PIV Measurements of Turbulent Flow in a Rectangular Channel over Superhydrophobic Surfaces with Riblets

Richard Mark Perkins
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PIV Measurements of Turbulent Flow in a Rectangular Channel

over Superhydrophobic Surfaces with Riblets

Richard M. Perkins

A thesis submitted to the faculty of
Brigham Young University
in partial fulfillment of the requirements for the degree of

Master of Science

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R. Daniel Maynes
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ABSTRACT

PIV Measurements of Turbulent Flow in a Rectangular Channel over Superhydrophobic Surfaces with Riblets

Richard M. Perkins
Department of Mechanical Engineering, BYU
Master of Science

In this thesis I investigate characteristics of turbulent flow in a channel where one of the walls has riblets, superhydrophobic microribs, or a hybrid surface with traditional riblets built on a superhydrophobic microrib surface. PIV measurements are used to find the velocity profile, the turbulent statistics, and shear stress profile in the rectangular channel with one wall having a structured test surface. Both riblets and superhydrophobic surfaces can each provide a reduction in the wall shear stress in a turbulent channel flow. Characterizing the features of the flow using particle image velocimetry (PIV) is the focus of this research. Superhydrophobicity results from the combination of a hydrophobic coating applied to a surface with microrib structures, resulting in a very low surface energy, such that the fluid does not penetrate in between the structures. The micro-rib structures are aligned in the streamwise flow direction. The riblets are larger than the micro-rib structure by an order of magnitude and protrude into the flow. All the test surfaces were produced on silicon wafers using photolithographic techniques. Pressure in the channel is maintained below the Laplace pressure for all testing, creating sustainable air pockets between the microribs. Velocity profiles, turbulent statistics, shear stress profiles, and friction factors are presented. Measurements were acquired for Reynolds numbers ranging from $4.5 \times 10^3$ to $2.0 \times 10^4$. Modest drag reductions were observed for the riblet surfaces. Substantial drag increase occurred over the superhydrophobic surfaces. The hybrid surfaces showed the greatest drag reduction. Turbulence production was strongly reduced during riblet and hybrid tests.

Keywords: turbulent, channel, experiment, riblet, superhydrophobic, skin friction, shear stress, drag reduction, particle image velocimetry, PIV, laser, velocity profile, fully-developed, photolithography, micromachining, microribs, etching, photoresist, flow visualization, wetting, plastron, turbulence production, law of the wall
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<th>Definition</th>
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<tr>
<td>A</td>
<td>Channel cross-sectional area</td>
</tr>
<tr>
<td>C</td>
<td>Image size calibration (m/px)</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Characteristic particle frequency (rad/s)</td>
</tr>
<tr>
<td>$D_h$</td>
<td>Hydraulic diameter</td>
</tr>
<tr>
<td>DR</td>
<td>Drag reduction</td>
</tr>
<tr>
<td>$F_c$</td>
<td>Cavity fraction</td>
</tr>
<tr>
<td>$FS$</td>
<td>Flow shift (vol. flow rate shifted to opposite half of channel)</td>
</tr>
<tr>
<td>H</td>
<td>Channel height</td>
</tr>
<tr>
<td>M</td>
<td>Image magnification</td>
</tr>
<tr>
<td>P</td>
<td>Turbulence production</td>
</tr>
<tr>
<td>$P_w$</td>
<td>Channel wetted perimeter</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$Re_{\tau}$</td>
<td>Turbulent Reynolds number</td>
</tr>
<tr>
<td>SD</td>
<td>Surface difference (between test and smooth surface)</td>
</tr>
<tr>
<td>St</td>
<td>Stokes number</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
</tr>
<tr>
<td>$T_{tot}$</td>
<td>Normalized total shear stress</td>
</tr>
<tr>
<td>$T_{turb}$</td>
<td>Normalized turbulent/Reynolds shear stress</td>
</tr>
<tr>
<td>$U$</td>
<td>Normalized streamwise velocity</td>
</tr>
<tr>
<td>$U_{max}$</td>
<td>Maximum time-averaged velocity of the flow</td>
</tr>
<tr>
<td>$U_{rms}$</td>
<td>Normalized turbulence intensity/RMS velocity in streamwise direction</td>
</tr>
<tr>
<td>$V_{rms}$</td>
<td>Normalized turbulence intensity/RMS velocity in wall-normal direction</td>
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<tr>
<td>$W_m$</td>
<td>Relative module width $p/D_h$</td>
</tr>
<tr>
<td>Y</td>
<td>Normalized wall-normal coordinate</td>
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<tr>
<td>$d_I$</td>
<td>Image diameter ($\approx$ window size)</td>
</tr>
<tr>
<td>$f$</td>
<td>Darcy friction factor</td>
</tr>
<tr>
<td>$f_{char}$</td>
<td>Particle characteristic frequency (Hz)</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational acceleration</td>
</tr>
<tr>
<td>h</td>
<td>Height</td>
</tr>
<tr>
<td>k</td>
<td>Turbulent kinetic energy</td>
</tr>
<tr>
<td>$p$</td>
<td>Static pressure</td>
</tr>
<tr>
<td>$p_h$</td>
<td>Protrusion height</td>
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<tr>
<td>$p_{air}$</td>
<td>Pressure of air in cavities</td>
</tr>
<tr>
<td>$p_{water}$</td>
<td>Pressure of water over cavities</td>
</tr>
<tr>
<td>$s$</td>
<td>Spacing/pitch/module width</td>
</tr>
<tr>
<td>t</td>
<td>Thickness</td>
</tr>
<tr>
<td>u</td>
<td>Streamwise component of flow velocity</td>
</tr>
<tr>
<td>$\bar{u}_{avg}$</td>
<td>Average streamwise velocity</td>
</tr>
<tr>
<td>$u_s$</td>
<td>Slip velocity</td>
</tr>
<tr>
<td>$u_{a,s}$</td>
<td>Apparent streamwise slip velocity</td>
</tr>
<tr>
<td>$u_{part}$</td>
<td>Seeding particle velocity</td>
</tr>
<tr>
<td>$u_{wall}$</td>
<td>Streamwise velocity at the wall</td>
</tr>
<tr>
<td>$u_{\tau}$</td>
<td>Friction velocity (turbulent velocity)</td>
</tr>
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</table>
$v$ Transverse component of flow velocity

$x$ Streamwise coordinate

$y$ Wall-normal coordinate

$\overline{u'u'}$ Reynolds normal stress: variance of $u'$ fluctuations

$\overline{v'v'}$ Reynolds normal stress: variance of $v'$ fluctuations

$\overline{u'v'}$ Reynolds shear stress: covariance of the $u'$ and $v'$ fluctuations

$\Delta t$ Change in time

$\Delta T$ Change in temperature

$\Delta p_{\text{Laplace}}$ Laplace pressure

$\Delta x$ Particle displacement

$\beta$ Bias error (systematic error)

$\gamma$ Liquid surface tension

$\delta x$ Change in displacement across half window

$\epsilon$ Dissipation of turbulent kinetic energy

$\theta_{A}$ Advancing contact angle

$\theta_{CA}$ Contact angle

$\lambda$ Slip length

$\mu$ Dynamic viscosity

$\nu$ Kinematic viscosity

$\pi$ Pi (mathematical number 3.1415...)

