choice of the wall offset. As a reference, we plot the smooth wall profile (grey) at $Ta = 6.2 \times 10^{12}$ (Huisman et al., 2013b), together with the rough wall profiles (colour).

Table 2.1 also presents the relevant dynamical length scales in the ex-

Figure 2.3: (a) Compensated gradient of the mean angular velocity profile versus the ratio S between the turbulence production by shear and that by curvature, see equation (2.10). Dotted and dashed lines represent the slope of the logarithmic velocity profile of the shear and the curvature dominated regimes, κ^{-1} and λ^{-1} , respectively. (b) Ratio S versus the wall-normal distance shifted with the wall offset, $y^+ = (r - r_i - 2k_\sigma)/\delta_\nu$, where $2k_\sigma^+$ is the wall offset of the sandpaper (Jiménez, 2004). Coloured lines are calculated from the PIV data of the rough wall cases. The grey line ($k_\sigma^+ = 0$) is the smooth wall profile at $Ta = 6.2 \times 10^{12}$ ($Re_\tau = 23093$), obtained from Huisman et al. (2013b). The dotted vertical lines represent the estimate of the maximum roughness height $6k_\sigma^+$. The dashed vertical lines represent the end of the shear logarithmic regime $y^+ = 0.2L_c^+$.

periments: Re_τ , L_c^+ and k_σ^+ respectively. The friction Reynolds number $Re_\tau = \frac{u_\tau(d/2)}{\nu}$ is the ratio of the largest dynamical length scale in the TC setup to the viscous length scale δ_v . Re_τ is on the same order as the smooth wall experiments by Huisman et al. (2013b) in TC, but also comparable to the rough BL experiments by Squire et al. (2016).

First of all, we find that $k_\sigma^+ \gg 1$. Thus, the roughness scale is much larger than the viscous length scale δ_v , pressure drag dominates viscous drag, and indeed we occupy the fully rough regime. Second, we find that the roughness sublayer height $\approx 3k_\sigma^+$ (Jiménez, 2004) is smaller than the outer bound of the shear dominated logarithmic region, $\approx 0.2L_c^+$ (vertical dashed lines in figure 2.3(b) (Berghout et al., 2020)). For the lowest roughness, $3k_\sigma^+/0.2L_c^+ = 0.40$, and for the highest roughness, $3k_\sigma^+/0.2L_c^+ = 0.10$. This separation of length scales allows for a region where the shear, self-similar, logarithmic region can form, albeit marginally. We finally find that the outer bound of the curvature dominated logarithmic region L_c^+ is smaller than the outer length scale Re_τ , so that $L_c^+/Re_\tau \approx 0.5$, which allows for a region where the curvature, self-similar, logarithmic region can form.

In short, from table 1 and figure 2.3(b), we find that the sandpaper roughness does affect the inertial region where curvature effects are negligible. Hence, we expect that the velocity shift of that region is very similar to that of identical sandpaper in a flat plate turbulent BL. In other words, we would expect a FRA with slope κ^{-1} , and a similar value of k_s , as we would measure for identical sandpaper in a flat plate turbulent BL.

2.4 Mean velocity profiles of the inner cylinder boundary layer

In this chapter we will show the angular velocity profile $\omega^+(y^+)$ rather than the azimuthal velocity profile $u^+(y^+)$, as it is $\omega^+(y^+)$ which is expected, based on the Navier–Stokes equations, to have a logarithmic profile (Grossmann et al., 2014). This theoretical expectation is supported by our analysis of experimental data for smooth wall TC flow (Berghout et al., 2020). Figure 2.4(a) shows the angular velocity profiles over the rough wall $\omega^+ = \langle \omega_i - \omega(r) \rangle_t / \omega_{\tau,i}$, with $\omega_{\tau,i} = u_{\tau,i}/r_i$, versus the wall-normal coordinate y^+ . In this and the next section, we focus our analysis on the mean velocity profiles of the inner cylinder BL, before in §2.7 we will report on the bulk profiles. We refer to Berghout et al. (2020) for an analysis of the smooth velocity profiles of the outer cylinder BL.

From figure 2.4(a) one sees immediately that with increasing roughness the rough wall profiles are increasingly shifted downwards, as expected. More

Figure 2.4: (a) Mean angular velocity ω^+ versus the wall-normal distance y^+ . The solid lines are the measured rough wall profiles. The dashed lines represent the theoretical smooth wall reference profiles (colours are the same), calculated from 2.11 and at matching L_c^+ and Re_τ with the rough wall profiles. (b) The compensated gradient of the rough wall profiles in (a), where the wall-normal distance is normalized with the BL height δ^+ . The colours are the same in both figures. The grey line is the smooth wall profile at $Ta = 6.2 \times 10^{12}$, obtained from Huisman et al. (2013b). The dashed horizontal line represents the slope λ^{-1} of the logarithmic velocity profile in the region where turbulence production is dominated by curvature effects. The dotted horizontal line represents the slope κ^{-1} of the logarithmic velocity profile in the region where turbulence production is dominated by shear.

importantly, we find from the diagnostic function $y^+ \frac{d\omega^+}{dy^+}$ (a useful representation of the gradients (Pope, 2000)) in figure 2.4(b), that the slope λ^{-1} of the curvature dominated logarithmic region is the same for rough wall TC turbulence as for smooth wall TC turbulence (grey line). Unfortunately, we could not resolve the very thin region in space where a shear dominated logarithmic was found by Huisman et al. (2013b), as the roughness peaks obstruct the view for the PIV very close to the wall.

For the inner cylinder BL the boundary layer height can be written as

$$\delta^+ = \alpha L_c^+ = \frac{\alpha u_{\tau,i} r_i}{\kappa \nu (\omega_i^+ + \Delta\omega^+)}, \quad (2.12)$$

where $\alpha = 0.65$ (figure 2.2). For a rough wall $\Delta\omega^+$ is a function of the equivalent sand grain roughness height k_s^+ and the curvature length L_c , so that $\Delta\omega^+(k_s^+, \frac{k_s}{L_c})$. When $k_s \ll L_c$, the angular velocity shift only depends on k_s , and the shift becomes $\Delta\omega^+(k_s^+)$. Since the inner cylinder rotates, the plus sign in the denominator of equation (2.12) is connected with the increase of angular fluid velocity in the inner cylinder BL due to the roughness. When we normalize the wall-normal distance with δ^+ , we expect the transition from a curvature logarithmic velocity profile to the constant angular momentum bulk velocity profile to occur at $y^+/\delta^+ = 1$. In figure 2.4(b) we find a fair collapse of both smooth and rough wall profiles in wall-normal direction, when normalized with the BL thickness δ^+ .

2.5 The fully rough asymptote

From the observation that both smooth and rough wall velocity profiles possess the same slope λ^{-1} of the curvature logarithmic region (figure 2.4b) we proceed to calculate the angular velocity shift $\Delta\omega^+$. Due to the roughness the angular velocity profiles in the shear logarithmic region are shifted, as already discussed in §2.3.1 and illustrated in figure 2.2. This shift remains also in the curvature logarithmic region, where we will now quantify it. The offset of that region scales with $\frac{1}{\kappa} \log(0.20L_c^+) + A$ (figure 2.2). Hence, it is imperative to calculate the angular velocity shift from the smooth wall velocity profile at matching L_c^+ .

Figure 2.4(a) shows these smooth wall profiles (dashed), where the colours match the respective rough wall cases, and both L_c^+ and Re_τ are matched. The velocity shift $\Delta\omega^+(y^+)$ from the theoretical smooth wall profile, equation (2.11), is plotted in figure 2.5(a). The horizontal plateaus confirm the similarity of the slopes of the velocity profiles. We extract $\Delta\omega^+$ at $y^+ \approx 0.3L_c^+ \approx 0.5\delta^+$ and plot the shift versus the roughness height in figure

Figure 2.5: $\Delta\omega^+$ of the rough wall profiles with respect to the reference smooth wall profiles. (a) Velocity shift versus the wall-normal distance y^+/δ^+ . (b) The velocity shift $\Delta\omega^+$, crosses in (a), versus the equivalent sand grain height k_s^+ . Black symbols are the experimental values. The solid grey line is the fully rough asymptote of Nikuradse (1933), equation (2.13). The solid blue line is an illustration of the curvature fully rough asymptote, with slope λ^{-1} and arbitrary height.

2.5(b). When we fit a function of the form $\Delta\omega^+ = \frac{1}{a} \log k^+ + b$ through all seven data points, we obtain $a = 0.34 \pm 0.02$, to within 15% of the von Kármán constant $\kappa \approx 0.38$, the slope of the shear dominated logarithmic profile. This confirms our hypothesis, as discussed in section 2.3.1, that the fully rough asymptote for $\delta_\nu \ll k < L_c^+$ has slope κ^{-1} . For reference, this is much higher than λ^{-1} .

To obtain a measure of the equivalent sandgrain roughness height k_s , we fit the data points to the fully rough asymptote of Nikuradse (1933)

$$\Delta\omega^+(k_s^+) = \frac{1}{\kappa} \log k_s^+ + 5.0 - 8.5, \quad (2.13)$$

and obtain $k_s = 5.54k_\sigma = 0.97 \text{ mm}$. For clarity, we also plot a ‘curvature fully rough asymptote’ with slope λ^{-1} (blue line) in figure 2.5(b).

2.6 The equivalent sand grain roughness height

The hypothesis in this research, postulated in §2.3.1, is that the fully rough asymptote in TC turbulence with $\delta_\nu \ll k < L_c$ is the same (or very similar) to the fully rough asymptote in flows without streamwise curvature. We have already demonstrated in §2.5 that the slope κ^{-1} of the fully rough asymptote is indeed the same. This leaves us with a comparison of the value of k_s , between TC turbulence and canonical systems without streamwise curvature.

In literature, we have found two reports on turbulent flows over sandpaper roughness: the work of Squire et al. (2016), employing grit 36 sandpaper in a turbulent BL, and Flack et al. (2007) who employed grit (12, 24, and 80) sandpaper in turbulent BL flow. In the rough wall TC experiments reported here we used grit 36 sandpaper. However, it is essential to realize that sandpaper is not only defined by the grit size. Other statistics, like the skewness (an important parameter (Forooghi et al., 2017), which is 0.93 here, and only 0.09 in Squire et al. (2016)), do vary with manufacturing methods. We have tried to use the very same sandpaper type (SP40F, Awuko Abrasives) as Squire et al. (2016). Unfortunately, the sandpaper turned out to be not waterproof, and detached from the inner cylinder. We then applied new water resistant sandpaper (VSM, P36 grit ceramic industrial grade), with different surface roughness statistics.

To compare the drag property of the sandpaper surfaces in TC, to the respective sandpaper surfaces in literature, we plot the relationship between k_s and the root-mean-square and skewness in figure 2.6. The surface properties of the sandpaper surface from Flack et al. (2007) are taken from Flack and

Figure 2.6: Relationship between the equivalent sand grain roughness height divided by the root-mean-square height k_s/k_{rms} and the skewness parameter Sk of different sandpaper surfaces. The solid black line is the empirical correlation for $Sk > 0$ from Flack et al. (2020). Data from turbulent BL flow using grit (12, 24, and 80) sandpaper (Flack et al., 2007), of which the surface statistics are listed in Flack and Schultz (2010), turbulent BL using grit 36 (Squire et al., 2016) and turbulent TC flow using grit 36 (this research).

Schultz (2010). The solid black line is the empirical correlation from Flack et al. (2020). We find that the relation between k_s and the Skewness Sk and the root-mean-square height k_{rms} of sandpaper used in our rough wall TC experiments is consistent with the empirical trend given for the sandpaper used in rough wall turbulent BL flow analysis. Whether, whether the deviation originates from the difference between TC and canonical systems without curvature, or originates from the different surface statistics, remains to be resolved.

2.7 The constant angular momentum region in the bulk

Thus far, we have discussed the velocity profiles of the inner cylinder BL. By means of matching this profile to the bulk velocity profile at BL height, one can derive the relationship between the torque ($Nu_\omega(Ta)$) and the velocity of the inner cylinder (Cheng et al., 2020; Berghout et al., 2020). For smooth wall inner cylinder rotating turbulent TC flow, it is well known that the angular momentum in the bulk (M_b) is constant (Wendt, 1933; Townsend, 1976), and, in fact, very close to half the inner cylinder angular momentum ($M_i = \omega_i r_i^2$), $M_b = 0.5M_i$. For rough wall TC flow however, and especially for asymmetric roughness when the inner cylinder is of a different roughness height than the outer cylinder, the exact value of M_b is a priori unknown. What however was shown is that for very rough walls (obstacles) the bulk azimuthal velocity profile is shifted towards the rough cylinder, due to the stronger coupling to that side thanks to the roughness (Zhu et al., 2017, 2018; Berghout et al., 2019).

If the bulk region velocity conforms to a constant angular momentum, it should match the angular momentum at the edge of the BL $r = r_i + \delta_r$. The momentum ratio (M_b/M_i) is the angular momentum in the bulk over the angular momentum of the inner cylinder

$$\frac{M_b}{M_i} = \frac{\omega|_{y=\delta_r}(r_i + \delta_r)^2}{\omega_i r_i^2}, \quad (2.14)$$

where $\omega|_{y=\delta_r} = \omega_{r,i}(\omega_i^+ - \omega_r^+(y^+ = \delta_r^+))$, and we fill in the velocity profile of the rough inner cylinder BL, $\omega_r^+(y^+ = \delta_r^+) = \frac{1}{\lambda} \log \delta_r^+ + (\frac{1}{\kappa} - \frac{1}{\lambda}) \log L_{c,r}^+ + 1.0 - \Delta\omega^+$. Figure 2.7 compares this prediction from equation 2.14 (dashed line) with the experimentally obtained velocity profiles (solid lines). Indeed, we find an excellent agreement between the predicted and the measured profiles. This supports the assumption that also the rough-wall velocity profiles conform to a constant angular momentum in the bulk. Finally, we

Figure 2.7: Bulk velocity profiles. (a) The mean angular velocity normalized with the inner velocity ω/ω_i , versus the radius $(r - r_i)/d$ normalized with the gap width d . The profiles for different roughness heights k_σ^+ are compared. The bulk profile is strongly shifted towards the rough inner cylinder, as the roughness there enhances the coupling between the inner BL and the bulk, similarly as the ribs have done in Zhu et al. (2018) (b) The angular momentum M , normalized with the inner cylinder angular momentum M_i . Solid lines are the PIV results and dashed lines (M_b/M_i) are calculated from equation (2.14). The colours are the same in both figures. The grey line is the smooth wall profile at $Ta = 6.2 \times 10^{12}$, obtained from Huisman et al. (2013b).

point out that the ‘overshooting’ of the profiles in the bulk, i.e. the slight increase in M with increasing r , is likely an effect of the turbulent Taylor vortices, and is thus expected to depend on the height coordinate z (Huisman et al., 2014). It is due to the detaching plumes which are transported to the other side of the gap by the Taylor rolls. Similar overshooting is well known from temperature profiles in turbulent Rayleigh–Bénard flow. (Tilgner et al., 1993; Ahlers et al., 2009)

2.8 Calculation of $Nu_\omega(Ta)$ and $C_f(Re)$

Since the angular momentum in the bulk is to a good approximation constant, we can match the angular momentum of the inner cylinder BL at BL height with the angular momentum of the outer cylinder BL at BL height: $M(\delta_{i,r}) = M(\delta_{o,s})$. Subscripts (i,o) refer to inner cylinder and outer cylinder BL quantities, where subscripts (s,r) refer to smooth and rough wall quantities. The matching argument becomes

$$(r_i + \delta_{i,r})^2 \omega_{\tau,i} \omega_{IC}^+(\delta_{i,r}^+) = (r_o - \delta_{o,s})^2 \omega_{\tau,o} \omega_{OC}^+(\delta_{o,s}^+), \quad (2.15)$$

where we realize that $\omega_{\tau,o} = \eta^2 \omega_{\tau,i}$. We substitute the BL equations for respectively rough and smooth walls into equation (2.15) and obtain

$$\begin{aligned} (r_i + \delta_{i,r})^2 \omega_{\tau,i} \left(\omega_i^+ - \frac{1}{\lambda} \log(\delta_{i,r}^+) - \left(\frac{1}{\kappa} - \frac{1}{\lambda} \right) \log(L_{c,i,r}^+) - C_i + \Delta\omega^+ \right) = \\ (r_o - \delta_{o,s})^2 \omega_{\tau,o} \left(\frac{1}{\lambda} \log(\delta_{o,s}^+) + \left(\frac{1}{\kappa} - \frac{1}{\lambda} \right) \log(L_{c,o,s}^+) + C_o \right). \end{aligned} \quad (2.16)$$

The rough wall, inner cylinder, BL height $\delta_{i,r}^+$, and the velocity shift $\Delta\omega^+$ are functions of the sand grain size. This makes the matching equation significantly more involved, in comparison to the smooth wall case (Berghout et al., 2020).

Following Cheng et al. (2020), we now rewrite the equation in terms of $Re_{\tau,i}$ and Re_i . The inner cylinder angular velocity becomes

$$\omega_i^+ = \frac{Re_i}{2Re_{\tau,i}}. \quad (2.17)$$

The equivalent sand grain size is

$$k_s^+ = 2 \frac{k_s}{d} Re_{\tau,i} = 2\epsilon Re_{\tau,i}. \quad (2.18)$$

Figure 2.8: Global response of the rough and smooth wall TC turbulence. (a) Nu_w versus Ta . (b) Friction factor C_f versus Re_i . Grey diamonds are the smooth wall experiments of van Gils et al. (2011), where the solid grey line shows the theory of Berghout et al. (2020) for smooth wall TC turbulence. Black squares are the rough IC measurements in this research, where the solid black line is the prediction from equation 2.22.

The fully rough asymptote from equation (2.13) can now be rewritten as

$$\Delta\omega^+ = \frac{1}{\kappa} \log(2\epsilon Re_{\tau,i}) + A - B. \quad (2.19)$$

The inner cylinder, rough wall, BL height $\delta_{i,r}^+$ is rewritten as

$$\delta_{i,r}^+ = \frac{2\alpha\eta Re_{\tau,i}}{\kappa(1-\eta)\mathcal{Z}}; \quad \text{with} \quad \mathcal{Z} = \left(\frac{Re_i}{2Re_{\tau,i}} + \frac{1}{\kappa} \log(2\epsilon Re_{\tau,i}) + A - B \right). \quad (2.20)$$

The outer cylinder, smooth wall, BL height $\delta_{o,s}^+$ is rewritten as

$$\delta_{o,s}^+ = \frac{4\alpha\eta^2 Re_{\tau,i}^2}{\kappa(1-\eta)Re_i}, \quad (2.21)$$

where we remember from equation (2.7) and equation (2.12) above that $L_{c,i,r}^+ = \delta_{i,r}^+/\alpha$, $L_{c,o,s}^+ = \delta_{o,s}^+/\alpha$. We can now substitute equations (2.17)–(2.21) into (2.16), and obtain

$$\begin{aligned} & \left(1 + \frac{\alpha}{\kappa\mathcal{Z}}\right)^2 \left(\frac{Re_i}{2Re_{\tau,i}} - \frac{1}{\lambda} \log \left(\frac{2\alpha\eta Re_{\tau,i}}{\kappa(1-\eta)\mathcal{Z}} \right) \right. \\ & \quad \left. - \left(\frac{1}{\kappa} - \frac{1}{\lambda} \right) \log \left(\frac{2\eta Re_{\tau,i}}{\kappa(1-\eta)\mathcal{Z}} \right) + \frac{1}{\kappa} \log(2\epsilon Re_{\tau,i}) + A - B - C_i \right) = \\ & \left(1 - \frac{2\alpha\eta Re_{\tau,i}}{\kappa Re_i} \right)^2 \left(\frac{1}{\lambda} \log \left(\frac{4\alpha\eta^2 Re_{\tau,i}^2}{\kappa(1-\eta)Re_i} \right) + \left(\frac{1}{\kappa} - \frac{1}{\lambda} \right) \log \left(\frac{4\eta^2 Re_{\tau,i}^2}{\kappa(1-\eta)Re_i} \right) + C_o \right). \end{aligned} \quad (2.22)$$

This implicit equation can be solved numerically to obtain $Re_{\tau,i}(Re_i)$. Finally, by means of equations (2.3) – (2.8), we express the result $Re_{\tau,i}(Re_i)$ into $Nu_\omega(Ta)$ and $C_f(Re_i)$ respectively.