$\rho$ Density of water

$\sigma$ Standard deviation (67% confidence)

$\sigma_x$ Particle displacement precision error

$\tau_{\text{visc}}$ Viscous shear stress

$\tau_{\text{tot}}$ Total shear stress

$\tau_{\text{turb}}$ Turbulent shear stress

$\tau_w$ Wall shear stress

$\tau_{xy}$ Shear stress stress in the $x$-$y$ plane

$\omega_c$ Turbulent velocity fluctuation frequency (rad/s)

$\chi_{\text{wall}}$ Wall location error

$\chi_{\text{fit}}$ Linear shear stress fit error

$\chi_{\text{therm}}$ Thermocouple error

**Subscripts, superscripts, and other indicators**

$[ ]_i$ value of a property for a given indice value

$[ ]_p$ indicates $[ ]$ is a seeding particle property

$[ ]_x$ indicates $[ ]$ in the streamwise direction

$[ ]_y$ indicates $[ ]$ in the wall-normal direction

$[ ]_z$ indicates $[ ]$ in the spanwise direction

$[ ]^+$ indicates $[ ]$ is in wall units

$[ ]^*$ indicates the quantity $[ ]$ is averaged across the lower half of the channel

$[ ]'$ indicates $[ ]$ is a fluctuation from the mean

$[ ]$ indicates an averaged property

$\chi[ ]$ Total error (95% confidence)

$\%[ ]$ Percentage of $[ ]$ relative to nominal value
CHAPTER 1. INTRODUCTION

1.1 Motivation

This thesis details the work performed investigating turbulent flow over several different microstructured surfaces. The surfaces tested included superhydrophobic surfaces with breaker ridges, riblet surfaces, and combined riblet-superhydrophobic surfaces. The ultimate goal of the research is to quantify the friction reduction when there is flow across these surfaces, which are designed to passively reduce friction drag. Skin friction on surfaces has always resulted in either increased energy cost or lower performance for the system. In some applications there are significant design needs or cost benefits related to drag reduction; such as submerged water vehicles, underground drilling, and high-efficiency power systems.

A large number of turbulent drag reduction techniques have been employed and studied. These include the use of acoustic waves, surfactants, polymer additives, microbubbles, air injection, compliant surfaces, active blowing/suction, turbulence control, electromagnetic turbulence control, and plasma actuators [1–7]. There are inherent advantages and disadvantages for each method and there isn’t a solution that is appropriate for all scenarios. As manufacturing and engineering capabilities continue to develop, many of these techniques may become increasingly relevant and practical for engineering applications.

This work focuses on the drag reducing capabilities of riblets and superhydrophobic surfaces, both of which are passive drag-reducing mechanisms. There has been considerable work to characterize these surfaces, but little work with both mechanisms combined. In this work particle image velocimetry (PIV) was used to investigate the influence of these two mechanisms both independently and conjointly on the skin friction in a turbulent flow.
1.2 Drag Reduction and Turbulent Flow

When a liquid flows along a flat surface it will encounter some resistance. This resistive force is called surface drag, which results from momentum transfer between the flowing liquid and the wall. This same process of momentum transfer will cause a pressure drop in a pipe, since the flow velocity must be constant due to conservation of mass. When the flow is laminar, momentum transfer occurs through viscous stresses in the fluid. In the case of one-dimensional Newtonian flow, such as in a two-dimensional fully-developed channel, the shear stress is directly proportional to the velocity gradient and scaled by the fluid’s viscosity [8]. This relationship is presented in Eqn. 1.1

\[ \tau_{xy} = \mu \frac{du}{dy} \]  

(1.1)

where \( x \) is the streamwise coordinate and \( y \) is the wall normal coordinate, \( \tau_{xy} \) is the shear stress on a fluid element in the \( x-y \) plane, \( u \) is the velocity in the streamwise direction, and \( \mu \) is the kinematic viscosity. The Navier-Stokes equation for momentum in the \( x \) direction shows that for laminar flow the shear stress will be linear across the channel and the velocity profile will be parabolic.

The Reynolds number, the ratio of the inertial forces to viscous forces, is defined as \( Re = \rho \bar{u}_{avg}D/\mu \), where, \( \rho \) is the fluid density, \( \bar{u}_{avg} \) is the time-averaged and spatially averaged velocity of the flow, and \( D \) is the pipe diameter. Transition to turbulent flow occurs at higher Reynolds numbers, when the inertial forces of a flow exceed the viscous damping forces and the flow near the wall becomes unstable, resulting in a cascade of turbulent vortices. As a result the walls of turbulent flows are subject to much higher viscous stresses and greater energy losses due to the greater momentum transfer and increased viscous dissipation that results from the turbulent eddies. Years of turbulence research have shown that the turbulent motion of the flow is initiated by the formation of near-wall vortical structures and related ejection/sweep events [9, 10]. By altering the surface geometry and flow conditions one can influence these basic turbulent structures and alter the friction between the flow and the wall [10, 11].

For turbulent flows it is most common to work in terms of wall units which are non-dimensional parameters that scale with the turbulent structures of the flow. The friction velocity
Figure 1.1: Analytical velocity profiles, $u^+$, for the viscous sublayer (dashed line) and log-law region (solid line) are plotted against the wall normal distance, $y^+$, on a semilog plot. Example velocity control data from this experiment for $Re = 1.8 \times 10^4$ are also shown on the plot as open circles. The various turbulent regions are labeled at the bottom.

is defined as $u_\tau = \sqrt{\tau_w/\rho}$, where $\tau_w$ is the shear stress at the wall. The velocity and wall-normal distances can be normalized by $u_\tau$, as shown in Eqns. 1.2 and 1.3,

$$u^+ = \frac{u}{u_\tau}$$  \hspace{1cm} (1.2)$$

$$y^+ = \frac{y u_\tau}{\nu}$$  \hspace{1cm} (1.3)$$

where $\nu$ is the kinematic viscosity of the fluid. The average turbulent velocity profile has a few distinct regions. These different turbulent regions are illustrated in Fig. 1.1, which shows an average turbulent velocity profile plotted in wall units. The velocity profile a of smooth wall rectangular
channel at Re = 1.8x10^4 was acquired using PIV as part of this research. Nearest the wall is a thin laminar region called the viscous sublayer. In this region the flow is diffusion dominated. For a smooth surface, the velocity in the viscous sublayer in wall units, u⁺, is equal to the wall-normal distance in wall units, y⁺ (shown with a dashed line). Outside the viscous sublayer, in the region 8 < y⁺ < 30 there is a buffer layer where both viscous dissipation and turbulent mixing are important. Further from the wall this transitions into the log-law layer where turbulent mixing is the dominant mode of momentum transport. In this region the resulting profile follows the log-law shown in Eqn. 1.4 [12],

\[ u^+ = 2.44 \ln y^+ + C^+ \]  \hspace{1cm} (1.4)

where C⁺ is a constant derived from the location of the edge of the viscous sublayer. The value is typically taken to be 5.0 [12] or 6.0 [13]. Numerous test runs in this experiment suggest 6.5 to be a more appropriate value for this particular channel geometry and subsequent two-dimensional analysis along the channel centerline. Beyond the log-law region is the outer layer, which is near the channel center where the log-law no longer holds. For turbulent flow it is useful to define a turbulent Reynolds number, Reᵣ, based on the friction velocity, shown in Eqn. 1.5

\[ Reᵣ = \frac{u_τ δ}{ν} = \frac{\bar{u}_{avg} \sqrt{\frac{f D_h}{8 ν}}}{ν} = Re \sqrt{\frac{f}{128}} \]  \hspace{1cm} (1.5)

where δ is the channel half height, defined as δ = H/2, ν is the kinematic viscosity, and f is the friction factor which will be defined shortly. Reᵣ is useful for comparing between turbulent flows and plays an important role in plots with wall units, as will be seen in the results section.

Because turbulent flow is chaotic, the velocity profile at any instant is changing, so statistical methods need to be used. The instantaneous velocity at a point, u, can be broken into two parts, as in Eqn. 1.6

\[ u = \bar{u} + u' \]  \hspace{1cm} (1.6)

where \bar{u} is the time-averaged velocity and u' is the velocity fluctuation. The same holds for the wall-normal and transverse velocities, v and w, respectively. Although the instantaneous turbulent
fluctuations and flow structure are very difficult to predict and quantify, the 2D Reynolds-averaged Navier Stokes equation can be used to quantify the average turbulent flow behavior over time. In these equations there are additional velocity fluctuation terms in the total stress tensor called the Reynolds stresses, which are not present in the laminar case. The turbulent pressure terms, $\rho u'u'$, $\rho v'v'$, and $\rho w'w'$, are velocity variances multiplied by $\rho$ and represent the turbulent velocity fluctuation contribution to the pressure. The turbulent shear stress terms, $\rho u'v'$, $\rho u'w'$, and $\rho v'w'$, are velocity cross covariance terms multiplied by $\rho$ and account for the turbulent transport of momentum. The 2D, incompressible, steady-state Reynolds-averaged Navier Stokes equation for $x$ momentum is shown in Eqn. 1.7 [14],

$$\rho \frac{\partial \bar{u}}{\partial x} + \rho \frac{\partial \bar{v}}{\partial y} = -\frac{\partial \bar{p}}{\partial x} + \mu \left( \frac{\partial^2 \bar{u}}{\partial x^2} + \frac{\partial^2 \bar{u}}{\partial y^2} \right) - \rho \left( \frac{\partial u'v'}{\partial x} + \frac{\partial u'w'}{\partial y} \right)$$

(1.7)

where $\bar{p}$ is the average static pressure at a point in the flow. In the case of fully-developed steady mean flow (no velocity gradients in $x$ direction) in an infinitely wide channel of fixed wall-to-wall spacing ($\bar{y} = 0$), Equation 1.7 is greatly simplified, becoming Eqn. 1.8

$$\frac{\partial \bar{p}}{\partial x} = \mu \frac{\partial^2 \bar{u}}{\partial y^2} - \rho \frac{\partial u'v'}{\partial y}$$

(1.8)

Because of the nature of fully-developed flow, the time-averaged pressure gradient in $x$ must be constant in both $x$ and $y$, even though there exists a changing pressure gradient in $y$ due to the Reynolds pressure $\rho \bar{v}' \bar{v}'$ [14]. By integrating this expression Eqn. 1.8 with respect to $y$, across the channel, one obtains Eqn. 1.9.

$$(d\bar{p}/dx)_y + C = \mu \frac{d\bar{u}}{dy} - \rho \bar{u}' \bar{v}'$$

(1.9)

where $C$ is a constant of integration. If Eqn. 1.9 is further integrated over the remaining $x$ and $z$ coordinates, one sees that the pressure force along the channel is equal to the shear force acting on the walls. The first term on right side of Eqn. 1.9 is the viscous shear stress, $\tau_{visc} = \mu \frac{d\bar{u}}{dy}$, which was defined in Eqn. 1.1. The second term is a turbulent or Reynolds shear stress, $\tau_{turb} = -\rho \bar{u}' \bar{v}'$, deriving from the turbulent fluctuations and mixing of the flow. The sum of these shear stresses is total shear stress, $\tau_{tot}$, shown in Eqn. 1.10 as
\[ \tau_{\text{tot}} = \tau_{\text{visc}} + \tau_{\text{turb}}. \]  

With this definition, Eqn. 1.9 may be rewritten in terms of the total shear stress, shown in Eqn. 1.11.

\[ \frac{dP}{dx} y + C = \tau_{\text{tot}} \]  

From Eqn. 1.10 it is clear that \( \tau_{\text{tot}} \) must vary linearly in \( y \) (across the channel) and scale with the pressure gradient. This linear profile can be characterized using PIV to measure both the average velocity gradient, \( d\pi/dy \), and the cross covariance of \( u \) and \( v \), \( u'v' \). The terms can then be combined as in Eqn. 1.10 to give the total shear stress distribution across the channel.

Figure 1.2 shows a typical plot of the the viscous, turbulent, and total shear stress as a function of wall distance, \( y^+ \), at \( Re = 7600 \). The shear stress in wall units is defined as \( \tau^+ = \tau/\tau_w \). The total shear stress (open squares) is linear from \( \tau_w \) at the wall \( (y^+ = 0) \) to zero in the channel center \( (y^+ = 120) \). The zero stress location in wall units is \( Re_{\tau} \), so in this case \( Re_{\tau} = u_{\tau} \delta / v = 120 \). In the figure the viscous stress (open circles) and Reynolds stress (shaded triangles) contributions can be clearly seen. The viscous sublayer is the region dominated by viscous stress nearest the wall, the buffer region has comparable values of both the viscous and Reynolds stress, and the log-law region, the largest portion of the channel, is dominated by the Reynolds stress. Once the \( \tau_{\text{tot}} \) profile across the channel is known, \( \tau_{\text{tot}} \) can be extrapolated linearly to the wall to find the wall shear stress, \( \tau_w \) [15].

A friction factor is a non-dimensional measure of momentum transfer used to quantify the friction drag for a given \( Re \) and surface type. The Darcy-Weisbach friction factor, used in duct and channel flow, is a ratio of the wall shear stress to the dynamic pressure of a flow, defined in Eqn. 1.12

\[ f = \frac{8 \tau_w}{\rho \bar{u}_{\text{avg}}^2}, \]  

where \( \tau_w \) is the wall shear stress and \( \bar{u}_{\text{avg}} \) is the averaged wall-to-wall flow velocity. The friction factors for pipe flow over both smooth and rough surfaces can be found in the familiar Moody diagram and is fit by the recursive Colebrook equation [16]. For a rectangular channel in place of the pipe diameter, the hydraulic diameter is used, \( D_h = 4A/P_w \), where \( A \) is the channel cross-
Figure 1.2: Normalized shear stress, $\tau^+$, as a function of $y^+$ showing the total stress contributions of the viscous stress and the turbulent shear stress at $Re = 7600$

sectional area and $P_w$ is the wetted perimeter. In this paper the correlation by Beavers et al. [17] is also used, which specifically addresses low $Re$ turbulent flow in large aspect ratio rectangular ducts, shown in Eqn. 1.13 [17].

$$f = 0.5072Re^{-0.3}$$ \hspace{1cm} (1.13)

Since the concepts of drag are all based on these fundamental principles, skin friction reduction techniques focus on three fundamental concepts: (1) lowering the effective viscosity, $\mu$, (2) reducing the effective wall velocity gradient, $du/dy$, which can be done by creating a slip boundary at the wall, $u_{wall} = u_s \neq 0$, and (3) reducing the momentum transfer through turbulent vortex structures, $\rho u'v'$ [10].
1.3 Superhydrophobic Surfaces

Superhydrophobic surfaces are a special class of surfaces that have very low surface energy, and by extension have very low attraction to water. There are two necessary conditions for a surface to be superhydrophobic. The superficial chemistry must be hydrophobic, meaning a low surface energy material or coating, and there must be some form of micro/nano-structure on the surface [18, 19]. This allows water to be suspended on top of the surface topography, thus limiting the effective contact area between the liquid and the surface [20]. The resulting superhydrophobic surfaces have a very high contact angle, $\theta_{CA}$ (typically greater than $150^\circ$), low contact angle hysteresis, and a small rolling angle [19, 21–24].