Figure 2.8 presents the final result, together with the experimental data from smooth walls (van Gils et al., 2011) and with the analytical prediction for smooth wall TC (Berghout et al., 2020) (grey). The black open squares represent the fully rough inner cylinder rotating TC experiments from this research. The black solid line is our calculation from equation (2.22). We emphasize that no fitting parameters are used. All parameters find their origin in the velocity profiles, and originate from the slopes of the logarithmic velocity profiles (κ^{-1} , λ^{-1}), the offset of the smooth velocity profile (A, C_i, C_o) or the BL thickness fit for smooth walls α . This reflects that all parameters are universal for all radius ratios, and cannot be ‘tuned’.

The agreement between equation (2.22) and the experimental data (the maximum error is only $\approx 5\%$) is convincing. It implies that from straightforward measurements of the torque, for given inner cylinder rotation speed,

we can calculate the value of k_s with a reasonable accuracy. This means that the TC facility can potentially be used for direct, fast, measurements of surface drag properties, as characterized by k_s .

2.9 Summary, conclusions, and outlook

We carried out experiments of inner cylinder rotating (and stationary outer cylinder) Taylor–Couette (TC) turbulence with a rough inner cylinder and a smooth outer cylinder. We measured the torque, and, by means of PIV, the mean angular velocity profiles. The rough surface consisted of P36 industrial grade sandpaper, where the maximum roughness height k over the gap width d was $k/d = 0.014$. The roughness height k was much larger than the viscous length scale δ_ν , such that $k/\delta_\nu = 204\text{--}762$. The velocity shift of the rough wall azimuthal velocity profiles was, compared to the reference smooth wall, in the log-law region $\Delta w^+ > 9$ over the whole range of $4.6 \times 10^5 < \text{Re}_i < 1.77 \times 10^6$. Hence the sandpaper was dynamically fully rough. Furthermore, the roughness height $k < 0.16L_c$, where L_c is the curvature Obukhov length scale (Berghout et al., 2020), which separates the region in the BL where production of turbulence is dominated by shear, and the region in the BL where production of turbulence is governed by effects of streamwise curvature.

Using the mean azimuthal velocity profiles, we found that the slope of the fully rough asymptote, characterized by $\kappa = 0.34 \pm 0.02$, was very similar to previous findings in flat plate BLs $\kappa \approx 0.38$. Also, the value of the equivalent sand grain roughness height k_s compared reasonably well with those found for sandpaper in flat plate BLs (Flack et al., 2007; Squire et al., 2016).

Finally, to derive the relationship between the dimensionless torque and dimensionless driving of the system $Nu_\omega(Ta)$, we employed a matching argument between the inner cylinder BL rough mean angular momentum profile at the inner cylinder BL height, and the smooth outer cylinder BL mean angular momentum profile, at the outer cylinder BL height, similarly as Cheng et al. (2020) and Berghout et al. (2020) used for smooth wall TC flow. To justify this, we first showed that for a rough wall inner cylinder, a region of constant angular momentum exists in the bulk. We find a very convincing overlap between the theoretical prediction of the torque (or wall shear stress), and the experimentally measured values of the torque, with a maximum error of $\approx 5\%$.

These findings indicate that the turbulent TC facility can be a valuable setup for characterizing the drag properties of any rough surface. Direct and straightforward measurements of the torque can now be translated to

a value of the equivalent sand grain roughness height k_s . It seems that the value of k_s found in TC is similar to the value of k_s found in flat plate BLs.

In outlook of future work, we propose that more studies in both turbulent flat plate BLs and turbulent TC flow, with identical rough surfaces, are carried out to further compare what drag properties of rough surfaces influence. Further unanswered questions include the effects of even more considerable roughness penetrating the curvature affected logarithmic regime of the BL, which is related to finding the slope of the fully rough asymptote in that region. This could also be achieved by employing a TC setup with a lower radius ratio η , thus increasing curvature effects.

Chapter 3

Effect of axially varying sandpaper roughness on bubbly drag reduction in Taylor–Couette turbulence¹

Abstract

We experimentally investigate the influence of alternating rough and smooth walls on bubbly drag reduction. To this end, we apply rough sandpaper bands of width s between 48.4 mm and 148.5 mm, and roughness height $k = 695\text{ }\mu\text{m}$, around the smooth inner cylinder of the Twente Turbulent Taylor–Couette facility. Between two sandpaper bands, the inner cylinder is left uncovered over similar width s , resulting in alternating rough and smooth bands, forming a constant pattern in axial direction. We measure drag reduction (DR) in water that originates from introducing air bubbles to the fluid at (shear) Reynolds numbers Re_s ranging from 0.5×10^6 to 1.8×10^6 . Results are compared to bubbly drag reduction measurements with a completely smooth inner cylinder and an inner cylinder that is completely covered with sandpaper of the same roughness k . The outer cylinder is left smooth for all variations. The results are also compared to bub-

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bly drag reduction measurements where a smooth outer cylinder is rotating in opposite direction to the smooth inner cylinder. This counterrotation induces secondary flow structures that are very similar to those observed when the inner cylinder is composed of alternating rough and smooth bands. For the measurements with roughness, the bubbly DR is found to initially increase more strongly with Re_s , before levelling off to reach a value that no longer depends on Re_s . This is attributed to a better axial distribution of the air bubbles, resulting from the increased turbulence intensity of the flow compared to flow over a completely smooth wall at the same Re_s . The air bubbles are seen to accumulate on top of the rough wall sections in the flow. Here, locally, the drag is largest and so the drag reducing effect of the bubbles is felt strongest. Therefore, a larger maximum value of bubbly drag reduction is found for the alternating rough and smooth walls compared to the completely rough wall.

3.1 Introduction

High Reynolds number flows are known to experience a significant increase in drag due to roughness (Jiménez, 2004; Marusic et al., 2010; Flack and Schultz, 2010, 2014). To reduce energy costs, the aim is to reduce this frictional resistance. Therefore bubbly drag reduction (DR) in wall bound turbulent flows using bubble injection has been a topic of study for many years (Ceccio, 2010; Murai, 2014; Verschoof et al., 2018a). Promising applications can be found in the maritime industry, where a reduction of the ship drag force will result in reduced fuel consumption. The total drag of a ship is composed of form drag (related to the design of the hull) and skin friction drag, of which the latter is dependent on the roughness of the hull and increases drastically with biofouling growth (Jiménez, 2004; Schultz, 2007; Flack and Schultz, 2010, 2014). While air bubble drag reduction is commonly studied in laboratory set ups that make use of smooth walls, we study air bubble drag reduction in turbulent flows over heterogeneous rough walls. We aim to gain a better understanding of the mechanisms involved, hoping our work provides guidelines for the industrial application of air bubble drag reduction.

3.1.1 Taylor–Couette

The flow geometry we use to study bubble drag reduction over rough walls is the Taylor–Couette (TC) geometry. In this geometry, the flow is between two concentric, independently rotating cylinders. The TC geometry is a mathematically well defined and closed system, with a well defined energy balance, making it one of the canonical systems to study the physics of fluids. See for instance the reviews by Fardin et al. (2014); Grossmann et al. (2016) for a broader introduction to, and an overview of different studies, on TC flow. The working fluid between the cylinders is set in motion by rotation of one either or both cylinders, which generates a shear flow. We define a characteristic shear Reynolds number Re_s using the different geometric parameters of the system, the properties of the fluid and the rotation rates of both cylinders as

$$Re_s = \frac{r_i(\omega_i - \omega_o)(r_o - r_i)}{\nu}. \quad (3.1)$$

Here $r_{i,o}$ and $\omega_{i,o}$ give the radii and rotation rates of the inner- (subscript i) and outer (subscript o) cylinder. The fluid kinematic viscosity is denoted ν .

In TC flow, angular momentum is transported from the inner- to the outer cylinder. The transport of this angular momentum is linearly related to the torque, which is needed to keep the cylinders spinning at constant

angular velocity, and hence to the energy input to the system. The torque can be measured with relative ease and accuracy, making the TC system very well suited to measure fluid drag. Thanks to the geometry, stability and structures can be discovered in its flow. Even when the flow is highly turbulent, some form of order can be discovered, when radially outward transported fluid forms organised structures with radially inward moving fluid in the form of so-called turbulent Taylor vortices (Andereck et al., 1986). For high Re_s this only occurs in the counter-rotating regime, when the outer cylinder is rotating in opposite direction to the inner cylinder (Ostilla-Mónico et al., 2014; Grossmann et al., 2016). These rolls enhance the angular momentum transport, while their strength varies with both Re_s and the rotation ratio between inner- and outer cylinder a defined as

$$a = -\frac{\omega_o}{\omega_i}. \quad (3.2)$$

When $a \approx 0.36$, the rolls are at their strongest and transport most momentum (Huisman et al., 2014), which translates to the largest value of the drag found for that specific shear Reynolds number. Also larger fluctuations in the velocity of the fluid in the gap are found with counter-rotation, compared to a flow with a stationary outer cylinder (Dong, 2008; Huisman et al., 2013a).

3.1.2 Bubbly drag reduction

An overview of different studies on air bubble injection DR is given in the review articles by Ceccio (2010) and Murai (2014). For bubbly DR to be effective, the injected air bubbles need to stay close to the wall (van den Berg et al., 2007). When the bubbles migrate away from the wall, the drag reduction effect will be lost (Watanabe et al., 1998; Lu et al., 2005; Y. et al., 2005; Sanders et al., 2006; Elbing et al., 2008; Lu and Tryggvason, 2008; Murai, 2014). To achieve high values of bubbly DR, bubbles also need to have a large Weber number (van Gils et al., 2013; Verschoof et al., 2016; Spandan et al., 2018). When the Weber number is small ($We \ll 1$), bubbles are more easily transported by the turbulent flow, moving away from the boundaries. The amount of bubbly DR scales linearly with the amount of injected air in an open system (Elbing et al., 2008). When the amount of bubbles near the surface is sufficiently large, an air layer is formed (Zverkhovskiy, 2014; Rotte et al., 2016).

In Taylor–Couette turbulence it was found that wall roughness rib elements induce strong secondary flows, that transported the bubbles away from the wall and decrease the DR (van den Berg et al., 2007; Verschoof

et al., 2018a). This might be an effect related to the flow geometry, since in channel flow, increased microbubble DR was found for turbulent flow over sandpaper rough walls compared to smooth walls (Deutsch et al., 2004). However, it could also be attributed to a larger baseline drag for the rough walls. The relation between DR and gas injection rate was very similar for all rough and smooth cases (Deutsch et al., 2004). In the limit of high gas injection rates, bubbly drag reduction transits to gas (air) layer drag reduction as described in Elbing et al. (2008). The excess gas no longer forms bubbles, but instead a thin sheet is formed, decoupling the wall from the working liquid. Typical values of drag reduction observed in this regime are $(90 \pm 10) \%$ (Elbing et al., 2008). Similar values of drag reduction were observed by Saranadhi et al. (2016), from vapour bubbles created at the inner cylinder (Saranadhi et al., 2016). When an air layer is formed, a further increase of air injection rate does not further decrease the drag (Elbing et al., 2008), a limit also reached by Deutsch et al. (2004) for all rough and smooth cases.

3.1.3 Spanwise-varying roughness

In practice, most surfaces are rough, or at least feature roughness to some degree. Especially when the Reynolds numbers are large and the flows become turbulent, even small (μm -scale) roughness elements are felt by the flow. Although these might seem too small to be of influence on the flow, compared to the smallest lengthscales found in such a turbulent flow, they are very relevant. As a result, turbulent flows over rough walls are extensively studied. For a complete overview of the influence of wall roughness, we refer to the reviews and work by Jiménez (2004) and Flack and Schultz (2010, 2014). The majority of the studies focus, however, on homogeneous roughness, with a typical roughness feature size k much smaller than the major length scale of the flow d , e.g. the (half) height for channel-flow systems. However, in practice not all surfaces have homogeneous roughness, but rather roughness of a distributed and heterogeneously rough kind. Examples on ship hulls in the maritime industry include clusters of biofouling (barnacles), connections (welding seams, rivets, screwheads), and damages for roughness of a larger length scale. For roughness variations of a smaller length scale, examples include corrosion, micro-fouling (bio-slime), and variations in coating condition (Yeginbayeva and Atlar, 2018). Therefore it is industrially very relevant to include surfaces of non-homogeneous roughness in studies.

An important parameter in quantifying heterogeneous spanwise-varying rough surfaces is the size of the alternating rough and smooth patches, L_r

and L_s . When the rough and smooth patches are of equal size, so when $L_r = L_s$, typically a single parameter s is used for the patch size. The flow over a rough section will experience a higher wall shear stress compared to the flow over a smooth section. At the sharp stress transition region on the edge between a rough and a smooth patch, streamwise roll motions are formed (Hinze, 1967; Barros and Christensen, 2014; Willingham et al., 2014; Chung et al., 2018). At the extremes of very small or very large patch spacings ($s/d \lesssim 0.39$ or $s/d \gtrsim 6.28$ in (Chung et al., 2018)), the induced secondary flows are either not strong and large enough, or are not able to interact such that they are of significant influence on the bulk flow (Chung et al., 2018). Between these extremes, the roll motions were seen to interact with each other, generating a wall-normal velocity that does influence the bulk flow, breaking with Townsend’s hypothesis (Townsend, 1976) of outer layer similarity, that states that the turbulent flow in the bulk region is determined by the wall shear stress only (Chung et al., 2018).

3.1.4 Bubble position in the flow

For achieving air bubbly DR, the position of air bubbles in the flow is important (Fokoua et al., 2015). Multiple forces act on air bubbles moving through a (turbulent) flow, which makes the analysis of their displacements and positions complicated. An excellent review of these forces and studies related to this topic is given in the work of Magnaudet and Eames (2000) and Lohse (2018). Generally, bubbles in turbulent flow are observed to cluster in regions of high vorticity and low pressure (Mazzitelli et al., 2003; Climent et al., 2007).

In Taylor–Couette turbulence, the bubble position depends on the interplay between buoyancy force, the centripetal forces of both the mean flow displacement (rotation of the inner cylinder) and the Taylor vortices as well as smaller turbulent structures (Djéridi et al., 1999; Chouippe et al., 2014; Fokoua et al., 2015; Lohse, 2018). The central control parameter in this is the bubble Froude number, defined as the ratio between centrifugal and gravitational forces acting on the bubble

$$Fr_b = \frac{r_i \omega_i}{\sqrt{g r_b}}, \quad (3.3)$$

with r_b the bubble radius and g the gravitational acceleration. For small Froude numbers $Fr_b < 1$, buoyancy effects are dominating. This typically occurs at low Reynolds numbers, when ω_i is small, and the strength of the Taylor rolls is only marginal, or when bubbles are large (Climent et al., 2007;

Lohse, 2018). Here the mechanism for drag reduction are the rising bubbles that destroy the Taylor rolls, reducing the transport of angular momentum from inner to outer cylinder (Spandan et al., 2018; Lohse, 2018).

The stronger Taylor rolls that form at larger Reynolds numbers however, can trap small enough bubbles (i.e., when Fr_b is large enough) near their cores, and also at outflow regions close to the inner cylinder (Climent et al., 2007; Fokoua et al., 2015). When the bubbles are trapped and simply are passively advected by the Taylor rolls, i.e., for large enough Fr_b , their influence on the global drag is minimal (Lohse, 2018; Spandan et al., 2018). With further increasing Reynolds numbers, the flow dominance of Taylor rolls decreases and we might expect the centripetal force from the mean flow displacement to push bubbles towards the inner cylinder. At the same time, smaller turbulent structures that are distributed throughout the flow will act on the smaller bubbles and disperse them (van Gils et al., 2013; Chouippe et al., 2014). In the high Reynolds number regime of ultimate turbulence where we operate, stable and unstable roll structures do however still persist in the form of turbulent Taylor rolls (Huisman et al., 2014).

To quantify the balance between the centripetal forces that push the bubbles towards the inner cylinder wall and the turbulent pressure fluctuations that distribute the bubbles away from the wall into the bulk, van Gils et al. (2013) defined a centripetal Froude number

$$Fr_{cent}(r) = \frac{\sigma(u_\theta)^2/r_b}{\langle u_\theta \rangle^2/r}. \quad (3.4)$$

In this equation, $\sigma(u_\theta)$ is the standard deviation of the azimuthal velocity u_θ , representing the turbulent pressure fluctuations.

In this study we build on the work of Bakhuis et al. (2020a), who have used spanwise-varying roughness to control the secondary flow configurations that show up as turbulent Taylor vortices in high Reynolds number Taylor–Couette flow. In the current work we will use the different secondary flow configurations to study the influence on bubbly drag reduction and the position of bubbles in the flow. With this we provide insight into the mechanisms involving bubbly drag reduction in high Reynolds number flows. This is relevant for flows over rough, smooth and heterogeneous rough surfaces, giving guidelines to industry for bubbly drag reduction opportunities for a variety of surfaces.

3.2 Methods

All experiments were performed in the Twente Turbulent Taylor–Couette facility (T³C) as introduced in van Gils et al. (2011) and shown schematically in figure 3.1. The set up consist of two concentric cylinders of height $L = 927$ mm and radii $r_o = 279.4$ mm and $r_i = 200$ mm, resulting in a gap of width $d = r_o - r_i = 79.4$ mm. This gives a radius ratio $\eta = r_i/r_o = 0.716$, and an aspect ration $\Gamma = L/d = 11.68$. The resulting gap has a volume of 111 L and is filled with water while leaving out a void fraction α for air to form bubbles when the working fluid is set in motion. When we study single-phase flow, no air bubbles are introduced to the working fluid and $\alpha = 0$.

Figure 3.1: Schematic overview of the measurement set up, showing the outer cylinder, three-section inner cylinder and the driving shaft. In a), a smooth inner cylinder is shown, together with the torque sensor that is placed inside the middle section of the inner cylinder and makes the connection to the driving shaft. Also depicted in a) is the void fraction α , defining the amount of free air in the system. Out of this air pocket, when the inner cylinder is rotating strongly enough, a two-phase bubbly flow is eventually formed as the result from turbulent mixing. Figure b) shows the inner cylinder coated with sandpaper bands, creating an axially alternating rough/smooth surface. Four different band widths s are used. Normalized by the gap width d these give values of $\tilde{s} = s/d = 0.61, 0.93, 1.23$, and 1.87 . Since the roughness coverage of the surface is kept constant at 56 %, we create patterns of 10, 6, 5 and 4 roughness bands, respectively.

We vary the rotational frequency of the inner cylinder between 5 Hz and

18 Hz, whilst keeping the outer cylinder stationary. For this $a = 0$ case, the shear Reynolds number then ranges from Re_s 0.5×10^6 to 1.8×10^6 . We also study the influence of the large roll structures that originate from outer cylinder counter-rotation ($a > 0$) on bubbly drag reduction. For this we use three different shear Reynolds numbers: $Re_s = 0.8 \times 10^6$, $Re_s = 1.2 \times 10^6$, and $Re_s = 1.6 \times 10^6$. The ratio of the rotation rates a is then varied between 0 and 1.

The inner cylinder is fabricated from stainless steel and machined in azimuthal direction such that the largest surface roughness is in axial direction and has a value of $k_{ic} = 1.6 \mu\text{m}$. Using normalization with the length scales of the inner wall boundary layer δ_ν , this can be expressed as a maximum roughness of $k_{ic}/\delta_\nu = k_{ic}^+ \approx 1.0$ in wall normal units, reached at the maximum shear Reynolds number. From this it is concluded that the surface of the inner cylinder can be considered to be hydrodynamically smooth in our measurement range (Schlichting and Gersten, 2000). The outer cylinder is fabricated from transparent polished PMMA, allowing for optical accessibility of the flow in the gap.

3.2.1 Torque measurements

The amount of energy required to drive the system at set rotational frequencies is determined by the torque measured between the drive shaft and the inner cylinder using a Honeywell 2404-1k hollow reaction torque sensor, and the rotation rate measured using a magnetic angular encoder. The torque is only measured on the middle part of the three-section inner cylinder ($L_{\text{mid}} = 536 \text{ mm}$), to account for end-plate influences that modify the flow near the top and bottom of the system. For our drag measurements, we continuously measure the torque while accelerating the inner cylinder from 5 Hz to 18 Hz over a period of 78 min. For the measurements with outer cylinder counter-rotation, we start at $a = 0$ and increase this to $a = 1$ at fixed shear Reynolds number. The acceleration of the cylinders is the same as for the measurements with a stationary outer cylinder. From the torque $\mathcal{T}$ we calculate the skin friction coefficient C_f defined as

$$C_f = \frac{\mathcal{T}}{L_{\text{mid}} \rho \nu^2 Re_s^2}, \quad (3.5)$$

where ρ and ν are the density and kinematic viscosity, respectively, of the liquid, and Re_s as defined in equation 3.1. The temperature of the working fluid is continuously measured using a PT100 temperature sensor placed inside the inner cylinder. The density and viscosity of the liquid are temperature corrected using these measurements. To limit the temperature changes of

the working liquid that are the result of viscous dissipation inside the liquid, cooling is applied through the top and bottom plate of the set up, controlling the temperature at $(21.0 \pm 0.5)^\circ\text{C}$.

3.2.2 Axially varying roughness

By making use of sandpaper belts that are attached to the inner cylinder, we changed the roughness (pattern) of the inner cylinder wall. This is done through the same method and using the same materials as in our previous work (Bakhuis et al., 2020a). Apart from a completely smooth inner cylinder (no sandpaper attached) and a completely rough inner cylinder (whole surface covered with sandpaper) we study different repeating patterns of alternating rough and smooth bands, as shown schematically in figure 3.1. By using bands of different widths s (48.4 mm, 73.8 mm, 97.7 mm, and 148.5 mm), four different patterns were formed of 10, 6, 5, and 4 roughness bands respectively. For each pattern the coverage of the surface with roughness was 56 %. This is the case for each section of the three-piece inner cylinder, and for the cylinder as a whole as well. We can normalize the roughness band width s by the gap width d , resulting in values of $\tilde{s} = s/d$ of 1.87, 1.23, 0.93, and 0.61. Together with a completely smooth and completely rough inner cylinder, this results in a total of 6 different variations of the inner cylinder roughness that we studied. The outer cylinder was kept smooth at all times.

To create the regions of roughness on the inner cylinder, commercially available P36 industrial grade sandpaper belts (VSM XK885Y ceramics plus) were applied using double-sided adhesive tape (Tesa 51970), that together form a 2.5 mm layer. A 20 mm by 20 mm sample of the same sandpaper had been characterised using confocal microscopy with a resolution of $2.5\text{ }\mu\text{m}$ (Bakhuis et al., 2020a), see figure 3.2. With most of the roughness height h'_r within $\pm 2\sigma$ of the mean, the roughness of the surface is defined as the peak-to-valley distance $k \equiv 4\sigma(h_r) \approx 695\text{ }\mu\text{m}$. In wall normal units this corresponds to a value of $k^+ \approx 434$ for the largest Re_s of 1.8×10^6 and $k^+ \approx 122$ for the smallest Re_s of 5.0×10^5 . Hence, we are in the fully rough regime, since over the whole range of Re_s $k^+ > 70$ (Schlichting and Gersten, 2000). The driving of the flow over the roughness is dominated by pressure forces, whereas on the smooth parts this is purely driven by viscous forces Zhu et al. (2017, 2018). On the hull of a ship a roughness of $k^+ = 122$ would translate to a roughness $k \approx 3\text{ mm}$, derived using a flat plate approximation for a 100 m vessel with a velocity of 10 m/s. The typical size of small barnacle biofouling that grows on underwater ship hulls is about 2.5 mm (Schultz, 2004; Demirel et al., 2017a,b). Our largest roughness $k^+ = 434$ would correspond to $k \approx 13\text{ mm}$, following the same flat plate

approximation, which is similar to the size of very large barnacles with a typical size of 10 mm (Schultz, 2004; Demirel et al., 2017a,b).

3.2.3 Counter rotating outer cylinder

To generate a flow pattern that is similar to the one encountered in the measurements with an axially varying rough inner cylinder, the outer cylinder was rotated in direction opposite to the smooth inner cylinder. Since now an additional parameter $a \in [0, 1]$ is added to the phase space, we choose to limit ourselves to three different shear Reynolds numbers: $Re_s = 0.8 \times 10^6$, $Re_s = 1.2 \times 10^6$, and $Re_s = 1.6 \times 10^6$. For a fixed Re_s , we quasistatically ramp up from $a = 0$ to $a = 1$ and measure the torque. The skin friction coefficient C_f is compared between a two-phase flow ($\alpha = 2\%$) and a single-phase flow ($\alpha = 0\%$).

3.2.4 Flow visualizations

For the flow visualizations a Nikon D800E camera was used with a Sigma 50 mm objective. All visualizations were done under the same flow conditions of $Re_s = 0.8 \times 10^6$ and $\alpha = 1\%$. Because of the transparent outer cylinder, the bubbles and the roughness patches can easily be observed by eye.

Figure 3.2: a) Confocal microscopy image of a sample of the sandpaper roughness. b) Height distribution (PDF) of the surface determined using the data from Bakhuis et al. (2020a). The roughness is determined as $k \approx 695 \mu\text{m}$ using the peak-to-through distance, with the peak and through at $+2\sigma$ and -2σ from the mean as described in (Bakhuis et al., 2020a).

Figure 3.3: Digitally enhanced photographs of the set up, taken at $Re_s = 0.8 \times 10^6$ and $\alpha = 1\%$. From visual inspection it is clear that most bubbles reside on top of the rough patches, where the turbulent intensity of the flow is higher compared to the smooth patches. The effect is therefore two-fold: 1) compared to a completely smooth inner cylinder, for the same Re_s there will be stronger turbulent mixing in the flow, resulting in a better axial distribution of the air bubbles. And 2), at the rough-wall regions, the transfer of energy to the flow (or drag) is higher compared to the smooth-wall regions. The drag-reducing effect of bubbles is therefore at these positions of largest influence on the total drag. Hence, the bubbles move to the locations in the flow where they are needed most.

3.3 Results

3.3.1 Flow visualizations

Shown in figure 3.3 are photographs of the experiment, taken at $Re_s = 0.8 \times 10^6$ with $\alpha = 1\%$ air in the working liquid. The bubbles show a preference to accumulate on top of the rough patches in the flow. This is best visible in the roughness configurations where the separation between the roughness bands is largest: $\tilde{s} = 1.87$ and $\tilde{s} = 1.23$. When the separation between roughness bands is smaller, in the $\tilde{s} = 0.93$ and $\tilde{s} = 0.61$ configuration, the bubbles can more easily travel between bands, leading to a more even axial bubble distribution.

3.3.2 Secondary flow structure

To gain insight in the local flow organization, we refer to the work by Bakhuis et al. (2020a). Based on their results from particle image velocimetry measurements, we draw in figure 3.4 the positions and directions of the roll structures that are induced by the rough patches on the inner cylinder. The radially in- and outward flow on top of the rough- and smooth bands together form a roll (Bakhuis et al., 2020a). The locations of the rolls are fixed by the boundaries between the rough and the smooth bands. For the configuration with the smallest bands, $\tilde{s} = 0.61$, the radially outward flows on top of two adjacent roughness bands are also seen to combine (Bakhuis et al., 2020a). This can also be seen in figure 3.4, and may lead to variations in size and position of the secondary flows. Although for smooth walls it has been reported in literature that roll structures such as those that are observed for all different $\tilde{s}$ transport the air bubbles away from the inner cylinder by trapping them in their core (Mazzitelli et al., 2003; Climent et al., 2007; Lohse, 2018), this is not what we observe here for the flow over rough surfaces. Instead, we find the bubbles in the outflow regions on top of the roughness, as that is where the turbulent intensity is highest. This was shown by Bakhuis et al. (2020a), who used Laser Doppler Anemometry to measure the velocities at midgap and different heights, to cover the flow above both smooth and rough bands. They found the standard deviation of the velocity to reach a peak at the centre of the rough bands, with a value of $\sigma(u_\theta)/u_i \approx 0.04$, where u_i is the surface velocity of the inner cylinder. On the smooth bands, this had a value of $\sigma(u_\theta)/u_i \approx 0.03$. When the separation between roughness bands is too small, $\tilde{s} \leq 0.61$, this effect is lost (Bakhuis et al., 2020a). Also the rolls do then not follow the topology of the roughness bands anymore, as is shown in figure 3.4.

We can use this data of Bakhuis et al. (2020a) to define a local Froude number Fr_{cent} , quantify the preferred location of bubble accumulation. When we take u_θ in equation 3.4 to be $\approx u_i/2$ (Huisman et al., 2013b), we can plug in the values reported by Bakhuis et al. (2020a). For the bubble radius we refer to the work of van Gils et al. (2013), who measured the bubble diameter to be about 1 mm in the same set up for similar shear Reynolds numbers. With the assumption that the bubble diameter is the same on the roughness as it is on the smooth surfaces, we find $Fr_{cent} = 0.9$ above the smooth bands, and $Fr_{cent} = 1.6$ above the rough bands. Hence, with a 75 % larger Froude number above the roughness, we explain that we find more bubble trapping at these locations.

Figure 3.4: Position, size and rotational direction of the secondary flow structures, based on the inflow and outflow velocities on top of the smooth- and rough bands as reported by Bakhuis et al. (2020a) that were obtained using particle image velocimetry. The rolls are drawn for different roughness configurations $\tilde{s}$, in the gap between inner- (left) and outer cylinder (right). The height is normalized with the gap width $\tilde{z} = z/(r_o - r_i)$. The sandpaper roughness bands are indicated with red on the inner cylinder, the smooth bands are grey. For $\tilde{s} = 0.61$ outflow regions from two adjacent roughness bands were seen to combine (Bakhuis et al., 2020a), hence, larger roll structures can also form and are indicated by the red-dashed arrows.

3.3.3 Torque measurements

We first show the influence of the roughness in its different configurations on the skin friction coefficient C_f versus Re_s for single phase flow in figure 3.5. A simple linear interpolation between the measured torque for a fully smooth and a fully rough inner cylinder to arrive at the same 56 % roughness coverage as for the partly rough surfaces with different $\tilde{s}$ underestimates the skin friction coefficient of those patchy rough surfaces. This is the result of secondary flow structures (rolls) that are created by the alternating rough and smooth bands. On the bottom of figure 3.5, we plot the difference in skin friction coefficient between the rough surfaces and the smooth surfaces. For the fully rough surface, the skin friction coefficient is nearly doubled as compared to the smooth surface at the highest Re_s . We find the largest increase in skin friction coefficient for the roughness configuration $\tilde{s} = 0.93$. Based on our previous work (Bakhuis et al., 2020a) and other studies (Chung et al., 2018), this is explained as the configuration inducing the strongest rolls, being closest to $\tilde{s} = 1$. These rolls transport angular momentum from the inner cylinder (that drives the flow) to the outer cylinder. Hence, stronger rolls, can transport more angular momentum, which results in more torque and a larger skin friction coefficient. The same reasoning is used to explain that the configuration $\tilde{s} = 1.87$ will generate the least strong roll structures, and therefore gives the smallest increase in skin friction coefficient compared to the entirely smooth surface.

In figure 3.6 we plot the skin friction coefficient versus Re_s for the same roughness configurations, but now for a bubbly two-phase flow. The working liquid contains 2 volume percent of air bubbles $\alpha = 2\%$. The bottom figure shows the difference in skin friction coefficient ΔC_f between a flow containing 2 % air bubbles and a flow containing no air bubbles over the same roughness configuration. As a reference this is also shown for a fully smooth surface. Initially, up to $Re_s \approx 1.0 \times 10^6$, we find a very strong decrease in C_f with Re_s for the rough surfaces compared to the smooth surface when bubbles are introduced. We partly attribute this to increased levels of turbulence in the flow, owing to the introduction of the roughness that leads to a better axial distribution of the air bubbles at the lower shear Reynolds numbers. The other effect is that the bubbles prefer the regions of high turbulence intensity close to the roughness. This is also visible for the larger $Re_s > 1.0 \times 10^6$, where the introduction of bubbles leads to a greater decrease of the skin friction coefficient compared to the smooth surface. Clearly, the drag reducing effect of the bubbles is stronger on rough surfaces compared to smooth surfaces, as is evident from the difference between ΔC_f on the fully rough and the fully smooth surface.

Figure 3.5: Results of torque measurements plotted as skin friction coefficient C_f versus (shear) Reynolds number Re_s for a single phase flow with no air bubbles ($\alpha = 0\%$) in the working liquid. Shown in the top graph are the individual results. We also included a linear interpolation between the fully rough and fully smooth data included to arrive at a 56/44 rough/smooth distribution similar to the patched roughness data of certain $\tilde{s}$ (dashed line). Also included are the results of the counter-rotation measurements at $a = 0.36$, where the value of the torque is maximum (black pluses). Shown in the bottom graph are the differences in skin friction coefficient ΔC_f between the different rough and the smooth surface. Errors bars are shown in both graphs, based on the error in the torque sensor and measurement repeatability.

Figure 3.6: Results of torque measurements plotted as skin friction coefficient C_f versus shear Reynolds number Re_s for a two-phase flow with 2 volume percent air bubbles ($\alpha = 2\%$) in the working liquid. Shown in the top graph are the individual results. Shown in the bottom graph are the differences in skin friction coefficient ΔC_f between the flow with $\alpha = 2\%$ and with $\alpha = 0\%$ over the same surface. Also included are the differences in C_f for $\alpha = 2\%$ and $\alpha = 0\%$ of the counter-rotation measurements at $a = 0.36$ (black pluses). Errors bars are shown in both graphs, based on the error in the torque sensor and measurement repeatability.

Figure 3.7: Digitally enhanced photographs of the set up, taken at $Re_s = 0.8 \times 10^6$ and $\alpha = 1\%$. The position of the air bubbles strongly depends on the ratio between rotation ratio $a = -\omega_o/\omega_i$. When $a = 0$, the mixing and axial distribution of bubbles is much better compared to when $a > 0$. For $a > 0$, stable roll structures are present in the flow that trap the bubbles, which is especially visible for the strongest roll structures at $a = 0.36$.

3.3.4 Counter rotating outer cylinder

Shown in figure 3.7 are snapshots of the flow with $Re_s = 0.8 \times 10^6$ and $\alpha = 1\%$ for different values of the rotation ratio a . For $a = 0$, no clear structure can be discovered in the flow. Although effects of buoyancy still play a role, the axial distribution of bubbles is very good compared to the other $a > 0$ cases. When counter-rotation of the outer cylinder is induced to the flow, structure is observed in form of bands in which the bubbles organise themselves. This is best observed for $a = 0.36$ when the rolls are strongest. When the rolls are weaker, for $a = 0.18$ and $a = 0.54$, they are not strong enough to overcome buoyancy effects. When the shear Reynolds number is larger, the rolls will be stronger and buoyancy effects smaller. Here we choose to match the shear Reynolds number between figure 3.3 and figure 3.7 to allow for a direct comparison. When these figures are compared, it also becomes evident that the roughness enhances mixing in the flow, resulting in a better axial distribution of the bubbles. This illustrates the aforementioned trend of a stronger decrease in C_f with Re_s observed in figure 3.6.

The rolls capture the bubbles in their core and keep them away from the inner cylinder, which leaves them useless for drag reducing purposes. This

Figure 3.8: Results of the counter-rotation torque measurements, plotted as skin friction coefficient C_f versus the rotation ratio $a = -\omega_o/\omega_i$ for three different Reynolds numbers. Shown are the results of measurements without air bubbles $\alpha = 0\%$ and with two volume percent of air bubbles $\alpha = 2\%$ in the working liquid. For the two highest Reynolds numbers, the rolls are so strong when $a > 0.2$, that they transport the air bubbles away from the inner cylinder and the drag reduction is lost. For the smallest Reynolds number are buoyancy effects still too strong to allow an even axial distribution of air, resulting in only minor drag reduction.

is shown in figure 3.8 where the skin friction coefficient C_f is plotted versus a for the three different Reynolds numbers for both $\alpha = 2\%$ and $\alpha = 0\%$. Whereas the configuration with the lowest $Re_s = 0.8 \times 10^6$ does show some drag reduction up to $a = 0.6$, for the other two values of Re_s the difference in C_f is quickly reducing when $a > 0$. The differences between the values of C_f at $\alpha = 2\%$ and $\alpha = 0\%$ at $a = 0.36$ are included in the bottom plot in figure 3.6

When we look into the results of Dong (2008), and Huisman et al. (2014), we expect that the turbulent fluctuations in the fluid reaches a maximum around $a \approx 0.36$. This is seen over the whole gap, but the increase is largest in the bulk region of the flow ($\approx 55\%$), and smallest near the walls ($\approx 25\%$) (Dong, 2008). The centripetal Froude number Fr_{cent} will therefore also increase more in the bulk than near the walls, when counter-rotation is introduced, resulting in more trapping of bubbles in the bulk region.

3.4 Discussion and conclusion

The torque measurements presented in the bottom plot of figure 3.6 tell us that air bubble drag reduction is more effective over sandpaper rough surfaces, compared to smooth surfaces. In particular when comparing this to our previous work, where we showed that riblet (obstacle) roughness on the inner cylinder of a Taylor–Couette set up reduces air bubble drag reduction (Verschoof et al., 2018a), it is more than obvious that the type of roughness is of large influence. For both types of roughness, secondary flows in the form of rolls are induced. However, for the sandpaper roughness the bubbles prefer to stay close to the roughness, instead of being carried away from the surface by the induced secondary flows. Therefore bubbles are also not suited to be used as tracer to visualise the flow over these kinds of rough walls. In the case of riblet roughness, the induced rolls do carry the bubbles away from the surface (Verschoof et al., 2018a). When we induce rolls by introducing outer cylinder counter-rotation to a rotating smooth inner cylinder, we find almost no drag reduction from the introduction of air bubbles to the flow. So in this case it is also the secondary flow structures that transport the air bubbles away from the surface.

As to why the bubbles prefer the regions in the flow near the rough patches, we argue that at these locations the turbulent intensity of the flow is largest. For heavy particles it is known that they accumulate in regions of minimum turbulent intensity, a phenomena known as turbophoresis. When air bubbles are thought of as light particles (compared to the working liquid) they move to the regions of maximum turbulent intensity, where the local pressure will also be lowest (Reeks, 1983; Mazzitelli et al., 2003; Climent et al., 2007; Loisy and Naso, 2017; Elghobashi, 2019; Mathai et al., 2019). As long as this is stronger than the effect of the rolls (secondary flows) that try to move the bubbles away from the inner cylinder, we predict that drag reduction will persist.

The influence of the turbulent intensity on the bubble position shows also up in the centripetal Froude number, that we used to explain the observed positions of the bubbles in flows with rough walls and flows with a counter-rotating outer cylinder. Although we made several assumptions regarding the equal bubble size on rough and smooth surfaces, and we did not take into account the larger flow velocities found in the bulk of the flow above rough surfaces, the 75 % larger Fr_{cent} for the rough surface is convincing. It could be expected that the bubbles are actually smaller above the rough surfaces, which might compensate for the velocity increase that is unaccounted for. Near the wall, the turbulent fluctuations will also be larger than at the mid-gap location where Bakhuis et al. (2020a) did their LDA measurements,

also in accordance with Huisman et al. (2013b) and Berghout et al. (2019).