1.3.1 Superhydrophobic Surfaces in Nature

Nature provides inspiration for superhydrophobic research. The canonical example is leaves from the lotus plant, which have microscale papillae structures coated in nanowax crystals. These naturally occurring superhydrophobic surfaces allow water to slide off, removing contaminant particles and allowing the plant to emerge clean from muddy water [25]. These surfaces take advantage of the reduced water-surface contact area which limits the adhesion of water to the surface [25, 26]. As a result of the reduced adhesion, water droplets are able to bounce or roll off the surface. Similar “lotus-effect” examples are found on rice leaves with micropapillae covered in wax nanobumps and on butterfly wings with shingle-like structures [26]. At the present time a number of coatings are available commercially to render a variety of surfaces self-cleaning due to the lotus-effect, even as researchers continues to develop more robust and effective surfaces.

Certain types of aquatic insects, such as the great diving beetle *Dytiscus marginalis*, are able to survive for extended periods in water, owing to water-repellent hairs that form a stable air layer [27, 28]. This layer is called a plastron and enables the insects to sustain air respiration underwater, due to the large air-water surface area which increases oxygen transfer to the plastron. Rather than relying on pressure for structures as a bubble does, a plastron relies on solid hydrophobic microstructures to suspend the water above the plastron. Plastron formation on artificial superhydrophobic surfaces has been considered for underwater breathing apparati [27, 28]. The formation of a plastron layer amidst posts or between microribs allows a shear-free wall bound-
ary to be created at the tops of the microstructures, which has great potential for drag reduction applications.

Other research takes advantage of defining the water attachment locations on a surface. One study, inspired by droplets on zoysia leaves in the wind, looked at using oscillating sessile droplets, pinned on hydrophilic regions and kept in place with superhydrophobic boundaries, to generate uniform illumination [29].

1.3.2 Superhydrophobic Surface Fundamentals

Figure 1.3 shows droplet interaction and $\theta_{CA}$ for a few different surface types. In Fig. 1.3 (A) a water droplet suspended by surface tension on top of microfeatures in the Cassie-Baxter state (cavities are unwetted). This state prevails when the pressure difference between the fluid and gas in the cavities is sufficiently small [21–23, 30]. The critical pressure for wetting is called the Laplace pressure, which is the maximum pressure that can be sustained by the meniscus without overcoming surface tension. The Laplace pressure for a given geometry is shown in Eqn. 1.14

$$\Delta p_{Laplace} = p_{water} - p_{air} = \gamma \left( \frac{1}{R_1} + \frac{1}{R_2} \right)$$

(1.14)

where $p_{water}$ is the water pressure, $p_{air}$ is air pressure in the cavity, $\gamma$ is the liquid surface tension, and $R_1$ and $R_2$ are the surface radii of curvature. This pressure depends on the geometry and spacing of a surface due to their influence on the radii of curvature. For microribs which have a single radius of curvature ($R_2 \to \infty$), the Laplace pressure simplifies to Eqn.1.15 [31]

$$\Delta p_{Laplace} = p_{water} - p_{air} = -\frac{2\gamma \cos \theta_A}{s - t}$$

(1.15)

where $\theta_A$ is the relevant advancing contact angle, $s$ is the microrib pitch/spacing, and $t$ is the thickness of the microrib. When the Laplace pressure of a fluid-gas interface is exceeded, the fluid will be forced down around the microstructures, causing transition to the Wenzel state, a fully wetted surface that has a lower contact angle [20], as seen in Fig. 1.3 (B). Figure 1.3(C) and (D) show a droplet on hydrophobic and hydrophilic surfaces, respectively. A non-structured hydrophobic surface is generally limited to a contact angle of $120^\circ > \theta_{CA} > 90^\circ$ and a hydrophilic
surface will have $\theta_{CA} < 90^\circ$. If the surface is sufficiently hydrophilic, water will wick along the material, leaving only a thin film.

It is important to realize that geometry and chemistry alone are not sufficient to specify fluid interaction with the surface. Pressures, impact velocities, vibrations, contamination effects, and two-phase thermodynamic interactions can affect the degree of wetting that can occur on a superhydrophobic surface. These other interactions can be significant and must be considered when working with superhydrophobic surfaces.

1.3.3 Superhydrophobic Surface Geometry

A variety of superhydrophobic structures have been developed. These micro- and nano-features can either be randomly distributed, follow substrate crystal planes, or be regularly constructed [13, 15, 21, 24, 32]. Figure 1.4 shows SEM images of three surface types used in our lab. Figure 1.4 (A) shows a microribbed surface and (B) shows a post surface, both of which are regularly patterned using photolithographic and etching techniques. Figure 1.4 (C) is the commercial coating Rust-Oleum®NeverWet®. This type of coating has micro- and nano-scale particles sprayed onto the surface with an aerosol nozzle.

Production and design of surfaces with randomly scattered roughness are more economical and commercially lucrative at present; however, surfaces with regular features, such as microribs or pillars/posts, are useful in research because exact features can be recreated and compared and an idealized geometry for a given purpose can be designed. For example, widely-spaced, regulur
microrib structures are ideal for turbulent drag reduction, but closely spaced random structures are ideal for standard non-wetting surfaces and self-cleaning applications.

Several studies have shown that surfaces with both micro- and nano-scale roughness have advantages. This type of surface has the benefits of different roughness scales, yielding the drag reducing properties of micro scale superhydrophobic surfaces and the extremely low surface energy and wetting resistance of nanoscale superhydrophobic surfaces. Other advantages include reducing contact angle hysteresis [19, 24], rendering the surfaces superhydrophobic at the microscale, and reducing the energy requirements necessary to reverse wetting [33, 34].

1.3.4 Superhydrophobic Heat Transfer

In a channel, superhydrophobic surfaces can reduce heat transfer at a liquid-surface interface, especially when the substrate material is a good conductor. This happens because of reduced contact between the liquid and the wall, due to insulating air pockets, and greater microfeature spacing, resulting in a lower Nusselt number. This effect becomes more pronounced in microchannels [35, 36]. Smaller channels with superhydrophobic surfaces must therefore be used judiciously in cases where heat transfer is important. Droplet evaporation and heat transfer can also be reduced on heated superhydrophobic surfaces, a result of decreased contact area between the suspended droplet and the surface [37]; however, when the fluid fills the cavities and wets, eliminating the insulative air layer, surface heat transfer is restored.
Superhydrophobic surfaces have been investigated for increasing the effectiveness of a condenser. If the condensing droplets can be removed quickly, much greater heat transfer can occur [38], although there are some difficulties that still need to be resolved. Some studies have observed that droplets forming on the surface can form in between the microstructures [20,38], and wet rather than being removed from the surface. This creates a film that will cause a lower overall heat transfer coefficient to the bulk fluid, even less than that of a smooth hydrophilic surface [38]. However, under the right conditions droplets successfully coalesce out from between the cavities and quickly roll off the surface, allowing increased condensation heat transfer to occur [39].