Deutsch et al. (2004) found in their experiments with rough surfaces in a water tunnel the largest values of DR for the lowest flow velocities. We find for the rough surfaces in the low Reynolds number regime $Re_s < 1.0 \times 10^6$ a strongly increasing DR with Re_s , as can be seen in the bottom plot of figure 3.6. This is attributed to the increased mixing effect with increasing Reynolds number that distributes the bubbles, versus a constant influence of gravity. This is also evident from the definition of the Froude number Fr_b in equation 3.3, since both Re_s and Fr_b scale linearly with the rotation rate of the inner cylinder ω_i . This can therefore be attributed as an effect related to our flow geometry. In the regime of large Reynolds numbers, $Re_s > 1.2 \times 10^6$, where mixing dominates over gravity, we find the drag reduction no longer changes with Re_s . Although the study of Deutsch et al. (2004) is more oriented towards the influence of air bubble injection rate and different surface roughness heights k , this is a surprising difference.

We have shown that bubble drag reduction is more effective on rough surfaces. However, a careful examination of figure 3.5 and 3.6 reveals that the skin friction coefficient of a smooth surface without air bubbles present in the flow ($\alpha = 0\%$), is still lower than that of a rough surface with air bubbles ($\alpha = 2\%$). For industrial applications of air bubble drag reduction in the maritime industry, this means that it remains very important to keep the hull of a ship clean and smooth. An important side note here is that in these kind of practical application, a perfect hydrodynamically smooth surface — like we used in this research — is never encountered, since the cost of such a surface finish is too large for these applications. These surfaces will therefore always feature some roughness. Therefore it is very relevant to realize that for such rough surfaces, although the principles and mechanisms we have learnt and observed from experiments using smooth surfaces are very similar, the roughness does have an influence on the bubble position and the resulting drag reduction, and should not be neglected.

*“Werk hebben is één,
werk houden is twee.”*

Henri L. Bullee

Chapter 4

Bubbly drag reduction using a hydrophobic inner cylinder in Taylor–Couette turbulence¹

Abstract

In this study we experimentally investigate bubbly drag reduction in a highly turbulent flow of water with dispersed air at $5.0 \times 10^5 \leq Re \leq 1.7 \times 10^6$ over a non-wetting surface containing micro-scale roughness. To do so, the Taylor–Couette geometry is used, allowing for both accurate global drag and local flow measurements. The inner cylinder – coated with a rough, hydrophobic material – is rotating, whereas the smooth outer cylinder is kept stationary. The crucial control parameter is the air volume fraction α present in the working fluid. For small volume fractions ($\alpha < 4\%$), we observe that the surface roughness from the coating increases the drag. For large volume fractions of air ($\alpha \geq 4\%$), the drag decreases compared to the case with both the inner and outer cylinders uncoated, i.e. smooth and hydrophilic, using the same volume fraction of air. This suggests that two competing mechanisms are at place: on the one hand the roughness invokes

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an extension of the log-layer – resulting in an increase in drag – and on the other hand there is a drag-reducing mechanism of the hydrophobic surface interacting with the bubbly liquid. The balance between these two effects determines whether there is overall drag reduction or drag enhancement. For further increased bubble concentration $\alpha = 6\%$ we find a saturation of the drag reduction effect. Our study gives guidelines for industrial applications of bubbly drag reduction in hydrophobic wall-bounded turbulent flows.

4.1 Introduction

Skin friction drag reduction (DR) in turbulent flow is a topic of research that is relevant for many industrial applications. In particular, the maritime industry may benefit from this, since reducing fuel consumption by only a few percent will lead to significant cost savings and reduction of pollutant emission (van den Berg et al., 2007; Ceccio, 2010; Murai, 2014; Park et al., 2014; Gose et al., 2018).

In this work we combine hydrophobic surfaces with two phase flow to study drag reduction, a combination that, to our best knowledge, has not often been studied before, especially not at the high Reynolds numbers Re of up to 1.8×10^6 that we reach. The physics behind this combination is interesting, since both hydrophobic surfaces and (air) bubble injection have shown individually to decrease the skin friction drag. At the same time, by increasing the amount of gas in the liquid, the effectivity and life span of a drag reducing superhydrophobic surface can be increased (Lv et al., 2014; Xiang et al., 2017). Compared to a hydrophilic surface, gas bubbles that impact a hydrophobic surface are more likely to attach to the surface and form a lubricating layer (Kim and Lee, 2017). Although the wall shear stress in our setup is much larger than what the bubbles in the work of Kim and Lee (2017) are exposed to, a possible result is that the number of bubbles close to the wall increases, which is beneficial for bubbly DR. A set of experiments of two-phase flow over a hydrophobic plate up to $Re = 5000$ by Kitagawa et al. (2019) showed two groups of bubbles. One group of medium-sized free bubbles, and a group of small wall-adhered bubbles, that coalesce into large bubbles. Since the bubbles that stick to the plate change the flow close to the plate, they suggest that the hydrophobic plate is likely to experience more friction drag. Based on this reasoning, they suggest that these results should be carefully considered, when air bubble behaviour is controlled using functionalized (hydrophobic) surfaces in bubbly DR applications Kitagawa et al. (2019). Hence, the two methods of drag reduction (bubbly and with hydrophobic surfaces) will influence one-another. However, it is yet unknown whether this is positive or negative for the total combined drag reduction and we want to find this out in this chapter.

We explore the difference in skin friction coefficient between two types of surfaces: a very smooth hydrophilic surface and a more rough hydrophobic surface. The hydrophobic surface is a sheet of porous polypropylene material, commercially available in large quantities. Representative to more practical applications, it has a sponge-like isotropic geometry of distributed

4.2 Overview over prior work

4.2.1 Drag reduction with hydrophobic surfaces

Superhydrophobic surfaces are typically created by combining a hydrophobic chemistry (resulting in low surface energy) with micro or nanoscale asperities on the surface (Li et al., 2007). The top of these asperities are in contact with the liquid, while air is captured between the asperities. This effectively reduces the solid-liquid contact area, partially replacing it with a gas-liquid interface, that locally changes the no-slip boundary condition to a shear-free boundary condition. The gas-liquid interface is supported by the capillary forces, which in general are larger for hydrophobic materials compared to hydrophilic materials of equal geometry. Dependent of chemistry and geometry, a gas-liquid interface can collapse under influence of a pressure or shear force, and transitions into a thermodynamically favoured wetting state. Various types of asperities exist, ranging from structures such as pillars and ridges to pyramids and mushroom-like shapes (Peters et al., 2009; Qi et al., 2009; Park et al., 2014; Domingues et al., 2017). However, such well-defined shapes are expensive and time-consuming to produce. Therefore, larger areas of SH surfaces ($> 100 \text{ cm}^2$) usually have a random roughness structure (Hokmabad and Ghaemi, 2016). We refer the reader to the review article by Li et al. (2007) for a broader introduction to SH surfaces.

An overview of various experimental and numerical studies in the laminar and low Reynolds number (Re) turbulent regime is given in the review article by Rothstein (2010). Under laminar flow conditions, the behaviour of SH surfaces is typically studied in microchannels. Drag reduction is then quantified by defining a slip length, a slip velocity or by a decrease in pressure drop over the channel (Tsai et al., 2009; Haase et al., 2013; Park et al., 2015). As many industrial flows are highly turbulent, it is crucial to study the behaviour of such surfaces in the high Reynolds number flow regime. For marine vessels for example, Reynolds numbers are of the order of $Re = \mathcal{O}(10^9)$.

Superhydrophobic DR in laminar flow only depends on the geometry of the asperities on the surface that set the slip length and determine the slip velocity. For turbulent flows, SH drag reduction also depends on the Reynolds number (Park et al., 2013). With increasing Re , the thickness of the viscous sublayer decreases, which is the most relevant length scale when comparing the geometric features of the superhydrophobic surface (Daniello et al., 2009). In the near-wall region inside the boundary layer of a turbulent flow, the momentum transfer is dominated by molecular interactions, whereas the role of turbulent momentum transfer is negligible. In other words, viscous

(roughness) length scales formed by the porous structure. To study the fully developed turbulence typical for maritime applications, it is desirable to experimentally achieve high Reynolds numbers, and have both the bulk flow and boundary layer in a state of turbulence. To this end, we use the Twente Turbulent Taylor–Couette facility (T^3C) described in van Gils et al. (2011), of which the inner cylinder is made hydrophobic using the porous polypropylene material. This closed system, with an exact energy balance between input (driving of the flow) and output (viscous energy dissipation), allows for accurate measurement of global drag. Due to its excellent optical accessibility, this can be combined with local flow measurements, for instance using particle image velocimetry (PIV), as well as visualisations of the flow structure and the hydrophobic surface using (high-speed) imaging techniques. Air bubbles are introduced to the working liquid to demonstrate the drag reducing effect of the hydrophobic inner cylinder. This combination of the T^3C with a SH inner cylinder and air bubbles in the working fluid, enables us to study hydrophobic bubbly drag reduction at industrially relevant high Reynolds numbers in a well controlled condition, giving a better understanding of the mechanisms involved.

The chapter is organized as follows: In §2 we give an extensive overview of prior work on bubbly drag reduction, drag on hydrophobic surfaces, and drag enhancement of rough walls, as all these effects are crucial to understand the competing effects explored in this chapter. In §3 the experimental methods are described. §4 presents the results and discusses them. The chapter ends with conclusions.

stress dominates over Reynolds stress. Altering this region affects the entire boundary layer and hence the drag. The outer edge of the viscous sublayer is typically given by a distance $y_{\text{vsl}} = 5\nu/u_\tau = 5\delta_\nu$ from the wall, where ν is the kinematic viscosity, and $u_\tau = \sqrt{\tau_w/\rho}$ the friction velocity for wall shear stress τ_w and density ρ . The viscous length scale $\delta_\nu = \frac{\nu}{u_\tau}$ is the usual scaling parameter for nondimensionalization to viscous wall units, indicated by a superscript ‘+’, e.g. $y^+ = y/\delta_\nu$.

In laminar flow, DR is a direct result of the shear-free (slip) boundary condition. An additional effect matters in turbulence, where near-wall turbulent structures are suppressed due to the slip boundary condition, resulting in additional DR (Park et al., 2013). The numerical work of Park et al. (2013) showed that DR increases with the slip length b^+ , defined as the length below the surface where the extrapolated velocity profile reaches zero. When $b^+ \gtrsim 30 - 40$, the drag is not further affected by an increase of b^+ . This length scale corresponds to the outer edge of the buffer layer $5 < y^+ < 30$ (Pope, 2000), where streamwise near-wall vortical structures primarily reside (Park et al., 2013). Both observations point in the direction that these near wall structures are very important for the larger DR that is found for turbulent flows over SH surfaces compared to laminar flow over SH surfaces (Park et al., 2013). In the work of Rastegari and Akhavan (2018) similar conclusions were drawn. A balance was found between the drag reducing mechanisms of superhydrophobic microgrooves and riblets in the form of a slip velocity together with weakened Reynolds shear stress and near wall vortical structures on the one hand, and a drag increase from the interactions between the microtextures and the flow on the other hand. Results from experiments by Daniello et al. (2009) suggest a critical Reynolds number that prompts the onset of DR, which corresponds to the transition to turbulent flow. For their system of streamwise-aligned SH ridges in channel flow, they find no DR in the laminar regime, whereas after the flow has transitioned to turbulent flow, significant drag reduction was found. Hence, the physics behind the onset of DR must be related to the structure of the wall-bounded turbulent flow (Daniello et al., 2009).

We divide the literature on turbulent flow over hydrophobic surfaces in two regimes: low (but still turbulent) Re turbulence ($Re < 10^5$) and high Re turbulence ($Re > 10^5$). In these regimes, a difference between single-phase and two-phase flow can be made, although most of the research so far has focussed on single-phase flow. Note that in single-phase flow, i.e., when no air is actively added to the working liquid, air might be trapped by the SH surface when the surface is submerged in the working liquid. In two-phase flow, gas is actively dispersed by (for instance air) bubble injection to the

working liquid.

Different design rules are suggested in literature for optimal size and spacing of the geometrical features forming the SH surface. In the low Re turbulence regime, authors mainly seem to use, or suggest to use, surfaces with pillar/ridge spacing $w^+ > 1$, or with a roughness parameter $k^+ > 1$. For the high Re turbulence regime, however, the opposite is the case: suggested is $w^+ < 1$, or $k^+ < 1$. The study by Gose et al. (2018) suggests to not only use the normalized roughness k^+ to predict the drag reducing properties of a superhydrophobic surface, but to also include the contact angle hysteresis measured at a pressure higher than atmospheric pressure. This is done to simulate the large pressure fluctuations and high shear rates generated by high Reynolds number flows (Gose et al., 2018). The roughness of the superhydrophobic surfaces they studied varied between $k^+ = 0.2$ and $k^+ = 4.5$, with corresponding drag reduction changing from -90% to 90% . Specifically around $DR = 0\%$, the trend of increasing DR with decreasing k^+ is absent, showing drag reduction for one surface with $k^+ = 1$ and an increase of drag for another surface with $k^+ < 1$. When k^+ was scaled with the roughness parameter and the wetted area fraction, or the high-pressure contact angle hysteresis (370 Pa for a 250 nl), the DR data collapsed to a single curve (Gose et al., 2018).

An overview of the different surface parameters found in literature focussing on DR with SH surfaces is shown in table 4.1.

Low Re turbulence

Using channel flow, Daniello et al. (2009) studied a variety of SH surfaces consisting of streamwise aligned ridges, with varying ridge spacing $w^+ = 1-4$. Over the whole range of $3 \times 10^3 \leq Re \leq 6 \times 10^3$, a DR of 50% was found (Daniello et al., 2009). The dependence of DR on surface feature size has also been studied using Direct Numerical Simulations (DNS) by Martell et al. (2009), finding good agreement to the work of Daniello et al. (2009). More recent DNS of streamwise SH ridges in channel flow by Park et al. (2013), showed a maximum DR when the ridge spacing was similar to the spacing between near-wall turbulent structures $w^+ = 100$. The work of Park et al. (2013) was able to isolate the effect of the SH surface, since it was modelled as a flat surface with an alternating no-slip and no-shear boundary condition. Effects of roughness on the flow that would play a role in experiments, either from a non-flat gas-liquid interface or from surface features that protrude through the viscous sublayer, could therefore be ruled out.

Rather than a surface of well defined feature size and geometry, a porous surface of random roughness structure was used by Srinivasan et al. (2011). The inner cylinder of their Taylor–Couette was made superhydrophobic by spraycoating a mixture of PMMA fibres and low surface energy fluoro-decyl POSS molecules. Nonetheless are the resulting surface roughness parameters similar to that of Daniello et al. (2009). From the work of Srinivasan et al. (2015) we calculate the average roughness height at the maximum $Re = 8 \times 10^4$ to be about $k^+ = 1.5$ and the mean roughness spacing $w^+ = 2.5$. The maximum Re also resulted in the largest DR of 22 %. Another study in Taylor–Couette, of similar Re , but with much larger surface roughness parameters of $k^+ = 27$ and $w^+ = 14$ formed by a SH pillar structure, found only 3 % DR (Panchanathan et al., 2018). When instead of large SH pillars, large streamwise aligned SH ridges were used in Taylor–Couette, an optimal groove spacing of $w^+ = 35$ was found for achieving a maximum DR of 35 % (Van Buren and Smits, 2017). For the smallest groove spacing tested, $w^+ = 2$, no DR was found. The base line drag used in the definition of the drag reduction is very important. Where Van Buren and Smits (2017) used their wetted surface as the baseline, was a smooth surface used for the baseline drag by Panchanathan et al. (2018). When the baselines are defined equally, the difference in DR found between both studies will be much smaller.

High Re turbulence

Ling et al. (2016) measured the velocity in the inner part of the turbulent boundary layers over SH surfaces subjected to single-phase flow. Surfaces were made SH by means of spraycoating, resulting in a random oriented roughness, and by etching and coating, giving both ridges and random oriented roughness. Measurements were done in a water tunnel, operated at $1 \times 10^5 \leq Re \leq 3 \times 10^5$. Their results revealed a delicate balance between the contribution of viscous stresses and Reynolds stresses to the wall shear stress. This balance determines whether DR is found (viscosity dominates), or the surface roughness increases the drag (turbulence dominates). It was found that when the roughness $k^+ \gtrsim 1$, the Reynolds stresses become the main contributor to the wall shear stress, and less DR was found (Ling et al., 2016).

The number of studies we found that combine a SH surface and air injection (two-phase flow) is limited. Du et al. (2017) only found DR when air was being injected through their SH surface. The DR was the result of weakened near-wall vortices, pushed away from the SH surface, and smaller shear rates on top of the SH surface (Du et al., 2017). A variety of flow geometries was studied by Fukuda et al. (2000): rectangular pipe flow ($5 \times$

Author	Phase	Flow type	SH surface	$Re [\times 10^3]$	$DR_{\max}$	Surface roughness
Srinivasan (2011)	single	TC	random	1.6 – 80	22 %	$k^+ = 1.4^a$ & $w^+ = 2.5^a$
Rastegari (2018)	single	ch. (LB)	grooves	3.6	61.1 %	$w^+ = 8^b$
Daniello (2009)	single	ch.	ridges	3.0 – 6.0	50 %	$w^+ > 1^c$
Martell (2009)	single	ch. (DNS)	ridges % posts	4.2 ^a	40 %	w^+ as large as poss.
Rosenberg (2016)	single	TC	triang. ridges	6.0 – 9.0	10 %	$k^+ = 3.9$ & $w^+ = 5.6$
van Buren (2017)	single	TC	ridges	6.0 – 10	9 %	$w^+ = 35$
Park (2013)	single	ch. (DNS)	ridges	4.2 – 17	90 %	$w^+ = 100^c$
Gose (2018)	single	ch.	random	10 – 30	90 %	$k^+ < 0.5^{c,d}$
Panchanathan (2018)	single	TC	square pillars	47	3 %	$k^+ = 13.6^a$ & $w^+ = 27.1^a$
Du (2017)	dual	ch.	random	120	20 %	not enough info
Fukuda (2000)	dual	ch.	random	50 – 400	50 % ^e	no info
		flat plate		300 – 17000	50 % ^e	
		ship model		900 – 800000	20 %	
Aljallis (2013)	single	flat plate	random	300 – 3000	30 % ^f	no info
Ling (2016)	single	ch.	random + ridges	100	36 %	$k^+ = 0.68^c$
Park (2014)	single	ch.	ridges	100 – 1000	75 %	$w^+ < 1^c$
Reholon (2018)	single	cylin. obj.	random	500 – 1500	36 % ^e	$k^+ = 0.5$
Bidkar (2014)	single	ch.	random	1000 – 9000	30 %	$k^+ < 0.5^c$

^a derived from data in paper ^b normalized with value for no-slip surface ^c optimal value obtained from parameter sweep ^d additional surface characterization ^e decreasing with Re ^f no DR for $Re_L > 10^6$

Table 4.1: Overview of literature on drag reduction (DR) of turbulent flows over superhydrophobic (SH) surfaces, illustrating different surface design parameters k^+ and w^+ corresponding to the largest drag reduction found by different authors. The horizontal line separates the *low Re turbulence* from the *high Re turbulence* as introduced in section 2. TC abbreviates Taylor–Couette, ch. stands for channel flow.

$10^4 \leq Re \leq 4 \times 10^5$), flat plate ($3 \times 10^5 \leq Re_L \leq 1.7 \times 10^7$), and ship models in a towing tank ($9 \times 10^5 \leq Re_L \leq 8 \times 10^8$). For the pipe flow and flat plate experiments, the maximum DR of 50 % was found to decrease with Re to ~ 0 %. Negligible influence of an increased air injection rate on DR was observed for all flow geometries (Fukuda et al., 2000).