Superhydrophobic coatings have been considered for water droplet removal in anti-icing applications, such as airplanes. Conceptually, reduced contact time, reduced heat transfer, and quicker droplet removal will prevent ice formation [40, 41]. From a practical standpoint superhydrophobic surfaces may not be ideal for de-icing applications, as they are rendered ineffective in humid conditions and can be easily damaged structurally by the ice [40, 41].

1.3.5 Laminar Flow over Superhydrophobic Surfaces

There can be significant reduction in wall friction when a laminar flow moves over a patterned superhydrophobic surface that is maintained in the Cassie state. This occurs because of slip,
the presence of non-zero velocity at a wall. Fig. 1.5 (A) illustrates the difference in local boundary conditions on a superhydrophobic wall. Depicted is a ribbed superhydrophobic surface with flow along the tops of the microribs. As result of cavity regions at the wall, the flow is locally not confined to a no-slip condition, instead a free-shear boundary prevails. A slip velocity can occur along the air-water interface as shear stress is applied [23,42]. This becomes especially significant in microscale applications. The combination of solid regions (no-slip) and cavities (slip), creates an averaged slip velocity along the wall [36]. Fig. 1.5 (B) shows the aggregate velocity profile for pressure-driven channel flow over a surface with slip. The slip over a given surface is generally characterized by the apparent slip length, a property that is independent of the wall velocity gradient. The apparent slip length is the wall-normal distance from the surface to the virtual origin of the velocity profile, found by linearly extrapolating the velocity profile past the wall. The slip velocity, \( u_s \), is related to slip length, \( \lambda \), as shown in Eqn. 1.17

\[
     u_s = \lambda \left. \frac{du}{dy} \right|_{\text{wall}(y=0)}
\]  

(1.16)

The slip length naturally increases as the cavity region increases relative to the solid surface area. It is useful to quantify the cavity region in terms of a cavity fraction, \( F_c \), which is the ratio of the relative cavity surface area to the total projected area of fluid on the surface. For a microrib surface \( F_c \) is defined \( F_c = (s - t)/s \), where \( s \) is the pitch/spacing of a the microrib pattern and \( t \) is the microrib thickness. The slip length increases substantially as the cavity fraction approaches unity (fluid is sitting exclusively on an air layer) [42–47], at which point the feature height and air viscosity become important [48]. For longitudinal microribs there is an analytical solution for slip as a function of \( F_c \) and \( s \), shown in Eqn. 1.17 [43,47,48].

\[
     \lambda = \frac{s}{\pi} \ln \left( \sec \left( F_c \frac{\pi}{2} \right) \right)
\]  

(1.17)

The analytical solutions shows that the slip length scales linearly with the pitch and increases dramatically for very small \( F_c \) [31,42,43,46,48,49].

Additionally for a given surface geometry, a channel with a smaller \( D_h \) will have a significantly higher relative reduction in frictional pressure drop [44–46,50]. This parameter is usually discussed in terms of the relative module width, \( W_m = s/H \), which is the ratio of microfeature pitch...
to the channel height. For large $W_m$ (small channel), a mid-value cavity fraction can yield a significant drag reduction, but for small $W_m$ (large channel), a very high cavity fraction is necessary to achieve any significant drag reduction [45,46,50]. This is because large slip lengths are necessary to reduce the velocity gradients at the wall in flows with large bulk flow rates. Ybert et al. [48] show that for both posts and ribs surfaces with a very large effective contact angle, $\theta_{CA} > 178^\circ$, very large slip lengths are possible, much larger than the feature spacing, $s$. Additional results show that in the laminar regime, the percent pressure drop reduction is independent of the flow rate, for a given cavity fraction [44,46].

Superhydrophobic surface geometry is very important when trying to achieve drag reduction. Longitudinal microribs consistently outperform post surfaces for a given cavity fraction [31], except at very large $F_c$ and $\theta_{CA}$ [31,42,48]. The drag reduction for longitudinally aligned microribs is greater than for transverse microribs, and for small channels the slip length of the longitudinal microribs is twice that of the transverse microribs [43]. It has been shown that for laminar flow over surfaces with a post configuration, lower drag is achieved with a regular array of posts aligned with the flow, as opposed to a staggered post array [50]. The increased drag for the staggered arrangement is likely due to the absence of wide continuos slip interface, being disrupted periodically by the staggered post arrangement.

Superhydrophobic surfaces have a significant effect on flow separation as well. In an experiment using longitudinal and transverse microribs on a cylinder, the longitudinal ribs caused the separation point to move forward and the vortex shedding frequency to increase, while transverse ribs moved the separation point rearward and decreased vortex shedding frequency [18].

One consideration for flow over superhydrophobic surfaces is the effect of meniscus deformation on slip length. Most studies indicate that there will be a reduction in slip length resulting from meniscus deformation [31,47,48,51]. Recently, a more in-depth treatment of meniscus deformation used a numerical computation of the Poisson equation and Laplace equation with appropriate meniscus curvature and boundary conditions to find the effective slip length [52]. They found that for longitudinal ribs and small $W_m$ (large channels), the slip length increases with meniscus protrusion angle, $\theta$ for all $F_c$ and decreases as the meniscus recedes into the cavities. They also found that longitudinally aligned ribs have larger slip than transverse ribs, especially for large $\theta$. For small $W_m$ both Couette and Pouseuille flows behave the same. However, for smaller channels
than considered in this thesis, meniscus deformation has different behavior resulting in a reduction of slip length with $\theta$, in part due to menisci blocking the flow. In small channels there is also an increase in slip length when the meniscus recedes into the cavity.

Recent research [28] has looked to quantify how the thickness of this plastron layer plays a role in slip and drag reduction. This work suggests there is an ideal thickness for which 20-30% drag reduction can be achieved for spheres in laminar flows when the gas is allowed to circulate optimally.

One of the key limitations for the use of superhydrophobic surfaces in practical applications is a narrow operating pressure range. A study by Xu et al. [53] explored the cavity wetting phenomenon by using an air-filled microcavity with transparent sidewalls. They observed that there is a hydrostatic pressure threshold for continuously stable Cassie-Baxter state. Once this pressure is exceeded pressure-driven gas diffusion leads to gradual failure of the plastron and a drag increase [34, 53]. In an effort to overcome the prohibitive pressure range necessary to sustain drag-reducing superhydrophobic surfaces, Lee and Kim [34] investigated and discussed several regeneration/de-wetting methods such as diffusion from water saturated with gas, pressured gas injection, chemical gas generation, thermal gas generation, and electrolysis. Surface regeneration shows some promise, but for practical applications most methods to restore the Cassie-Baxter state are difficult and problematic.

1.4 Breaker Ridges on Superhydrophobic Surfaces

In flow situations, superhydrophobic surfaces have a tendency for a cavity to completely flood a cavity region once a single location has wetted. In order to combat universal flooding, sparsely spaced transverse microridges have sometimes been added to localize the flooding and effectively divide the surface into smaller cavities/cells, which flood less easily. These micro barriers have been called breaker ridges [18, 54] and corrals [55].

It is not fully understood how these features affect the drag on superhydrophobic surfaces in the flow; however, Woolford [55] reported a small decrease in the drag reduction on superhydrophobic surfaces with breaker ridges in turbulent flow.

During preliminary testing and wetting observations done preparatory to the results presented in this thesis, wetting of superhydrophobic surfaces proved to be a significant issue. The
addition of breaker ridges to the superhydrophobic surfaces caused a substantial improvement and resistance to wetting. As a result, breaker ridges were added to the superhydrophobic surfaces in the present work in an effort to ensure the surfaces remained in the Cassie-Baxter state.

1.5 Superhydrophobic Surfaces in Turbulent Flow

There have been a number of numerical studies on superhydrophobic surfaces in turbulent flow. In a study by Min and Kim [56], the effects of slip in turbulent flow were evaluated using direct numerical simulation (DNS). Their results showed that streamwise slip reduces turbulence, spanwise slip increases turbulence, and the combined effect is one of reduced turbulence, leading to an overall reduction in friction. They found that the drag reduction is a strong function of the wall slip length, \( \lambda^+ (\lambda^+ = \lambda u_\tau / \nu) \), which increases with increasing \( Re \), and that the effective slip length must be on the same order as the viscous sublayer thickness. They reported a maximum drag reduction of 17% for the highest slip length tested, where \( \lambda^+ = 3.566 \). Fukagata et al. [57] used DNS as well and confirmed this result. Additionally they created a theoretical prediction for the drag reduction as a function of slip length. They showed that significant drag reduction (30%-90%) is possible when the slip length exceeds 10 wall units. Busse and Sandham [58] further built upon this work by mapping the friction change as a function of streamwise slip, \( \lambda_x \), and spanwise slip, \( \lambda_c \) using DNS. For large \( \lambda_x \) drag reduction beyond 80% is possible and for large \( \lambda_c \) drag increase of 60% is possible. Martell et al. studied rib and post geometries using DNS [59]. In keeping with previous studies, they found larger microfeature gaps lead to increased slip, higher cavity fraction surfaces provide greater drag reduction, and inadvertently showed that ribs out-perform posts for the same cavity fraction. They showed a max drag reduction of 37% for a \( F_c = 0.93 \) post surface.

An expansive study by Jeffs et al. [60], used a k-\( \omega \) CFD model to study the effects of the cavity fraction, \( F_c \), Reynolds number, \( Re \), and relative module width, \( W_m \), on the friction factor, \( f \). For all turbulent flows, \( f \) decreased substantially as \( W_m \) increased, making the slip regions closer in scale to the channel size. Once \( W_m \) was sufficiently small, increased \( F_c \) created larger slip regions and significant reduction in \( f \) (up to 90%). It was shown that the drag reduction for all parameters can be collapsed onto a single curve that is a function of \( U_{a,s} \), the apparent slip velocity normalized by the channel average velocity, shown in Eqn. 1.18,
where \( \bar{u}_s \) is the average apparent slip velocity. This means that the same friction reduction will be present for a slip velocity relative to the average velocity. Subsequently, to achieve a given drag reduction, there must be a large enough slip length (due to \( F_c, W_m, Re \)) to generate the target slip velocity. This work matched well with the results of the DNS studies mentioned previously.

A recent study by Park et al. [61] used DNS studies to further the understanding of superhydrophobic surfaces in the absence of spanwise slip. They related slip length to rib geometry and drag reduction, clarified the role of superhydrophobic surfaces in turbulent vortex damping, and showed that drag reduction of over 90% is possible for turbulent flow in the absence of spanwise slip over a flat shear-free interface with \( F_c = 0.93 \) and \( W_m = 0.375 \).

It is clear, upon examination and comparison with numerical works, that many of the experimental superhydrophobic surfaces that have been tested have had poor geometry for generating slip. Less optimal conditions include poor interface geometry, low cavity fraction, low relative module width, or large spanwise slip. As a result of very mixed results from experiments, there has been considerable debate about the actual drag reduction provided by superhydrophobic surfaces in turbulent flow; however, there is some movement towards a consensus.

One of the first papers on turbulent flow over a superhydrophobic surface was presented at the AIAA 3rd Annual Flow Conference in 2006 by Henoch et al. [32]. This paper showed some modest drag reduction just after turbulent transition for boundary layer flow in a water tunnel with a nanograss (post) surface. Some early turbulent experiments were performed by Zhao et al. [62], where a turbulent force on a plate was measured in a rotary flume and in a water tunnel. Few details are given on the geometry of the roughened silane-coated surface, but the results showed no drag reduction in turbulent flow and a modest reduction in laminar flow, suggesting the surfaces tested had small slip lengths and/or non-ideal geometry.

These initial studies were followed by experimental papers on superhydrophobic drag reduction in turbulent flow by Daniello et al., Woolford et al., and Peguero and Breuer [13, 15, 63]. Daniello et al. [63] used both PIV and pressure drop measurements to measure the drag reduction of their PDMS surfaces, which are characterized by \( F_c = 0.5 \) and \( W_m = 0.01, 0.02 \). For the single
surface PIV measurement a maximum drag reduction of 21% was reported using results from a high speed camera. The pressure drop measurements show drag reduction of 50% for two surfaces, more than had been anticipated by numerical studies [60]. Woolford et al. [13] performed a similar experiment with both PIV and pressure measurements over wafers with silicon microribs, which had $F_c=0.8$ and $W_m = 0.01$. The results differed from those of Daniello et al. with a modest 11% reduction in the friction factor for longitudinal ribs. Results for transverse microribs show a modest 6.5% increase in the friction factor. Peguero and Breuer [15] investigated turbulent flow over three surface types: silane-coated aluminum plate with 20 µm round grooves etched, a hydrophobic coated sand blasted aluminum plate with 15 µm RMS roughness, and nanograss surface. Their results showed no definitive evidence of drag reduction.

In an experiment by Aljallis et al. [64], superhydrophobic flat plates were pulled through a towing tank. The two superhydrophobic test surfaces had nanoscale topography. The surface with the highest $\theta_{CA}$ showed 30% drag reduction at low, mostly laminar speeds. In the turbulent regime no drag reduction was observed. A surface with a lower $\theta_{CA}$ was also tested, and a substantial drag increase was measured. Depletion of air bubble and wetting was observed and may be a factor.

Two recent studies again highlight the disparity between superhydrophobic surface data. In his dissertation and a conference paper, Prince [54, 65] measured the pressure drop in a channel with superhydrophobic walls, $F_c=0.8$ and $W_m = 0.01$, that had breaker ridges. The largest drag reduction was 7%. The testing also simultaneously measured control surface data in the same channel and had results for riblets that compare closely with reported riblet results, which will be discussed further in the next section. The channel was opaque so the condition of air in the surfaces during testing is not known. The other recent works were by Park et al. [66, 67], where a built-in shear sensor was used to directly measure the drag on the wall in turbulent boundary layer flow where $Re_{\tau} \approx 250$. The surfaces had 50 and 100 µm pitch and $F_c$ that ranged from 0.3 to 0.95. At lower $F_c$, the surfaces with larger pitch showed larger drag reduction than surfaces with a lower pitch. Interestingly, for very high $F_c$, the drag reduction became independent of $s$, reaching a maximum drag reduction of 75% for $F_c = 0.98$. Noting the direct measurement results by Park et al., which agreed well with analytical predictions, it is possible that the breaker ridges used by Prince severely limited drag reduction.
The consensus is that superhydrophobic surface do reduce drag, even if the disparity in the results hasn’t been fully resolved at present. There can be significant variability between experimental setups. Beyond measurement and data processing variability, varied surface performance can result from breaker ridges, other surface geometry, turbulent flow characteristics, hydrostatic pressure, plastron thickness [28], air layer compliance [68], development length, diffusion time, coating durability and contamination effects. Many of these parameters need to be more fully-resolved and understood in order to move forward.