One of the few experiments in the high Re turbulence regime that uses a surface with a geometrically well defined pattern is done by Park et al. (2014) ($1 \times 10^5 \leq Re \leq 1 \times 10^6$), allowing for a direct comparison to the work of Daniello et al. (2009) ($3 \times 10^3 \leq Re \leq 6 \times 10^3$) in the low Re turbulence. Both studies made use of a fully turbulent, single-phase channel flow over a surface of streamwise SH ridges. Daniello et al. (2009) suggested an optimum ridge spacing of $w^+ = 5$, which is equal to the size of the viscous sublayer. Park et al. (2014) however, found their maximum DR for $w^+ < 1$. This is a difference typically found between studies in the low- and the high Re turbulence regime, as can also be seen in table 4.1.

4.2.2 The air plastron

The air layer captured between the SH surface and the water is commonly referred to as the air plastron. When the SH surface transits from a non-wetted Cassie-Baxter state to a wetted Wenzel state, the plastron and the DR are lost. Since the Wenzel state is typically the thermodynamically more favoured state, it is therefore crucial to prevent or delay this transition. This can for instance be achieved by reducing the size (diameter or w^+) of the asperities in which the gas is trapped to increase the Laplace pressure, or by increasing the hydrophobicity of the surface. The diffusion of gas from the plastron into the liquid is another factor to minimize in order to sustain DR, which can for instance be achieved by increasing the amount of saturated gas in the liquid (Lv et al., 2014; Xiang et al., 2017).

In the experiments by Srinivasan et al. (2015), the SH surface was not fully submerged, resulting in a connection between the plastron and the air present in the room. More DR (22 %) was found compared to the case where the air-layer is isolated (DR = 15 %) for the same Re (Srinivasan et al., 2015). When the surface is exposed to flow, the loss of plastron volume can be described by a convection-diffusion mechanism. Larger flow velocities give shorter effective diffusion lengths, resulting in an accelerated transport of gas from the plastron into the the liquid (Xiang et al., 2016). Video recordings of the plastron exposed to turbulent flow ($5.0 \times 10^5 \leq Re_L \leq 1.5 \times 10^6$) showed constant movement and variations in the thickness of the plastron, caused by pressure fluctuations in the turbulent boundary layer (Reholon and Ghaemi, 2018). Du et al. (2017) found DR when injecting air through a pinhole in

their SH surface. The amount of injected air was not enough to form an air bubbly flow, but enough to maintain a plastron that was thick enough to prevent the surface roughness features from contacting the liquid. When the air injection was stopped, the air plastron became thinner, and roughness effects started to play a role (Du et al., 2017). When the roughness elements are exposed to the flow, the Reynolds stresses become the main contributor to the wall shear stress, resulting in less DR (Ling et al., 2016).

For this particular reason, Gose et al. (2018) suggested to measure the surface characteristic contact angle hysteresis under higher than ambient pressures. Also mechanical interactions between the plastron and solid pollutants in the liquid phase can decrease the plastron stability. Collisions between particles added to the flow and the plastron were shown to shorten its lifetime by about 50 % (Hokmabad and Ghaemi, 2017). Once the air plastron is destroyed and the surface has transited to the wetted state, energy is required to reverse the surface to the un-wetted state. Different studies explored for instance film boiling, water splitting by electrolysis and the injection of air bubbles into the boundary layer (dual-phase flow) to achieve this (Saranadhi et al., 2016; Panchanathan et al., 2018).

4.2.3 Bubbly drag reduction

The introduction of air bubbles to a flow can also result in reduced skin friction drag. A typical approach is to inject air bubbles close to (or in) the boundary layer. We refer to the review articles by Ceccio (2010) and Murai (2014) for an overview of various studies on bubbly DR. Early air-lubrication DR experiments, in which gas micro-bubbles were injected (or created) in the (turbulent) boundary layer, showed an increase in DR with increasing air injection rate, and a decrease in DR with increasing Reynolds number (McCormick and Bhattacharyya, 1973; Madavan et al., 1985; Watanabe et al., 1998). Up to 80 % reduction of skin friction drag using microbubble injection was reported (Madavan et al., 1984). This DR was attributed to a thickening of the viscous sublayer (so a smaller gradient in the velocity profile near the wall) caused by the microbubbles that were present in the near-wall buffer layer (Ceccio, 2010).

For Taylor–Couette flow, for low Re and microbubble injection, the drag reduction was shown to be due to the weakening or even destruction of the Taylor-rolls, due to the rising microbubbles. This gravity effect (controlled by the Froude number) indeed decreases with increasing Reynolds numbers (Sugiyama et al., 2008; Lohse, 2018). More recent research showed the influence of the bubble size on DR, concluding that the existence of large, deformable bubbles, i.e. those that have a large Weber number, is crucial

for drag reduction in high Re turbulent flows (Lu et al., 2005; van den Berg et al., 2005; van Gils et al., 2013; Verschoof et al., 2016; Spandan et al., 2018). In these papers, the increase of the Weber number with increasing Re is used to explain the enhanced bubbly DR that is typically found for larger Re (van den Berg et al., 2005).

Although the principle of air bubbly DR is not yet fully understood, it is clear that the effect is largest when the bubbles are close to, or in, the boundary layer. For flat plate experiments, the skin friction bubbly drag reduction is commonly limited to the first few metres downstream of the air injector (Watanabe et al., 1998; Sanders et al., 2006). Further downstream, turbulent diffusion causes bubbles to move away from the wall (Murai, 2014). A similar mechanism was observed in Taylor–Couette flow, where strong secondary flows transport bubbles away from the inner cylinder, resulting in a decrease of DR (van den Berg et al., 2007; Fokoua et al., 2015; Verschoof et al., 2018a).

4.2.4 Roughness

To create a SH surface, some form of roughness has to be introduced on the surface, to facilitate an asperity where air can be trapped. Since any form of surface roughness increases the drag on a wall-bound flow, we therefore deal with opposing effects in drag reduction using SH surfaces: drag reduction due to air (plastron) lubrication and drag increase from the added roughness. We refer to the reviews of Jiménez (2004) and Flack and Schultz (2010) for a comprehensive overview of studies towards the influence of roughness on turbulent flows.

Three different roughness regimes are distinguished. In the hydrodynamically smooth regime, when the equivalent sand roughness is less than the thickness of the viscous sublayer ($k^+ < 5$), the surface can be regarded as smooth (Schlichting and Gersten, 2000). The perturbations in the flow that are generated by the roughness features of the surface are completely damped out by the viscosity (Flack and Schultz, 2014). When the roughness k^+ increases, parts of it will extend through the viscous sublayer, corresponding to the transitionally rough regime ($5 \leq k^+ < 70$). The log-law that describes the velocity profile close to the wall shifts inwards, maintaining its shape, but reduced in magnitude. The mean velocity profile in the bulk of the flow however stays unaffected by the roughness (Flack and Schultz, 2014). Hence, universality is only seen for the larger length scales of the flows (Pope, 2000).

The wall shear stress in the transitionally rough regime is composed of a combination of viscosity and pressure drag on the roughness elements. With

increasing roughness height, the contribution of pressure drag increases (Verschoof et al., 2018b). In the fully rough regime ($k^+ \geq 70$), the pressure drag heavily dominates over viscosity. As a result, the shift in the log law (the roughness function ΔU^+), scales linearly with k^+ , and the skin-friction coefficient becomes independent of Re (Flack and Schultz, 2014).

Although the size of the roughness k^+ gives a good indication for the state of roughness: hydrodynamically smooth, transitionally rough or fully rough, which also depends on the geometry of the roughness. For instance, a stepwise geometry that consists of steep slopes will transit to the fully rough regime at smaller k^+ than a roughness of more gentle slopes (Busse et al., 2017). Similarly, a surface of very closely packed roughness elements (high solidity), or a surface where the roughness elements are sparse (high porosity) will behave more like a surface of smaller k^+ (MacDonald et al., 2016). In the context of Taylor–Couette turbulence, roughness effects were analysed by Zhu et al. (2018) and Berghout et al. (2019), who found the same universal $\Delta U^+(k^+)$ for the velocity reduction as was found by Nikuradse (1933) for pipe flow.

4.3 Experimental method

4.3.1 Experimental setup

All experiments were performed in the Twente Turbulent Taylor-Couette (T³C) facility described in van Gils et al. (2011) and shown in figure 4.1. It consists of two independently rotating concentric cylinders of length $L = 0.927$ m. The inner cylinder is fabricated from grade 316 hydrophilic stainless steel. The outer cylinder is cast from clear acrylic, which allows for full optical access to the flow between the cylinders. The inner radius of the outer cylinder is $r_o = 0.279$ m and the outer radius of the inner cylinder equals $r_i = 0.200$ m, thus the radius ratio is $\eta = r_i/r_o = 0.716$. The resulting gap has a width $d = r_o - r_i = 0.079$ m and was filled with fully air saturated deionized water. For inner cylinder rotation, the Reynolds defined, based on the gap width and velocity of the inner cylinder is

$$Re = \frac{\omega_i r_i d}{\nu}, \quad (4.1)$$

where ω_i is the angular velocity of the inner cylinder and ν is the kinematic viscosity of the working fluids. The inner cylinder rotates at frequencies in the range $\omega_i = 5\text{--}18$ Hz, while the outer cylinder is kept stationary. Typical values used in this research range from $Re = 5 \times 10^5$ to $Re = 1.8 \times 10^6$. The system is actively cooled to keep the temperature of the working fluid at $21^\circ\text{C} \pm 0.5^\circ\text{C}$.

By partly filling the apparatus, as in figure 4.1, we vary the volume fraction of air α in the working fluid from 0 % to 6 %. The turbulence mixes the air and water, generating bubbles that are distributed over the height and over the gap between cylinders (van Gils et al., 2013).

4.3.2 Torque measurements

The inner cylinder is composed of three sections. The torque exerted by the fluid on the inner cylinder is measured using a Honeywell 2404-1K hollow reaction torque sensor that is placed inside the middle section of the inner cylinder as indicated in figure 4.1. Only the torque on the middle section of length $L_{\text{mid}} = 0.536$ m is taken into account to reduce end-plate effects between the rotating lid of the inner cylinder and the stationary lid of the outer cylinder. We express the torque in non-dimensional form using the skin friction coefficient:

$$C_f = \frac{\mathcal{T}}{L_{\text{mid}} \rho \nu^2 Re^2} \quad (4.2)$$

Figure 4.1: Schematic overview of the measurement setup. Shown are the outer and inner cylinder, of which the latter consists of three sections. The middle section is connected to the driving shaft by means of a torque sensor, which is also shown in the figure. The gap between the two cylinders $r_o - r_i$ is filled with water and air, of which the quantity of the air is expressed by means of a void fraction α , ranging between 0% and 6%. When the inner cylinder is rotating ($\omega_i > 0$ Hz), bubbles are formed and distributed in radial and axial direction over the gap due to turbulent mixing. Particle Image Velocimetry (PIV) measurements can be done only when there are no bubbles present in the working liquid ($\alpha = 0\%$). The PIV lasersheet is placed at cylinder mid-height and the flow is observed through a window in the bottom-plate using a mirror and a camera.

where $\mathcal{T}$ denotes the torque, ρ and ν are the density and kinematic viscosity, respectively, of the working fluid.

The drag reduction for the hydrophobic coating and the hydrophilic reference is determined using equation (4.3).

$$\text{DR}(\alpha) = 1 - \frac{C_f(\alpha)}{C_f(\alpha = 0)} \quad (4.3)$$

This shows the influence of adding bubbles to the flow on the drag. The difference in drag reduction ΔDR between a hydrophobic inner cylinder (IC) and a hydrophilic IC is defined as

$$\Delta\text{DR}(\alpha) = \text{DR}_{\text{hydrophobic}}(\alpha) - \text{DR}_{\text{hydrophilic}}(\alpha) \quad (4.4)$$

In order to provide insight into the influence of the hydrophobic IC on the flow, we define a *net* drag reduction as

$$\text{DR}_{\text{net}}(\alpha) = 1 - \frac{C_{f,\text{hydrophobic}}(\alpha)}{C_{f,\text{hydrophilic}}(\alpha = 0)} \quad (4.5)$$

Here $C_{f,\text{hydrophobic}}(\alpha)$ is the skin friction coefficient for the hydrophobic IC and different values of α , and $C_{f,\text{hydrophilic}}(\alpha = 0)$ is the skin friction coefficient for the smooth hydrophilic inner cylinder, without air bubbles present in the flow ($\alpha = 0$). In equation (4.3) the focus is only on the influence of bubbles on the drag, with either a hydrophobic or a hydrophilic IC, whereas equation (4.5) show the influence of both drag reducing measures: bubbles and a hydrophobic coating.

4.3.3 hydrophobic coating

In the hydrophobic case, the IC of the T³C is fully coated with a 3M Membrana Accurel[®] PP 2E HF flatsheet membrane. This porous hydrophobic polypropylene material is commercially available in the large quantities that are needed to cover the complete IC. This coating is supplied on rolls that have a width of about 27 cm. It is attached to the inner cylinder using double-sided adhesive tape. From visual inspection it is estimated that 99 % of the IC is covered by the coating, see figure 4.4.

We used a Dataphysics OCA 15EC device to measure the contact angle hysteresis for both the hydrophobic coating and the reference case, which is the hydrophilic, uncoated steel IC. The hydrophobic coating has advancing and receding water contact angles of $152^\circ \pm 2^\circ$ and $120^\circ \pm 5^\circ$, respectively. For the hydrophilic IC, an advancing contact angle of $93^\circ \pm 2^\circ$ was found. The receding contact angle is set at 10° , which is the lowest angle the setup could measure.

Figure 4.2: SEM photos of the side of the coating that is exposed to the flow. The top row focuses on a region composed of smaller pores. The bottom row shows a region with larger pores.

4.3.4 Roughness

The machining process to fabricate the standard hydrophilic IC gives a surface roughness of $k_{ic} = 1.6 \mu\text{m}$. The viscous length scale δ_ν is derived from the measured torque data, as discussed in (Huisman et al., 2014). For the maximum Reynolds number $Re_{\max} = 1.8 \times 10^6$ used in this research, the viscous length scale reaches its lowest value of $\delta_\nu = 1.9 \mu\text{m}$. The resulting roughness in wall units $k_{ic}^+ \approx 0.8$. Therefore, the uncoated hydrophilic IC can be assumed to be a hydrodynamically smooth surface.

The average roughness of the coating is analysed from its pore size using 24 different SEM images, made using three different magnifications, as shown in figure 4.2. The coating consists of an isotropic sponge-like structure, meaning that the cross section looks similar to the top and bottom surface. Therefore we use SEM images of the top surface to evaluate the size and roughness of the pores. The SEM images show a distribution of pore sizes, in the range of 1–10 μm . The pores that correspond to the smaller length scale are found in regions separated by pores of the larger length scale. The size distribution is quantified with the image processing program ImageJ, using edge detection of the thresholded image (Analyse Particles tool). Eight different images of the smallest magnification were used for this. The images were pre-processed by subtracting a sliding background and by applying a local mean threshold algorithm. Eroding and dilation was used to remove small scale noise. It is difficult to define the error for the

Figure 4.3: Pore size diameter D_p (roughness) distribution of the coating as fraction of coverage A of total area of the coating A_0 . A range of length scales is observed, corresponding to the different regions identified in figure 4.2. The equivalent circle diameter has been used as a measure of the size $D_p = 2\sqrt{A_p/\pi}$.

pore size distribution, since it is difficult to evaluate the edge of a pore from the perspective of the flow. For instance in figure 4.2, when inspecting the large pores at highest magnification, we see thin thread-like fibres that span across a pore. Whereas in the image analysis this might be detected as an edge, the flow might experience this differently. However, from the SEM images and the pore size distribution it is clear that multiple roughness length scales are present on the surface of the coating. Hence, dependent of Re , a larger or smaller fraction of the surface plays a role in influencing the flow. The resulting distribution of binned pore sizes is shown in figure 4.3. The combined area of all pores with diameter D_p over the total area, the fraction A/A_0 , is plotted versus D_p . A maximum is found at $D_p = 2.5 \mu\text{m}$, although the larger length scales that are more relevant to the flow are also found.

4.3.5 Experimental procedure

During a measurement period of one hour, the IC was accelerated in steps from 5 Hz to 18 Hz, corresponding to a range of Re between 5×10^5 and 1.8×10^6 , while continuously measuring torque exerted by the fluid on the

Surface	α [%]	ω_i [Hz]	Measurements
hydrophobic	0	5–18	2
	2	5–18	4
	0	5–18	2
	4	5–18	4
	0	5–18	2
	6	5–14	4
	0	5–18	2
hydrophilic	0	5–18	2
	2	5–18	3
	4	5–13.4	3
	4	15.8–18	3
	6	5–14	3

Table 4.2: Overview of the measurement parameter space, in order of execution. Between changing the volume percentage of air α , the reference case of $\alpha = 0\%$ air was measured twice, to account for changes to the coating caused by the flow itself. Deviations from the standard frequency range $\omega_i = 5\text{--}18\text{ Hz}$ were the result of heavy vibrations in the system, forcing us to skip a certain frequency range.

inner cylinder. Every variation of a hydrophobic IC with $\alpha > 0\%$ was measured four times. Between changing the volume percent of air α , the reference case of $\alpha = 0\%$ air is measured twice, to account for changes to the coating caused by the flow itself. An overview of the measurements is shown in order of execution in table 4.2.

4.3.6 Flow visualisation

A Nikon D800E camera was used to capture still images of the flow. This provides insight on the presence of an air plastron: air captured by the coating is visible in the form of a silvery reflection on the surface Shirtcliffe et al. (2006); McHale et al. (2009); Daniello et al. (2009); Poetes et al. (2010); McHale et al. (2011); Dong et al. (2013); Park et al. (2014, 2015); Saranadhi et al. (2016). This can be seen in figure 4.4, where the highlighted area points out locations where the incident light is of the right angle to see the plastron. It was found that an air plastron was present during the measurements featuring a hydrophobic IC. To test the stability of the plastron and force the surface in a wetted state, the surface tension was lowered by adding TritonX surfactant whilst rotating the inner cylinder at $Re = 1.0 \times 10^6$

with single phase flow conditions. From image analysis it was determined that only after 1.5×10^{-4} mol/l TritonX was added, the silvery reflection had completely dissapeared. At this concentration of TritonX the surface tension is about 35 mN/m (Göbel and Joppien, 1997). So both surface tension and Laplace pressure are about half the value of that for pure water.

Velocity profile measurements

Particle Image Velocimetry (PIV) was used to obtain local flow field information. We measured the velocity field in the (r, θ) plane; $u_\theta = u_\theta(r, \theta, t)$ and $u_r = u_r(r, \theta, t)$. This can only be achieved for single-phase flow with $\alpha = 0$, since the air bubbles otherwise scatter the light significantly. The laser light sheet (Quantel Evergreen 145 laser, 532 nm) used to illuminate the seeding particles added to the flow (Dantec fluorescent polyamide, with a distribution of diameters $\leq 20 \mu\text{m}$) was placed at mid-height of the cylinder. Images were captured using a LaVision sCMOS (2560×2160 pixel) camera through the window in the bottom plate of the setup. Figure 4.1 gives a schematic overview of the measurement setup. Average velocity fields were calculated from 1000 image pairs using LaVision DaVis software in a multi-pass method, starting at a window size of 64×64 pixel decreasing to a final size of 24×24 pixel with 50 % overlap. A calibration is required to transform pixels to meters. To this end image analysis is used to locate the edges of the inner and outer cylinder. Since the measured fields are in Cartesian coordinates, a coordinate transformation is necessary to obtain finally the radial and azimuthal velocities u_r and u_θ in the cylindrical coordinate system.

Figure 4.4: Digitally enhanced photograph of the inner cylinder, covered with the hydrophobic coating and visible through the transparent outer cylinder. Estimated is that 99% of the inner cylinder is covered by the hydrophobic coating. The silvery reflection that is typically associated with the presence of an air plastron, is visible as a darker shaded region. This plastron can only be observed under certain angles of incident light. The curved surface of the inner cylinder explains why the plastron is only visible in a narrow vertical band.

4.4 Results and discussion

4.4.1 Torque measurements

First, the results of the torque measurements single-phase flow are presented and discussed, where drag reduction (DR) is purely the result of a hydrophobic surface capturing an air plastron. Second, we show the results for two-phase flow, where air bubbles are added to the flow, that provide bubbly DR and might also add to the stability of the air plastron.

Single-phase flow

In the top of figure 4.5, the drag reduction (DR) as defined in equation (4.5) is plotted versus Re , for the hydrophobic inner cylinder (IC) with $\alpha = 0$. This shows an increase in the drag of about 14% over the whole range of Re measured. The bottom figure shows the evolution of the thickness of the viscous sublayer ($y^+ = 5$), the viscous length scale and the design parameter suggested by Park et al. (2014) ($y^+ = 1$), and the design parameter suggested by Bidkar et al. (2014) ($y^+ = 0.5$), with the Reynolds number. In figure 4.6 we show the roughness of the surface expressed in wall units for four different values of Re : the minimum, the maximum and two intermediate values, for which we use the same data as in figure 4.3. For the lowest $Re = 0.5 \times 10^6$, the majority of the roughness length scales is less than $k^+ = 1$ and part of it is even less than $k^+ = 0.5$. However, over the whole range of Re measured we find a constant increase of the drag by about 14%. For a rough, wetted surface, an increase of drag with Re is expected (Flack and Schultz, 2010). However, we do not observe wetting of the surface, which would manifest itself through disappearance of the silvery reflection that indicates the presence of an air plastron. Hence, we assume the surface to maintain its Cassie-Baxter state throughout the course of the experiments. Nonetheless, even for a wetted surface, an increase in C_f is very surprising, given the average value of $k^+ < 5$, which would indicate a hydrodynamically smooth surface following (Schlichting and Gersten, 2000). A similar result of drag increase with $k^+ < 5$ was also reported by Gose et al. (2018). Contrary to the results observed of these authors of increasing DR with decreasing k^+ for the same surface, we find a more or less increased constant drag over the whole range of our values of Re and hence k^+ . Our results might be explained by following the analysis of Gose et al. (2018), who state that the value of the k^+ roughness alone is not sufficient to predict the DR of a superhydrophobic surface. The work of Reholon and Ghaemi (2018) showed less DR at larger Re due to a thinner and fragmented plastron. However,

we find the DR to be more or less constant at -14% . When the surface is forced into a wetted state by lowering the surface tension from the addition of TritonX, no significant difference in drag is found. Obviously, the surface roughness strongly influences the drag even in the non-wetted state.

Two-phase flow

In figure 4.7, DR is shown as defined in equation (4.3), comparing results for a *hydrophobic* inner cylinder to a *hydrophilic* inner cylinder, plotted versus Re . For both hydrophobic and hydrophilic cases, $\alpha = 0\%$ is the reference case for determining the level of drag reduction that results from the introduction of bubbles ($\alpha > 0\%$) to the flow. As an additional effect, the introduction of air bubbles to the working liquid might also lead to enhanced stability of the air plastron Lv et al. (2014). With increasing Re , more DR is found for all measurements. This is in line with the previous findings (van den Berg et al., 2005; van Gils et al., 2013; Spandan et al., 2018).

Comparing the hydrophobic IC to the hydrophilic IC, more bubbly DR is found (figure 4.7) over nearly the whole range of Re when $\alpha \geq 4\%$. Only when $Re < 7 \times 10^5$, a slight increase of drag is found for $\alpha = 6\%$. In this range of low Re , the uncertainty of the torque sensor is the highest, as can be seen from the shaded areas in figure 4.7 that give an indication of the repeatability of the experiments by showing the spread in the data by comparing the extremes of individual measurements. For the smallest amount of air added, $\alpha = 2\%$, the hydrophobic IC gives more DR compared to the hydrophilic IC up to $Re = 10^6$. For larger Re , between 1.0×10^6 and 1.8×10^6 , the hydrophobic IC gives less DR compared to the hydrophilic IC.

We explain the difference between $\alpha = 2\%$ and $\alpha \geq 4\%$ in figure 4.7 as a result of two competing effects: (i) a more effective bubbly drag reduction in the presence of a hydrophobic wall, and (ii) a drag increase due to the roughness of this same hydrophobic wall. When the thickness of the viscous sublayer $y_{\text{vsl}} = 5y^+$ decreases with increasing Re , a larger fraction of the pores on the surface of the hydrophobic IC are of a length scale relevant to the flow, as can be seen in figure 4.5. The balance between these two competing effects determines whether the bubbly DR is more effective when using a rough hydrophobic IC compared to a smooth hydrophilic IC. For larger void fractions $\alpha \geq 4\%$, the more effective bubbly DR dominates, resulting in more overall DR.

Figure 4.8 shows the difference ΔDR between *hydrophobic* and *hydrophilic* from figure 4.7, as defined in equation (4.4). The difference in ΔDR between $\alpha = 4\%$ and $\alpha = 6\%$ in figure 4.8 is small compared to the difference between $\alpha = 2\%$ and $\alpha = 4\%$, or $\alpha = 2\%$ and $\alpha = 6\%$. This suggests that

Figure 4.5: Top figure: plot of the drag reduction (DR) as in equation (4.5) versus Re , with $\alpha = 0$ for the hydrophobic inner cylinder. Bottom figure: evolution of the thickness of the viscous sublayer ($y^+ = 5$), the viscous length scale ($y^+ = 1$) and half the viscous length scale ($y^+ = 0.5$) with Re . The design parameters $w^+ < 1$ and $k^+ < 0.5$ for the hydrophobic surface are suggested by Park et al. (2014) and (Bidkar et al., 2014) respectively to result in drag reducing behaviour of the surface and are shown here as a reference for the reader. These values are derived from the torque measurements and give therefore an averaged, global value. From figure 4.6 we find that for our lowest Re tested, the majority of the roughness length scales is below $k^+ = 1$ and part of it is below $k^+ = 0.5$. The DR plot however shows a nearly constant increase of the drag by about 14% over the whole range of Re measured.

Figure 4.6: Roughness distribution of the surface as coverage fraction of the total area A/A_0 , expressed in wall units for four different values of Re . Apart from the maximum and the minimum value of Re used in this research, the normalized roughness for two intermediate values of Re are shown as well. The wall unit normalization is obtained using data from the torque measurements that gives an averaged, global value of the wall shear stress.

the influence of the hydrophobic IC on the bubbly DR is limited, i.e. that a minimum amount of air is required to effectively reduce the drag — here $\alpha = 4\%$ — but further increasing α will not result in an even stronger influence of the hydrophobic IC on the bubbly DR. Nor does ΔDR increase with Re for $Re > 10^6$ and $\alpha \geq 4\%$, but rather levels off and starts to decrease. This indicates that of the two effects, the more effective DR from the hydrophobic wall initially grows faster with Re , until ΔDR is maximum. With further increasing Re , around $Re = 1.1 \times 10^6$, the balance starts to tilt and the drag increase from the roughness is now the faster growing effect, indicated by the negative slope of ΔDR for $\alpha = 4\%$ and $\alpha = 6\%$ in figure 4.8. It is tempting to attribute this to wetting of the coating, caused by the wall shear stresses that become larger with Re . However, if this were the case, a clear difference would be visible between the initial, and its repeated measurements for the same α , since the wetted state is the energetically more stable one. It is safe to assume that if the coating – or part of it – is wetted, it will not transit back to a non-wetted state between measurements. The reason for the change in balance should therefore be sought elsewhere.

The definition of DR_{net} in equation (4.5) is used in figure 4.9 to study the combined effect of both air bubbles ($\alpha > 0\%$) and a hydrophobic IC, using the hydrophilic IC with $\alpha = 0\%$ as a reference for all cases. Compared to figure 4.7, all data obtained using the hydrophobic IC is shifted downwards by roughly 15%. For the lines $\alpha = 0\%$ and $\alpha = 2\%$, the shift in DR_{net} is nearly constant, whereas for $\alpha = 4\%$ and $\alpha = 6\%$ the difference in DR_{net} goes down with increasing Re , owing to the same competing mechanisms as discussed previously. Nonetheless it is clear from figure 4.9 that for all values of α , the use of a hydrophobic IC results in a less efficient net DR compared to a hydrophilic IC. The introduction of the hydrophobic coating adds roughness to the otherwise hydrodynamically smooth hydrophilic surface, which in this case explains the significant change in the net drag force.

4.4.2 Repeatability and measurement errors of torque measurements

For the highest Reynolds number achieved in this research, namely $Re = 1.8 \times 10^6$, the shear stress at the surface $\tau_w = \mathcal{T}/2\pi r_i^2 L_{\text{mid}}$ is 274 Pa. To ensure that the hydrophobic coating applied to the IC is not adversely affected by this high shear stress, we measured the $\alpha = 0\%$ reference measurement twice after every series of $\alpha > 0\%$ measurements, as indicated in table 4.2. The skin friction coefficients found in the $\alpha = 0\%$ measurements have a spread of 3%. This spread shows no trend, which suggests that the prop-

Figure 4.7: Plot of the drag reduction (DR) based on skin friction coefficient C_f as defined in equation (4.3) versus Re . Compared to figure 4.9, is the drag reduction here determined using the same IC as used for the $\alpha > 0$ measurement (so either hydrophobic or hydrophilic) with $\alpha = 0$ ($C_{f,0}$), whereas in figure 4.9 the reference is the hydrophilic IC with $\alpha = 0$. A more efficient bubbly drag reduction mechanism is found for the hydrophobic coating when the void fraction $\alpha \geq 4\%$. For $\alpha = 2\%$ roughness effects dominate, resulting in less overall DR. Only every second data point is shown to improve readability of the plot. The shaded regions represent the spread in data. The size of the error bars based on the accuracy of the torque sensor is smaller than the marker size. Due to heavy vibrations in the setup resulting from a non-symmetric distribution of air, no data was acquired in the region between $Re = 1.3 \times 10^6$ and $Re = 1.6 \times 10^6$ for $\alpha = 4\%$ and for $Re > 1.4 \times 10^6$ when $\alpha = 6\%$.

Figure 4.8: Plot of $\Delta DR = DR_{\text{hydrophobic}} - DR_{\text{hydrophilic}}$ from figure 4.7 versus Re . For $\alpha = 2$, ΔDR decreasing with increasing Re , owing to the influence of roughness. For $\alpha \geq 4$, ΔDR shows an increase with Re , meaning that the effect of increase in bubbly drag reduction due the hydrophobic coating is stronger than the effect of roughness.

Figure 4.9: Plot of the net drag reduction (DR_{net}) based on skin friction coefficient C_f (equation (4.5)) versus Re . Compared to figure 4.7, is the drag reduction here determined using the hydrophilic IC with $\alpha = 0$ as the reference ($C_{f,0,\text{hydrophilic}}$), whereas in figure 4.7 the reference is the same IC as used for the $\alpha > 0$ measurement (so either hydrophobic or hydrophilic) with $\alpha = 0$. For all values of α , the use of the coating results in less efficient net DR compared to an uncoated cylinder. Only every second data point is shown to improve readability of the plot. The shaded regions represent the spread in data. The size of the error bars based on the accuracy of the torque sensor is smaller than the marker size. Due to heavy vibrations in the setup resulting from a non-symmetric distribution of air, no data was acquired in the region between $Re = 1.3 \times 10^6$ and $Re = 1.6 \times 10^6$ for $\alpha = 4\%$ and for $Re > 1.4 \times 10^6$ when $\alpha = 6\%$.

erties of the coating remain constant throughout the experiments. This was also confirmed by visual inspection of the coating and by comparison of SEM images of the coating taken before and after the measurements.

The repeated measurements of $\alpha = 2\%$, 4% , and 6% show a spread in skin friction coefficient of 2% , 4% , and 2% , respectively, again suggesting good measurement repeatability. The accuracy of the torque sensor is rated by the manufacturer to be $\pm 0.25\%$ of its maximum rated output. Error regression analysis has shown that the largest error in figures 4.7 and 4.9 is slightly smaller than the markers used. The data spread is reflected by the shaded regions in figures 4.7 and 4.9. These regions are derived by using the minimum and maximum values from the repeated measurements for $C_{f,0}$ and $C_{f,\alpha}$.

4.4.3 Velocity profiles

Shown in figure 4.10 are the results on the velocity profiles from the PIV measurements. Plotted are the azimuthal velocities normalized with the velocity of the inner cylinder u_θ/u_i , versus the normalized position between the inner and outer cylinder $(r - r_i)/d$, for different Reynolds numbers. We compare the *hydrophilic* IC to the *hydrophobic* IC, for a single-phase flow with $\alpha = 0\%$. A clear difference is seen in the region close to the IC, with larger velocities for the hydrophobic IC compared to the hydrophilic IC at similar Reynolds numbers. This is attributed to the larger roughness of the hydrophobic IC compared to the hydrophilic IC. A rougher surface can transport more energy to the flow compared to a smooth surface, resulting in larger velocities in the near wall region for the same driving of the flow (Zhu et al., 2018). In other flow configurations, when the wall is not used to drive the flow, lower velocities will be found for more rough surfaces (Flack and Schultz, 2010; MacDonald et al., 2016; Busse et al., 2017). For larger Re we find larger velocities close to the IC, indicating that the effect of the roughness is stronger with larger Re . This reflects the reduction in thickness of the viscous sublayer, which results in a larger fraction of the small scale roughness (pores) of the hydrophobic IC becoming a relevant length scale to the flow.

Figure 4.10: Plot of azimuthal velocity normalized with inner cylinder velocity u_θ/u_i versus the normalized gap width between inner and outer cylinder $(r - r_i)/d$. The working fluid is without air, so $\alpha = 0\%$. Compared is the rough *hydrophobic* coating with the smooth *hydrophilic* steel inner cylinder. The *hydrophilic* data is provided by Huisman et al. (2014) using the same experimental setup. The inset shows the region close to the inner cylinder.

4.5 Summary and conclusions

The influence of a hydrophobic wall on drag reduction was studied in a highly turbulent Taylor–Couette flow. We applied a hydrophobic coating to the otherwise smooth and hydrophilic inner cylinder (IC) of the Taylor–Couette setup. In single-phase flow, we found a constant increase in drag of about 14% for the rough hydrophobic wall compared to the smooth hydrophilic wall over the whole range of Reynolds numbers $5.0 \times 10^5 \leq Re \leq 1.8 \times 10^6$ measured. For bubbly two-phase flow however, the addition of air bubbles to the flow resulted in more drag reduction for the rough hydrophobic IC as compared to the hydrophilic IC, using the same volume fraction of air bubbles α . For $\alpha \geq 4\%$, more DR was found over nearly the whole range of Re . Only in the region of $Re < 7.0 \times 10^5$ – where the measurement uncertainty is highest – a slight drag increase was found for the hydrophobic IC when $\alpha = 6\%$. A strong difference in DR behaviour is found when comparing $\alpha = 2\%$ with $\alpha \geq 4\%$. The void fraction $\alpha = 2\%$ gives a clear drag increase above $Re = 10^6$, indicating that the bubble drag mechanism is more effective with a superhydrophobic IC when sufficient air bubbles are present in the flow. This can be explained by the micro-scale surface geometry of the surface, acting as roughness to the flow and hence increasing the drag. The effect of the hydrophobic coating is therefore twofold: (i) a more effective bubble drag reduction mechanism, and (ii) an increase in drag from the surface roughness. The role of roughness is confirmed by comparing drag measurements of the hydrophobic IC to drag measurements of the smooth hydrophilic IC with $\alpha = 0$, which shows more drag for all values of α .

The effect of roughness is more pronounced with larger Re , since the thickness of the viscous sublayer is then smaller, making it compatible with the roughness length scales. This is confirmed by velocity profile measurements, showing that the normalized azimuthal flow velocities near the hydrophobic IC are larger compared to the smooth hydrophilic IC for the same Re . This indicates that for larger Re the hydrophobic IC appears rougher for the flow.

Whereas the drag continues to increase with Re for the hydrophobic IC compared to the hydrophilic IC when $\alpha < 4\%$, the difference in drag appears to level off for $\alpha \geq 4\%$, showing a much weaker dependence on Re . Apparently, above a certain Re with $\alpha \geq 4\%$, the difference between the two competing effects reaches a constant value. The result is a constant increase in DR for the hydrophobic IC. The difference in DR does also not vary significantly with α for $\alpha \geq 4\%$. This leads us to the conclusion that, although a minimum amount of air is required for the hydrophobic coating to provide more effective bubble DR, adding more air beyond this minimum

barrier will not necessarily result in more drag reduction. We hope that our work will give guidelines for industrial applications of bubbly drag reduction in hydrophobic wall-bounded turbulence, such as in naval applications or in pipelines. Berghout2019

Conclusions

In this thesis, the influence of certain surface properties on wall-bound flows has been studied. Specifically, surfaces that allow for patches of air being captured between the surface and the liquid above the surface, have been studied in both laminar and turbulent single-phase flow conditions. The influence of surface roughness on drag, and on the drag reducing effect of air bubbles, has been studied in turbulent Taylor–Couette (TC) flow. Also the effect that a hydrophobic surface chemistry has on air bubbly drag reduction, has been studied in turbulent TC flow.

In chapter 1, fluid slip is studied over surfaces in which double re-entrant (micro-) cavities are etched. The unique double re-entrant structure facilitates large, stable gas-liquid interfaces (diameter 150 μm and 200 μm in this research), and hence this results in a long-term stable superhydrophobic surface. These surfaces were used to create a microfluidic channel, in which particle image velocimetry (PIV) was used to locally determine the velocities above the superhydrophobic surface. It was found that the liquid accelerates over the first half of the gas-liquid interface, and decelerates over the second half to match the no-slip boundary condition of solid-liquid interface at the edge. This is reflected by the slip length, reaching its maximum value at the centre of the gas-liquid interface.

The influence of surfactants that adsorb at the gas-liquid interface on the observed slip was investigated using solutions of four different sodium dodecyl sulphate (SDS) concentrations. Surprisingly, no significant difference was found between pure water and the four different solutions. This is in strong contrast to what is suggested in literature, where a dramatic decrease of slip is predicted for our experimental conditions (Peaudecerf et al., 2017). A possible explanation lies in the difference in the mean flow velocities, that is $\mathcal{O}(10^2)$ larger in our experiments compared to the experiments and numerical work of Peaudecerf et al. (2017). This can have an (indirect) influence on the adsorption and desorption kinetics of the SDS molecules at the gas-liquid interface. To test this hypothesis, we suggest an additional

set of experiments in our system at lower flow velocities.

The TC geometry was used in chapters 2–4 to study the drag of wall-bounded turbulent flows. TC flow is the flow between two co-axial cylinders of different radii and equal height. By rotating either of the cylinders, or both, the fluid, typically water, in the annulus between the cylinders is set in motion. The wall shear stress can accurately be determined from measurements of the torque that is required to rotate the cylinders at set rotational frequency. The Twente Turbulent Taylor–Couette (T³C) was used, that can reach Reynolds numbers up to 2×10^6 based upon the width of the gap between the cylinders, for inner cylinder rotation only.

Measurements of torque were in chapter 2 combined with high accuracy single pixel PIV measurements of the velocities of a turbulent flow over a fully rough inner cylinder (IC). The roughness was created by attaching P36 grit sandpaper to the IC. It was shown, that the slope of the fully rough asymptote of the Prandtl–von Kármán log-layer velocity profile is very similar to previous findings in flat plate geometries. Hence, the effects of the stream-wise curvature of the TC appears to be of little influence on these results.