On a related note, Kwon et al. [69] recently took advantage of the increase in slip with convex bubbles embedded in channel walls to achieve a 10% drag reduction for turbulent flow in a 2.84 cm diameter pipe. Their turbulent results seem to run counter to the reduced slip predictions by Davis and Lauga [51]. Further work in this area may be warranted.

1.6 Riblets

The other passive drag reduction mechanism considered is riblets. Riblets are an application of the extensive turbulent research done prior to the 1980’s, and have been considered for use on pipe walls, aerodynamic bodies [70], and fan blades [71]. Studies have shown that a great majority of turbulence production occurs near the wall, where elongated, counter-rotating, streamwise vortices have been observed [72, 73].

Riblets are small streamwise-oriented ribs which act as barriers to the spanwise turbulent motions. They are sized based on the size of the turbulent structures near the wall. Figure 1.6 shows several of the various geometries have been considered for riblets including (A) blade, (B) sawtooth, (C) trapezoidal, and (D) scalloped riblets [11, 74, 75]. Drag reduction measurements for riblets show the ideal riblet shape is an infinitely thin blade type feature and the ideal size (in wall units) is a height of $h^+ = h u_\tau / \nu \approx 8$ and a spacing of $s^+ = s u_\tau / \nu \approx 16$ [11, 74].

DNS studies show that correctly sized riblets are able to dampen spanwise interactions of the streamwise surface vortices [76]. This has been confirmed with three-dimensional particle tracking velocimetry (PTV) measurements which show the riblets interact with and dampen the conversion of streamwise kinetic energy into the spanwise direction, causing a reduction in the turbulent velocity fluctuations and Reynolds shear stress near the wall [77]. Typical measurements for optimally sized riblets yield a reduction in drag of 5-10% [11, 71, 74].
An large-eddy simulation (LES) study of riblets suggests a further 50% increase in drag reduction may be possible by the addition of a sinusoidal oscillation of the riblets in the spanwise direction as the flow progresses downstream [75].

1.7 Combined Riblet and Superhydrophobic Surface

Some exploratory research has been accomplished in our lab in fabricating microrib superhydrophobic surfaces with riblets incorporated on the surface [54]. This compound surface type seeks to take advantage of both the riblet and superhydrophobic surface and examine the potential complementary effects of the surfaces. It is theorized that the superhydrophobic surface would reduce the effective surface area of riblets and that riblets would reduce the spanwise turbulence generated by the superhydrophobic surface. Initial pressure drop measurements show modest drag reduction up to 7% [54].

Barbier et al. [78] recently published a study on superhydrophobic riblets in Couette flow. They cut concentric grooves in an aluminum surface, forming riblets with heights of 10, 100, and 1000 µm and then etched the surface in a series of steps with citric acid and tetramethylammonium hydroxide, rendering the entire surface with submicron pores and spikes. The textured surface was coated in Hyflon AD60, creating superhydrophobic riblets. The samples were tested with a rheometer. In the laminar regime the surfaces showed drag reduction that increased with rib spacing, indicating the presence of superhydrophobic behavior on the surface. There is some evidence of delayed transition to turbulence. The surface with 100 µm grooves, corresponding to $s^+ = 8$ showed the best performance, with a maximum drag reduction of 20%, significantly more than riblets provide on their own.

In the research presented in this thesis, PIV was used to gather data and visual access to observe the wetting behavior of a superhydrophobic microrib surface with superimposed riblet
structures. This thesis presents the velocity profiles, shear stress, turbulence production, and friction factors obtained using PIV techniques for riblets and superhydrophobics surfaces individually and for the hybrid surface.

1.8 Division and Topics of the Remaining Chapters

The rest of the paper will proceed as follows. Chapter 2 will discuss the surface fabrication, experimental setup, and data processing methods. Chapter 3 will present and discuss the results obtained during testing. Lastly, the important conclusions are presented in Chapter 4, as well as ideas for future work. Supplementary information and materials are included in the appendices.
CHAPTER 2. EXPERIMENTAL SETUP

2.1 Test Surface Fabrication

For this investigation, four surface types were fabricated: smooth control surfaces, micro-ribbed superhydrophobic surfaces, riblet surfaces, and combined riblet/superhydrophobic microrib hybrid surfaces. All surfaces were fabricated in the BYU Cleanroom – Integrated Microfabrication Lab on nominally 500 µm thick silicon wafers. Common photolithography and etching techniques were used to create the desired feature geometry. After preparation the wafers were diced to fit inside the test channel. The following section describes the processes used to create the surfaces. For complete details and processing instructions, see Appendix A.

2.1.1 Superhydrophobic Surface

For superhydrophobic surface fabrication a 2 µm layer of AZ nLOF 2020 negative photoresist is first spun onto silicon wafers. The wafers are then placed on a hotplate for a softbake to evaporate the solvent, leaving the solidified photoresist. The photoresist is then exposed to UV light through the microrib mask in a Karl Suss MA150 Aligner. The pattern that is exposed on the wafer, shown in Fig. 2.1, consists of 8 µm microribs oriented in the longitudinal direction with a 40 µm pitch and 8 µm breaker ridges oriented transverse to the flow at intervals of 2.5 mm. Additionally there is a nominally 80 µm gap at the beginning of the wafer before the first breaker ridge occurs.

Following exposure, the wafer is again placed on the hot plate to bake allowing crosslinking to occur. The wafer is then immersed in AZ300 MIF developer for 60 seconds where the uncured photoresist is stripped away, leaving the exposed photoresist to act as a mask for a DRIE (deep reactive ion-etch) Bosch-type process. The wafer is loaded into the STS Multiplex ICP Etch system where multiple plasma cycles etch 16 µm into the silicon, leaving vertical walls beneath
the photoresist. The remaining photoresist is stripped off in Nano-Strip®, leaving a bare silicon surface with silicon microribs that are nominally 8 µm wide, 16 µm deep, and repeated with a 40 µm pitch. These structures are shown in Fig. 2.2.

To prepare the surface for its superhydrophobic coating, a 200 µm layer of chromium is deposited on the surface using thermal evaporation. After the deposition of chromium, a solution of 0.2% Teflon®AF, dissolved in an FC-40/FC-75 solvent, is spun onto the patterned surface and then baked to evaporate the solvent and precipitate the Teflon® AF. The precipitate is a thermoplastic, so the substrate must be heated above the glass transition temperature for 30 minutes to promote a uniform surface coating. The surface at this point is superhydrophobic and ready to be diced to
a size that will fit into the test channel. The surface is recoated prior to use, to account for any contamination or handling damage during dicing.

The Laplace pressure for the microribs is 2 kPa, therefore the superhydrophobic surfaces were maintained in a narrow pressure range of 0.25 to 1 kPa (1 to 4 inH2O) to sustain air in the cavities during testing [34].

2.1.2 Riblet Process

Riblet structures for water must be much taller than the microribs. We used an additive process to produce the riblets. The riblet surface begins with a virgin silicon wafer. First two preparatory coats of MicroChem Omnicat™ are spun onto the surface. Each coat is allowed to bake and set. Then Microchem SU8-2075 photoresist is spun onto the surface and soft baked leaving a nominally 75 µm layer. An edge bead is created during the spinning process and must be removed with acetone to promote tighter feature patterning and closer alignment during exposure. The riblet pattern is exposed in the photoresist and the surface undergoes the post-exposure bake process to further cure the exposed photoresist. Then the photoresist is developed for 10 minutes and the remaining photoresist forms 75 µm riblets which repeat with a pitch of 160 µm. Although the riblet mask is designed to create 8 µm features, the 80+ µm separation between the wafer surface and the mask allows the UV light to spread [79], forming riblets that are nominally 14 µm.
Figure 2.3: SEM images of the riblet test surface.

wide. Because this spread of the UV light, the exposure time needs to increase accordingly to ensure the base is fully exposed.