The angular momentum of the smooth outer cylinder (OC) boundary layer, can at its edge be matched to the angular momentum of the IC boundary layer at the edge of that boundary layer, through the assumption of constant angular momentum in the bulk region. This resulted in an equation that relates the wall shear stress (torque) to the equivalent sand grain roughness of the surface. By plugging in the equivalent sand grain roughness from the fully rough asymptote that was determined using the PIV results, it was found that this theoretically predicted torque, matched the experimentally measured torque to within 5% for all of our measurements. This indicates, that from relatively simple torque measurements alone, for any rough surface the equivalent sand grain roughness can be determined. This offers a very promising perspective to the roughness community, that is involved in linking different roughness geometries to flow characteristics.

In chapters 3 and 4 the drag reduction in TC flow that resulted from the addition of air bubbles to the working liquid was measured. In chapter 3, bands of sandpaper were attached to the inner cylinder, resulting in an axially varying pattern of rough and smooth bands. By changing the number and the width of both the rough and smooth bands, different configurations of a partly rough inner cylinder were created of the same roughness coverage.

Results were compared between these different configurations, and with a completely smooth and a completely rough inner cylinder. For all roughness cases it was found that the air bubbly drag reduction was more effective compared to a smooth inner cylinder. This is attributed to a better axial mixing of the air bubbles due to the higher turbulence intensity of the flows over rough walls, but also to turbophoresis. This causes the air bubbles to move to the regions of high turbulent intensity close to the roughness at the inner cylinder. Close to the inner cylinder the air bubbles are most effective to reduce the drag. This is demonstrated by measurements with a smooth rotating IC and a smooth rotating OC in counter direction, which creates strong secondary flow structures that capture the air bubbles in their core, away from the inner cylinder. In this case, negligible drag reduction is found. The high radially outward velocity above the roughness bands, however, triggers the formation of very similar secondary structures in the flow. In this case, we still found drag reduction. This confirmed us that the air bubbles indeed have a strong tendency to move to the regions of high turbulent intensity close to the inner cylinder. As most surfaces used in industrial application that can benefit from air bubbly drag reduction are rough, these results are very relevant.

In chapter 4 it was shown that a hydrophobic inner cylinder of small-scale roughness gives more effective air bubbly drag reduction compared to a smooth hydrophilic inner cylinder. However, a minimum amount of air is required to achieve this. Above a certain maximum amount of air, a further increase of the air volume fraction does not further increase the drag reduction. For (too) low amounts of air bubbles in the working liquid, a strong influence of the small-scale surface roughness is found, leading to less drag reduction compared to the smooth hydrophilic inner cylinder with the same amount of air in the working liquid. The effect of roughness is confirmed by velocity profiles from PIV measurements, showing that the normalized azimuthal flow velocities near the hydrophobic IC are larger compared to the smooth hydrophilic IC for the same Reynolds number. For increasing Reynolds numbers, the thickness of the viscous sublayer decreases, making the effects of roughness more pronounced. For single-phase flow, with the hydrophobic surface in the non-wetted Cassie–Baxter state, also an increase of drag was found. Hence, the non-wetted state is no guarantee for drag reduction.

Outlook

In this thesis, superhydrophobic surfaces were studied in a microfluidic channel under laminar flow conditions, and under turbulent flow conditions in Taylor–Couette (TC) flow. The separation between the typical length scales of both systems is $\mathcal{O}(10^3)$, and $\mathcal{O}(10^4)$ for the typical mean flow velocities. The area of the surface in the TC experiment, is $\mathcal{O}(10^4)$ times the area of the surface in the microfluidic channel. This illustrates one of the big challenges for practical applications of superhydrophobic surfaces. A lot of research is focussed on finding a perfect superhydrophobic surface, which usually results in complex micro-geometries. These are, however, not very well suited to be produced (affordably) in the large quantities required to cover the large surface areas found in practice. Hence, there is a balance between having a well defined microstructure as a repeating unit on the surface (control), and being able to produce large areas of the superhydrophobic surface (producibility).

Large areas of superhydrophobic surfaces that are used in drag reduction research, typically have a random roughness structure, such as the hydrophobic porous membrane used in chapter 4 of this thesis. Another, in literature commonly found, superhydrophobic surface is fabricated using spraycoating. This gives excellent results in terms of producibility, however, it lacks the control of creating well defined structures. As the research in chapter 4 showed, effects of roughness are very dominant for such random superhydrophobic surfaces. Future research should therefore focus on bridging between control and producibility, and develop new production techniques that find a balance between the two.

In chapter 1, no influence of the sodium dodecyl sulphate (SDS) concentration was found on the measured slip over a gas-liquid interface. We hypothesized that this is related to the adsorption and desorption kinetics of the SDS at the interface, being influenced by the relatively large flow velocities, compared to lower velocities used in literature. This would indicate that under these kind of (surfactant) conditions, the slip length is a function of velocity. A first step in future research is therefore to repeat these experi-

ments at lower flow velocities, to find if this is indeed the case. This could be a very important result, as the translation of laboratory results to real-life applications of superhydrophobic surfaces for drag reduction, are commonly expected to be influenced by the contaminations that are present in liquids found in typical applications.

For studies with roughness, chapter 2 showed that the TC geometry can be further exploited to characterise the influence of surface roughness properties on the flow. Direct measurements of torque with a fully rough cylinder surface, can be related to the equivalent sand grain roughness. To confirm this, additional measurements are required, comparing results with the same surface roughness geometry between the TC setup and for instance flat-plate boundary layer setups. Such studies will further strengthen the position of TC in the field of roughness characterization.

The results of air bubbly drag reduction with roughness in TC (chapter 3) can help to translate knowledge of air bubbly drag reduction on smooth walls to industrial and maritime applications where typically roughness is involved. The main finding that air bubble drag reduction is just as effective (or perhaps even more) as it is for smooth walls, is in that sense a positive result. Nonetheless will the roughness always increase the drag compared to a smooth surface.

Regarding the Gas-Drive ship concept introduced in the introduction of this thesis, and linking the results of this thesis to the whole maritime sector, is the use of superhydrophobic surfaces for drag reduction still a promising idea. It is however important to realize, that there are still quite some hurdles to overcome. This field of research is still developing and building its knowledge base through challenging laboratory experiments. Production technologies need to be further refined to create the precisely controlled surfaces in the large quantities that are required.

For contributions on a shorter term, it is perhaps more rewarding to focus efforts on roughness control. It is evident that even the small scale roughness (whether or not the result of bio-fouling) lead to an increase of the drag force. As an active solution in the Gas-Drive concept, is the injection of CO₂ bubbles under the hull a two-folded approach. The gas bubbles reduce the drag, and the CO₂ prevents the growth of bio-fouling, and weakens the organisms already present on the hull, making the removal of those more easy.

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Summary

Many flows in nature and industry are wall-bound. The interaction between the walls and the fluid influences the drag force that the fluid experiences from these walls. We tested the influence of different wall conditions: smooth, rough, and superhydrophobic; on different flow conditions: laminar single-phase flow — with and without surfactants — and turbulent single-phase and two-phase flows.

The influence of surfactants on the observed slip length b was studied in microfluidic channels (chapter 1), of which the bottom wall was made superhydrophobic by etching double re-entrant cavities of diameters 150 μm and 200 μm in the silica. Using micro particle image velocimetry, an average slip length of 4 μm ($b_{\text{max}} = 14 \mu\text{m}$ at the centre of the gas-liquid interface) was measured for the surface with 150 μm cavities using Milli Q water. For the surface with 200 μm cavities, this was 9 μm ($b_{\text{max}} = 28 \mu\text{m}$ at centre of the interface). Four different sodium dodecyl sulphate (SDS) solution were prepared of varying concentrations between $c = 0.08\text{--}1.00 \text{ mmol/l}$. No significant difference in average slip lengths and velocities over the gas-liquid interface were found between the four different solutions and the pure Milli Q water. We attribute this to the relatively large flow velocities in our system, compared to results from literature where an influence of SDS on the slip length was reported in a similar system, but at lower flow velocities. We expect this to be of (indirect) influence on the adsorption and desorption kinetics of the surfactants at the interface, which will be the topic of further studies.

Hydrophobic surfaces with micro-pores were studied for their drag reducing potential in turbulent Taylor–Couette (TC) flow: the flow between two co-axially, independently rotating cylinders (chapter 4). The torque that is required to rotate the cylinders at set frequency, can be measured with relative ease and high accuracy, and is directly related to the wall shear stress and the skin friction coefficient. Experiments were performed with Reynolds numbers ranging between $Re = 0.5 \times 10^6\text{--}1.8 \times 10^6$. It was found that air bubbly drag reduction is more effective with a micro-porous

hydrophobic inner cylinder, compared to a smooth hydrophilic inner cylinder. This observed difference reached a maximum around $Re = 1.1 \times 10^6$. For larger Reynolds numbers, the effects of the surface roughness became more significant, compared to the enhanced bubbly drag reducing effect of the hydrophobic surface. This is the result of a shorter viscous length scale δ_v , which makes the roughness effectively larger. This strong effect of the roughness was also observed in the single-phase measurements, that showed a constant increase of 14% of the skin friction coefficient over the whole range of Reynolds numbers tested, even though the surface was in the non-wetted Cassie–Baxter state.

The TC geometry can potentially be used to determine the equivalent sand grain roughness for any fully rough surface, from torque measurements only (chapter 2). Because of the stream-wise curved nature of the geometry, it is important that this roughness does not extend into the region of the turbulent boundary layer where effects related to this curvature play a role in the production of turbulence. When this condition is met, a very similar value of the *von Kármán* constant κ , that gives the slope κ^{-1} of the fully rough asymptote of the log-layer velocity profile, is found for the TC geometry compared to flat geometries. From the assumption of constant angular momentum in the bulk region of the flow, an equation was derived that relates the wall shear stress (torque) to the equivalent sand grain roughness of the surface. This was shown to be within $\approx 5\%$ agreement with the torque measured during our experiments, for the corresponding equivalent sand grain roughness derived from the velocity profiles obtained from particle image velocimetry.

The influence of wall roughness on bubbly drag reduction was studied in turbulent TC flow for (shear) Reynolds numbers in the range $Re_s = 0.5 \times 10^6 - 1.8 \times 10^6$, by applying bands of grit P36 sandpaper to the inner cylinder. Parts of the inner cylinder were left uncovered, resulting in an axial pattern of rough and smooth bands. The air bubbles in the flow showed a preference for the regions of higher intensity turbulence above the rough parts of the inner cylinder. For all variations of a partly rough inner cylinder, and an inner cylinder completely covered with sandpaper, a larger difference in skin friction coefficient was found when air bubbles were added to the flow, compared to a completely smooth inner cylinder.

Samenvatting

Het merendeel van de (vloeistof- en gas-) stromingen in de natuur of in industriële toepassingen zijn begrensd door wanden. De interactie tussen deze wanden en de stroming is van invloed op de wrijvingsweerstand die de stroming ondervindt. In dit proefschrift is de invloed van verschillende wandcondities: glad, ruw, onmeunig waterafstotend; getest op verschillende stromingscondities: laminair éénfase — met en zonder oppervlakteactieve stoffen — en turbulent één- en tweefase.

De invloed van oppervlakteactieve stoffen op de wandsliplengte b is onderzocht in microfluidische kanalen (hoofdstuk 1), waarvan de bodem waterafstotend is door in het silica geëtste ronde gaten van diameter $150\text{ }\mu\text{m}$ and $200\text{ }\mu\text{m}$, met dubbel inspringende wanden. Met behulp van micro *particle image velocimetry* werd voor het oppervlak bestaande uit de $150\text{ }\mu\text{m}$ gaten een gemiddelde wandsliplengte gemeten van $4\text{ }\mu\text{m}$ ($b_{\text{max}} = 14\text{ }\mu\text{m}$ in het midden van het gas-vloeistof contactoppervlak) voor Milli Q (ultrapuur) water. Voor het oppervlak met $200\text{ }\mu\text{m}$ diameter gaten, was deze waarde $9\text{ }\mu\text{m}$ ($b_{\text{max}} = 28\text{ }\mu\text{m}$). Vier verschillende oplossingen van de oppervlakteactieve stof natriumdodecylsulfaat (NDS) met concentraties binnen het bereik $c = 0.08\text{--}1.00\text{ mmol/l}$ zijn getoetst op hun wandsliplengte. Geen significant verschil in de gemiddelde wandsliplengte en de snelheid over gas-vloeistof contactoppervlak werd gemeten tussen de vier verschillende oplossingen en het Milli Q water. Wij schrijven dit toe aan de hogere stromingssnelheden in onze opstelling, in verhouding tot de snelheden in vergelijkbaar onderzoek uit de literatuur, waar wel een invloed van NDS op de wandsliplengte werd gerapporteerd. Wij verwachten dat dit snelheidsverschil een (indirecte) invloed uitoefent op de adsorptie en desorptie kinetiek van de oppervlakteactieve stoffen aan het gas-vloeistof contactoppervlak. Dit zal het onderwerp zijn van vervolgonderzoek.

Mircoporeuze waterafstotende oppervlakken zijn onderzocht op hun weerstandsreducerende eigenschappen in een turbulente Taylor–Couette (TC) stroming: de stroming tussen twee co-axiaal, onafhankelijk ronddraaiende

cilinders (hoofdstuk 4). De torsie die uitgeoefend moet worden om de cilinders te laten draaien, kan relatief eenvoudig met grote nauwkeurigheid gemeten worden. Deze torsie is direct gerelateerd aan de afschuifspanning aan de wand en de wandfrictiecoëfficiënt. De experimenten zijn uitgevoerd voor Reynolds getallen variërend tussen $Re = 0.5 \times 10^6 - 1.8 \times 10^6$, en wezen uit dat het weerstandsreducerende effect van luchtbellens op de stroming groter is voor de microporeuze waterafstotende binnencilinder, vergeleken met een gladde niet-waterafstotende binnencilinder. Het verschil in het weerstandsreducerende effect was maximaal voor $Re \approx 1.1 \times 10^6$. Voor grotere Reynolds getallen werd het ruwheidseffect van het poreuze oppervlak signifikanter, in verhouding tot het grotere weerstandsreducerende effect van luchtbellens nabij het waterafstotende oppervlak. Dit is te verklaren door een kortere visceuze lengte schaal δ_v , waardoor de ruwheid in verhouding groter wordt. Het ruwheidseffect werd ook teruggevonden in éénfase experimenten, waar een constante toename in de wandfrictiecoëfficiënt van 14% werd gemeten voor alle Reynolds getallen, waarbij het oppervlak gedurende de hele meting in een niet-bevochtigde Cassie–Baxter toestand verkeerde.

Het potentieel van de TC geometrie strekt zich uit tot het domein van ruwheidsmetingen, waarbij de equivalente zandkorrelruwheid bepaald kan worden voor een willekeurig hydrodynamisch volledig ruw oppervlak, door enkel gebruikt te maken van torsie metingen (hoofdstuk 2). Vanwege de in-stromings-richting gekromde geometrie van de opstelling, is het belangrijk dat de ruwheid zich niet uitstrekt tot in het domein in de turbulente grenslaag waar effecten gerelateerd aan deze kromming van invloed zijn op de productie van turbulentie. Als aan deze voorwaarde voldaan is, blijkt dat in de TC geometrie een waarde wordt gevonden voor de *von Kármán* constante κ , die de helling κ^{-1} geeft van de hydrodynamisch ruwe asymptoot van het genormaliseerde snelheidsprofiel in de logaritmische laag van de grenslaag, die vergelijkbaar is met de waarde die gevonden wordt in opstellingen met een vlakke geometrie. Uitgaande van de aanname van een constant impulsmoment in het bulkdomein van de stroming, werd een vergelijking opgesteld die een verband geeft tussen de torsie en de equivalente zandkorrelruwheid. Deze waarde komt binnen een bereik van $\approx 5\%$ overeen met de torsie die gemeten is in onze experimenten, met de overeenkomstige equivalente zandkorrelruwheid bepaald met behulp van de snelheidsprofielen uit *particle image velocimetry*.

De invloed van ruwheid op het weerstandsreducerende effect van luchtbellens is onderzocht in turbulente TC stromingen waarvan de Reynolds getallen varieerden in het bereik $Re_s = 0.5 \times 10^6 - 1.8 \times 10^6$, door het bevestigen van grit 36 schuurpapierbanden aan de binnencilinder. Bepaalde gedeel-

ten van de binnencilinder werden onbedekt gelaten, waardoor er een axiaal alternerend patroon van ruwe en gladde banden ontstond. De luchtbellen lieten zich graag zien in de hoog-turbulente stromingsregionen boven de ruwe banden. Voor alle variaties met een gedeeltelijk ruwe binnencilinder, en een volledig ruwe binnencilinder, werd een groter verschil in wandfrictiecoëfficiënt gemeten na de toevoeging van luchtbellen aan de stroming, vergeleken met een volledig gladde binnencilinder.

*“There is no fun in making music,
it is cold hard labour.”*

Paul Rademaker
UTmost bigband conductor

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I am not sure if you still remember this first meeting, Rob, but after the first one we had many more meetings, in which we discussed many topics. A lot were science related, about our research, but also on academia in general: teaching, funding, organisation, career paths, and so on. Rob his eyes would however would always light up when he got the chance to talk about motorsports, motorcycles, and mountain biking. I am really happy we could share and discuss these, and other not-work-related topics, as well. Not only during our weekly meetings, but also in a bar during APS, at the membrane winterschool in Austria, during coffee breaks, and so on. I also very much enjoyed the sports events we joined, the mountain bike tours, the Batavierenrace and all the skiing at the winterschool. I am glad that I can say that although Rob is the faster runner and mountainbike rider, I can still beat him on ski’s, be it marginally. Thank you Rob, for your guidance and all you have taught me the past four years.

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I have been very lucky to go on some rather nice trips. One of the better ones was the JMBC course in Udine, North-Italy, with Peter Dung, Mazi en Alexander. It was the good life, with many coffee and wine bars, and *gelaterias* on everyone corner of the street. Very good were also the trips to the APS DFD conferences in the US. It was the one in Atlanta where Dennis Bakhuis had found out that our hotel featured a rooftop-jacuzzi. Here we spent a very memorable last night of the conference together with Jelle, Myrthe, Pieter and Vamsi. Also the trip afterwards was very good, spending a week in a big lake house with eleven colleagues. Upon arrival it turned out that the first group to arrive had mixed up houses, and had entered the neighbours’ house. We found out about this while Jelle and Pieter were already very busy testing their Whirlpool, whereas Martin Klein Schaarsberg (dressed in nothing but his ruby red dressing gown) was looting their fridge. I am very happy to have this image stuck in my mind for the rest of my life. The year after, APS brought us to Seattle and the North Cascades, where a good week of holiday was spent with Myrthe, Michiel and Pieter. I have also very good memories of the four(!) membrane winterschools I joined. Every year we would have a different epic ski-team (among which were Rob, Tymen, Dennis Reurink, Rian, Renaud, Aura, Slawomir, Jurjen, Shuyana, Moritz, Nicole and Wouter), and had lots of fun racing the perfect *groomers*, plowing through knee-deep snow, and playing chicken on the black slopes. Just as good were the late and long nights in the *Kellerbar*, with all the people from Twente and Aachen together.

Also outside of work I always managed to enjoy myself. In the beginning of my PhD years I still lived in my old student house, where I had a lot of

fun. But after half a year or so, I decided it might be better to move out, which was however not an easy decision. Luckily, a lot of old roommates are now dear friends, and we regularly meet. I was also always very lucky to play a lot of music. For instance with the UTmost bigband, led by Paul Rademaker, where I enjoyed some truly great years. Paul really learned me how good music is supposed to be played. However, nothing made me more happy than all the shows we played with Ciska de Rat. Many times I thought: “It won’t get any better than this”, and then we would top it at the next gig. The whole atmosphere around and after the rehearsals, the shows, and the trips we made was always epic. Thanks a lot. Enzo, Stefan, Fabian, Robin, Amber, Maurits, Floor, Thijs, Reinier, and Koen: you have no idea how much I will be going to miss you when I leave Enschede.

Together with Iris I have shared a lot of happy memories and moments over the past years. I know the both of us didn’t want it to end this way, however, please know that I am happy and grateful for all you have given me and what we have shared.

I am very lucky to share the day of my defence with my two paranymphs at my side, Enzo and Fabian, two of my best friends. Thanks a lot guys, the past four years would not have been the same without you.

Finally I would like to thank my family. My grandma, who I always like to visit and who always makes me laugh. My brother, Jan-Willem, my role model. If you wouldn’t have pursued a PhD, I for sure wouldn’t have started mine either. So thank you for that. My parents, this thesis starts and ends with you, and would have never existed without you. Thank you for all your help, support and trust in me.

*“Muziek maken is niet leuk,
het is keihard werken.”*

Paul Rademaker
dirigent UTmost bigband

Dankwoord

De afgelopen vier jaar zijn echt voorbij gevlogen. Het voelt haast alsof het vorige week was, dat ik licht overdonderd het kantoor van Detlef uitliep, omdat ik een promotieplek aangeboden had gekregen, zonder te beseffen dat ik erop gesolliciteerd had. Niet veel later (althans, in mijn herinnering) was ik in het kantoor van Rob, aangezien het een gedeeld project betrof tussen Detlef en Rob. Ik zei tegen Rob: “Ik weet het nog niet zo goed Rob, ik moet er nog even over nadenken.” Rob zei: “Nou Pim, volgens mij ga jij het gewoon doen.” Ik ben deze promotie inderdaad gaan doen, en heb er geen moment spijt van gehad. Terwijl ik dit dankwoord schrijf, ontvouwt de tijd die de afgelopen vier jaar zo snel voorbij is gegaan zich langzaam.

Ik weet niet zo goed of jij je deze eerste bespreking nog kunt herinneren, Rob, maar na deze eerste volgenden er nog vele andere, waarin we veel onderwerpen hebben besproken. Veel hadden te maken met ons onderzoek, of de academische wereld: onderwijs, *funding*, de organisatie ervan, carrières, enzovoort. Gelukkig konden we ook andere onderwerpen bespreken, en er kwam altijd een twinkeling in Rob zijn ogen als het over motorsport, motorfietsen, of mountainbiken kon gaan. Ik ben blij dat dit kon. Niet alleen tijdens ons wekelijkse overleg, maar ook tijdens de membraan winterschool, in een bar tijdens conferenties, tijdens koffiepauzes, enzovoort. Ook genoot ik erg van de sportevenementen waar we aan mee hebben gedaan: de mountainbike tours, de Batavierenrace en het skiën tijdens de winterschool. Alhoewel je overduidelijk de snellere loper en fietser bent, ben ik blij dat ik je op de ski's nog achter me kan laten, al scheelt het niet veel. Bedankt Rob, voor al je begeleiding en alles wat je me de afgelopen jaren hebt geleerd.

Beste Detlef, bedankt voor je vertrouwen en de kans die je me hebt gegeven te werken in de Physics of Fluids (POF) groep. Ik heb veel van

je geleerd over het schrijven van papers, het doen van onderzoek en het bedrijven van wetenschap in de breedste zin. Het zijn dingen die ik in de rest van mijn carrière nog toe zal passen. Ik bewonder je streven naar perfectie, en *quality control*, maar ook je competitiviteit, in de breedste zin. Eén keer heb ik de fout gemaakt om tussen neus en lippen door op te merken dat Rob sneller had gelopen met de Batavierenrace dan je zelf had, wat resulteerde in een betoog dat Rob zeker tien jaar jonger is, en dat dit daarom niet meer dan logisch was. Op jouw beurt kon je het dan ook niet laten om bij me in te wrijven dat Utkarsh van me gewonnen had met tennis, en je teleurstelling daarover te laten blijken. Ter mijn verdediging: Utkarsh had gewoon een hele goede dag. Bedankt Detlef, voor je begeleiding, en alles wat ik van je heb mogen leren.

Beste Sander, ik heb altijd het gevoel gehad dat met jouw terugkeer naar POF, en de Taylor–Couette en waterkanaal laboratoria, alle projecten een boost hebben gekregen. Niet alleen de dingen waar ik zelf mee bezig was, maar ook de projecten van de anderen. Als dagelijkse begeleider liet je weinig te wensen over. Je wist alles van de opstellingen en meetmethoden die we gebruikten. Daar bovenop was je inventief, slim, vol humor, altijd bereid om te helpen, en soms simpelweg de motivatie die ik nodig had. Het hebben van een lade vol snoepgoed helpt natuurlijk ook. Eigenlijk zou iedere PhD student een Sander moeten hebben, maar helaas zijn er niet zoveel van gemaakt, dus ik bof dat ik er eentje had. Bedankt dat je altijd voor me klaar stond.

Chao, het was altijd fantastisch om je te zien, met je schier eindeloze enthousiasme, vrolijkheid en motivatie. Ik koester warme herinneringen aan onze groepsoverleggen via Skype. Met name als jij inbelde vanuit huis, met je kinderen spelend op de achtergrond, en vanuit eenzelfde situatie Ruben ook, met z'n kleine dochtertje op schoot. Ik vond het ook altijd mooi als je ons bezocht in Twente, en de dineetjes die we hadden met de 'experimental turbulence' groep. Bedankt voor al je support over de jaren, en ik ben erg blij dat je één van de commissieleden bent tijdens mijn verdediging.

Beste Jeff, in de afgelopen jaren ben je een goede vriend geworden en stond je altijd klaar met raad, advies of een ongezouten mening. Over alles wat we bespraken wist je altijd wel iets zinnigs te zeggen. Hoewel het vaak over ons onderzoek ging, hadden de wijste lessen die je me geleerd hebt daar meestal niks mee te maken.

Dit onderzoek was niks geworden zonder de fantastische technische ondersteuning die in beide vakgroepen aanwezig was. Bij POF in de vorm van het illustere trio Bos, Bruggert, en Benschop (Martin, Gert-Wim, en Bas)

— voor al uw vakwerk, adviezen en praat tussendoor. Regelmatig kwam ik even langs bij jullie op kantoor, en hoewel het meestal over werk ging, kwam ik toch ook wel vaak voor adviezen, gereedschap, of een praatje over klussen in en rond het huis, het wagenpark, reparaties aan m'n ski's, en van alles en nog wat. Dank, voor al jullie hulp. Ook jij bedankt, Dennis van Gils, voor al je adviezen als “één der oervaders” van de T³C. Natuurlijk ook jij bedankt Joanita. Op onovertroffen wijze neem jij de organisatie van heel veel zaken binnen de POF groep op je. Memorabel was de ruim vijf minuten durende staande ovatie die je kreeg tijdens ‘POF 20’ conferentie van alle aanwezige (oud) collega's.

Mijn terugkeer bij de Soft matter, Fluidics and Interfaces groep (SFI), was een warm weerzien met Ineke, Sander Haase, Lidy, Damon, Vic, Rob, en veel andere bekende gezichten van het membraan cluster. Ineke, je bent echt mijn heldin geweest op het lab, bedankt daarvoor. Bij alles wat ik niet zo goed wist (en als werktuigbouwer op een chemisch lab was dat eigenlijk best veel) kon jij me helpen, uitleggen, en laten zien hoe het moest. Bij SFI leerde ik ook Jan van Nieuwkastele kennen. Eén van de eerste dingen die hij tegen mij zei (met een licht Rotterdams accent) was: “Weet je wat het is, ik heb gewoon zoveel zoi, ik gebruik dit bureau ook want anders kan ik het allemaal niet kwijt”. Toen wist ik meteen, deze man en ik, wij begrijpen elkaar. Bedankt Jan, voor al je hulp en adviezen rondom de μ -PIV, voor alle chipjes die je voor me gemaakt hebt, de avonturen in het nano-lab (ik vond het in ieder geval een avontuur), en dat je mee bent gegaan naar Aachen. Bedankt ook Lidy, voor het uit handen nemen van heel veel zorgen binnen de SFI groep en je altijd oprechte betrokkenheid. Brigitte, bedankt voor alle hartelijkheid en gezelligheid die je toevoegt aan onze groep.

Als ik terugdenk aan de laatste vier jaar, heb ik veel mensen leren kennen, goede vrienden gemaakt en veel gelachen. Het is onmogelijk om iedereen te noemen zonder in een saaie opsomming van namen terecht te komen. Maar, bij deze mijn dank aan iedereen, met wie ik ooit gelachen of iets gedronken heb (waarschijnlijk koffie of bier). Weet dat ik tijdens het schrijven van dit dankwoord aan je heb gedacht, en dat zonder jou de afgelopen vier jaar echt heel veel saaiër zouden zijn geweest.

Gelukkig heb ik aan deze vier jaar veel mooie herinneringen overgehouden, waarvan ik er hier een aantal wil uitlichten. Veel hiervan vonden plaats in het Taylor–Couette lab, waar ik veel heb mogen leren, lachen, en samenwerken met Ruben (veel gezocht naar gereedschap ook), Dennis Bakhuis, Rodrigo, en Yoan. Dit lab vormde ook deels het decor van de *Taco Wars* film, wat een geweldig project was om deel van uit te maken, en

wat me de kans bood om samen te werken met talenten als Carola, Martin Klein Schaarsberg, Álvaro Moreno Soto, and Misha. Ook hebben we goede sportzomers gehad, waarin er veel wielrennen werd gekeken op kantoor met Ruben en Mike, de BIOS vakgroep werd vernederd bij het beachvolleybal toernooi met Pieter, Myrthe, Misha, en Mazi, en later nog een keer met de Batavierenrace. Tijdens deze sportzomers heb ik sowieso veel gespeeld op het zand, in veel verschillende samenstellingen, bij deze allemaal bedankt.

Bij SFI werkte ik in mijn eerste jaren veel op het lab samen met Hani aan microfluidische chips. Ik denk dat wij goed in balans waren wat betreft Hani's secuurheid en gevoel voor priegelwerk, en mijn eigen neiging om soms wat grotere stappen te willen maken. Ook bracht ik toen veel tijd op het lab door samen met Maaïke, die ik mocht begeleiden tijdens haar afstudeeronderzoek. Ik denk dat dit voor ons beiden erg leerzaam was, voor mij niet in de laatste plaats omdat ik feitelijk zelf pas net een paar maanden eerder was afgestudeerd. Met de hulp van Jeff hebben we er ons bijzonder goed doorheen geslagen, en ik kijk er dan ook met een tevreden gevoel op terug.

Ook bij SFI heb ik altijd geboft met mijn kantoorgenoten, met helemaal in het begin Dona (mijn PhD *twin-sister*), Arputha, en Abimbola, totdat het me daar (vrij letterlijk) te heet onder de voeten werd. Daarna mocht ik intrekken in het *coole* kantoor bij Khalid, Kristianne, en Özlem. Met het Membraan cluster werd er bijna iedere vrijdagmiddag geborreld. Het mag geen verrassing zijn dat ik hier toch ook wel graag bij mocht aanschuiven, met in de harde kern onder anderen Josh, Harmen, Antoine, Youri, Alberto, Nikos, Renaud, Ömer en Dennis Reurink. Mooi was ook de studietour naar Denemarken. Het hardlopen door Kopenhagen met Bob en Anne richting de zeemeermin, en de kroegentocht van Antoine.

Door de jaren heen heb ik ook het geluk gehad wat reisjes te mogen maken. Eén van de mooiste was toch wel de JMBC cursus in Udine, Noord Italië, samen met Pieter Dung, Mazi en Alexander. Op iedere straathoek was wel een koffietentje, *gelateria*, of wijnbar te vinden. Ook altijd goed waren de tripjes naar de APS DFD conferenties in de VS. In Atlanta had Dennis Bakhuis al vrij gauw de *rooftop-jacuzzi* van het hotel ontdekt en hebben we de laatste avond daar vrij memorabel in doorgebracht samen met Jelle, Myrthe, Pieter, en Vamsi. Ook het tripje na afloop van de conferentie was onvergetelijk: met elf man een week in een groot *lake house*. Bij aankomst bleek dat de eerste groep het verkeerde *lake house* was binnen gegaan (dat van de burens), en daar kwamen we achter terwijl Jelle en Pieter het bubbelbad al aan het uittesten waren, en Martin Klein Schaarsberg (slechts gekleed in zijn robijnrode ochtendjas) bezig was de koelkast te plunderen. Wat ben ik blij dat dit beeld op mijn netvlies staat gebrand. Het jaar erop bracht

APS ons naar Seattle en de North Cascades, waar Myrthe, Michiel, Pieter en ik nog een goede week vakantie hebben gevierd. Goede herinneringen heb ik ook aan de vier(!) jaar op rij dat ik mee mocht naar de membraan winterschool in Oostenrijk. Ieder jaar hadden we weer een ander mooi ski-team bij elkaar (met onder anderen Rob, Tymen, Dennis Reurink, Rian, Renaud, Aura, Slawomir, Jurjen, Shuyana, Moritz, Nicole en Wouter), waarmee niets leuker was dan racen over het strak geprepareerde corduroy, en gezamenlijk ploeteren door kniediepe sneeuw of op iets te zwarte pistes. Minstens zo mooi waren de late avonden in de *Kellerbar*, met alle andere Twente- en Aachenaren.

Ook buiten werktijd heb ik me altijd prima kunnen vermaken. Eerst woonde ik nog op mijn oude studentenflat, waar ik het erg gezellig had. Na een half jaar besloot ik dat het toch maar beter was ergens anders te gaan wonen. Dit viel me niet mee, omdat de Sarijopen niet iets is wat je zo 1, 2, 3 achter je laat. Dat blijkt gelukkig ook wel uit de (kerst)diners, feestjes en tripjes die nog altijd gepland worden met oud-huisgenoten, die inmiddels goede vrienden zijn geworden. Ook heb ik altijd met veel plezier in allerlei bandjes gespeeld. Het waren gouden tijden met de UTmost bigband, waar ik dankzij de leiding van Paul Rademaker écht muziek heb leren maken. Maar er waren eigenlijk maar weinig dingen waar ik zo gelukkig van werd als de optredens met Ciska de Rat. Iedere keer dacht ik weer “Beter dan dit wordt het niet”, en dan gingen we er een volgende keer toch weer overheen. Ook de sfeer tijdens en na repetities, en rondom de optredens en de tripjes die we gemaakt hebben was echt altijd te gek. Daarom, bedankt. Enzo, Stefan, Fabian, Robin, Amber, Maurits, Floor, Thijs, Reinier, en Koen: ik ga jullie gruwelijk missen als ik straks Enschede ga verlaten.

Met Iris heb ik de afgelopen jaren ook veel mooie dingen en momenten mogen beleven. Ik weet dat we allebei niet hadden gewild dat het zo zou eindigen, maar weet ook dat ik je dankbaar bent voor alles wat je me gegeven hebt en wat we hebben mogen delen.

Ik ben erg blij dat ik deze dag mag meemaken samen met mijn paranimfen, Enzo en Fabian, twee van mijn beste vrienden. Bedankt jongens, de afgelopen vier jaar zouden niet hetzelfde zijn geweest zonder jullie erbij.

Ten slotte wil ik ook graag mijn familie bedanken. Mijn oma, waar ik altijd graag op bezoek mag gaan en om wie ik altijd vreselijk moet lachen. Mijn broer, Jan-Willem, mijn grote voorbeeld. Als jij niet was gaan promoveren, dan was ik er zelf nooit aan begonnen. Dus bedankt daarvoor. En mijn ouders. Deze thesis begint met jullie en eindigt met jullie, en had er zonder jullie nooit gelegen. Bedankt voor al jullie hulp, steun en vertrouwen.

Bio

Pim Adriaan Bullee was born on March the 2nd, 1989 in Gorinchem, the Netherlands. After having completed his basic education, he moved to Enschede and enrolled in the bachelor's programme Advanced Technology at the University of Twente. For his bachelor's graduation project, Pim worked on superhydrophobic surfaces in the Soft matter, Fluidics and Interfaces group of professor Rob Lammertink. His interest in fluid dynamics made him choose to enrol in the master's programme of Mechanical Engineering, taking courses with the Engineering Fluid Dynamics group of professor Harry Hoeijmakers and professor Kees Venner. In this group he also wrote his master's thesis, on the topic of ice accretion in air craft engines. As a PhD candidate, Pim joined the GasDrive project, a collaboration between the Universities of Twente, Wageningen and Delft, in which he worked on superhydrophobic surfaces for drag reduction in turbulent flows. This research was a collaboration between the Soft matter, Fluidics and Interfaces group of professor Rob Lammertink, and the Physics of Fluids group of professor Detlef Lohse. His research interests include turbulent flows, single and two-phase flows, roughness in turbulence, and microfluidics.

Scientific output

Publications

- **2020:** *Catastrophic Phase Inversion in Turbulent Taylor–Couette Flow*
D. Bakhuis, R. Ezeta, **P.A. Bullee**, A. Marin, D. Lohse, C. Sun, and S.G. Huisman
Phys. Rev. Lett. (to be submitted)
- **2020:** *Characterizing the drag properties of rough surfaces with a Taylor–Couette setup*
P. Berghout, **P.A. Bullee**, T. Fuchs, S. Scharnowski, C.J. Kähler, D. Chung, D. Lohse, and S.G. Huisman
J. Fluid Mech. (to be submitted)
- **2020:** *Effect of axially varying sandpaper roughness on bubbly drag reduction in Taylor–Couette turbulence*
P.A. Bullee, D. Bakhuis, R. Ezeta, S.G. Huisman, C. Sun, R.G.H. Lammertink, and D. Lohse
Int. J. Multiphase Flow (submitted)
- **2020:** *Bubbly drag reduction using a hydrophobic inner cylinder in Taylor–Couette turbulence*
P.A. Bullee, R.A. Verschoof, D. Bakhuis, S.G. Huisman, C. Sun, R.G.H. Lammertink, and D. Lohse
J. Fluid Mech. **883**, A61
- **2020:** *Controlling secondary flow in Taylor–Couette turbulence through spanwise-varying roughness*
D. Bakhuis, R. Ezeta, P. Berghout, **P.A. Bullee**, N.C. Tai, D. Chung, R. Verzicco, D. Lohse, S.G. Huisman, and C. Sun
J. Fluid Mech. **883**, A15

- **2018:** *The influence of wall roughness on bubble drag reduction in Taylor–Couette turbulence*
R.A. Verschoof, D. Bakhuis, **P.A. Bullee**, S.G. Huisman, C. Sun, and D. Lohse
J. Fluid Mech. **851**, 436–446
- **2018:** *Air cavities at the inner cylinder of turbulent Taylor–Couette flow*
R.A. Verschoof, D. Bakhuis, **P.A. Bullee**, S.G. Huisman, C. Sun, and D. Lohse
Int. J. Multiphase Flow **105**, 264–273

Selected talks

- **2020: Membrane Winterschool (Hirschegg, AUT)**
Pitch on research with double re-entrant cavities
Membrane Winterschool pitch prize winner
- **2019: APS Division Fluid Dynamics (Seattle, USA)**
Effect of axially varying sandpaper roughness on bubbly drag reduction in Taylor–Couette turbulence (poster)
- **2019: Soft Matter+ Day (Enschede, NL)**
Bridging between measurements of slip at the gas-liquid interface in microfluidic systems and air bubble drag reduction at the large turbulent scale
Invited talk
- **2019: JMBC Burgers Symposium (Lunteren, NL)**
Effects of roughness on air bubble drag reduction in turbulent flows
- **2018: International Couette–Taylor Workshop (Marseille, FR)**
Turbulent drag reduction using superhydrophobic surfaces and air bubble injection
- **2017: Membrane Winterschool (Hirschegg, AUT)**
Slip of shear thinning liquids in a microfluidic bubble mattress