After development, the remaining riblet structures undergo the hard bake process to further crosslink the photoresist in order to promote structural integrity and adhesion to the surface. An additional 1 µm layer of aluminum is added using the thermal evaporator. The purpose of this layer is to provide additional strength to the high aspect ratio features. A consequence is that is further widens the riblets to 16 µm. The surfaces are coated in a sacrificial layer of SU-8 2025 and given a soft bake to protect the riblets during dicing, and removed after dicing in SU-8 developer. Two SEM images of the riblet surfaces are shown in Fig. 2.3.

The same process used in creating the riblets, also works on a surface with small microfeatures to some degree. The riblet sturctures on the hybrid surfaces were manufactured in a similar manner, as will be seen in the next section.
2.1.3 Hybrid Feature Surface

The hybrid surface is made by combining the previous two processes. First the microribs are etched into the silicon, with every fourth rib being widened to form a 30 µm-wide platform, see Fig. 2.4. SU-8 2075 photoresist is then spun onto the etched surfaces. The wafers are allowed to bake at 55°C in a container sealed with parafilm to allow the thick photoresist to settle into the microfeatures and remove bubbles. After that the normal riblet photolithography steps are taken to form nominally 80 µm-tall riblets on top of the platform features. In a thermal evaporator the surfaces receive an aluminium layer and chromium layer to promote nano-scale uniformity and hydrophobic coating adhesion to the surface. As with the riblets the hybrid surfaces are coated with

<table>
<thead>
<tr>
<th>Feature</th>
<th>Dimension</th>
<th>Length (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Riblet</td>
<td>Height</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>Thickness</td>
<td>18</td>
</tr>
<tr>
<td></td>
<td>Pitch/Spacing</td>
<td>160</td>
</tr>
<tr>
<td>Riblet Platform</td>
<td>Width</td>
<td>30</td>
</tr>
<tr>
<td>Microrib</td>
<td>Height</td>
<td>15</td>
</tr>
<tr>
<td></td>
<td>Thickness</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Pitch/Spacing</td>
<td>40</td>
</tr>
</tbody>
</table>

Table 2.1: Hybrid surface dimensions.
SU-8 prior to dicing to protect the riblet features. The protective coat of photoresist is removed and the hybrid surfaces are then coated with the hydrophobic Teflon®AF coating. The completed surface is shown in Fig. 2.4. Geometry dimensions are listed in Table 2.1. The riblets were sized for optimum riblet performance and the microribs were designed consistent with the superhydrophobic surface microrib geometry.

It is worth noting at this point that the riblet structures may also function as superhydrophobic structures since they add three-dimensional texture to the surface and are given a hydrophobic coating. This is the case if the pressure is maintained between 0 and 0.44 kPa, the Laplace pressure for these superhydrophobic riblets. It was expected that once the Laplace pressure is exceeded or sufficient air is transferred into the flow, water would be forced in between the riblets and then the superhydrophobic microribs will contact the flow. This was not necessarily the case during actual testing. The behavior of the hybrid surfaces will be discussed in detail in the results section.

2.2 Experimental Setup

The test channel and PIV setup is shown in Fig. 2.5. The channel is composed of four acrylic sections machined from cast acrylic sheets. The section are a base, two side walls, and a top surface. The channel has a height of approximately 5.08 mm and width of 40.03 mm. Due to variation in wafer thickness the actual height of a given test may vary, but it was measured within ±10 µm using the calibrated PIV camera. Because of the material used in channel construction, tolerances along the channel are to ±50 µm. Fig. 2.6 shows a typical example of the height variation along the assembled channel. Machining, temperature, moisture absorption, and bolt torque all affect the actual dimension of the channel. All measurements were made at the same mid-wafer location on the fourth wafer to ensure consistent channel location and channel width. There is a small degree of variability in viewing location (±1mm), but no clear change in the measured values was observed.

The channel hydraulic diameter, defined as \( D_h = \frac{4A}{P_w} \), where \( A \) is the cross-sectional area and \( P_w \) is the wetted perimeter, is \( D_h = 9.02 \) mm with an aspect ratio of 7.87. The effective hydraulic diameter for the measurements is that of a two-dimensional channel, where \( D_h = 2H \approx 10.17 \) mm, \( H \) being the measured channel height from the top of the test surface to the smooth
wall on the opposite surface. This particular $D_h$ is more accurate since the tests measure a two-dimensional illuminated cross-section along the channel centerline, which will have comparable behavior to an infinite parallel plate channel. In the case of riblets, where the riblets protrude nominally 80 µm into the flow, $H$ included the protrusion height, $p_h$, which is the distance the riblets protrude into flow beyond the $u$ velocity profile origin, which is discussed later in this chapter.

A 500 µm depression is machined in the base wall to hold the test surfaces. The test section begins after a developing entrance length of 64 $D_h$. The section containing the test surfaces is 27 $D_h$ long, allowing the fully developed flow to further develop over the test surfaces [80, 81]. The transparent top surface retains its factory finish and allows the laser plane to shine through and illuminate a longitudinal slice of the flow near the channel center. The sidewalls are also
Figure 2.6: Example of height variation and the tolerance along the assembled channel.

Figure 2.7: Schematic of flow loop components.

transparent so that light scattered by particles can be imaged by the camera. Detailed drawing are available in Section B.1 of Appendix B.
2.2.1 Flow Loop Components

The flow loop, illustrated in Fig. 2.7, begins at a reservoir exposed to atmospheric pressure. At this point air can be bubbled to ensure the distilled water is saturated with air, which is critical in long duration testing of flow over superhydrophobic surfaces for preventing removal of the air in the cavities between the microribs [34]. It also helps regulate the water temperature by slowing system heating to 0.2°C per test. The temperature is measured in the reservoir using a type-K thermocouple, calibrated in an ice bath. The reservoir itself can be elevated or lowered to control the static pressure in the flow loop. The water then passes into a March AC-3CP-MD centrifugal pump running at full capacity. The flow rate is adjusted using a gate valve immediately downstream of the pump. An initial flow measurement is obtained with a rotameter-type flowmeter so that the gate valve can target the correct flow rate. The entire system can handle operating flow rates from 2-22 L/min, which corresponds a Reynolds number range from $1 \times 10^3$ to $2.5 \times 10^4$.

To dampen large fluctuations and upstream turbulence, the water passes through a flow conditioner which prepares the flow to enter the test section. In the flow conditioner the flow is expanded and passed through a viscous damping section filled with marbles, followed by a straightening section comprised of packed straws. The flow then accelerates smoothly in a machined converging nozzle as it approaches the main channel. Custom connections were used to smoothly transition from circular tubing to a rectangular channel.

At the channel entrance a rectangular mesh grating is present to force transition to turbulence in the channel as the flow develops. The mesh is standard 18-16 window screen and has a spacing of 1.4 mm and diameter of 0.25 mm. The mesh is necessary to ensure transition to turbulent flow in the low turbulent $Re$ range. After passing through the test channel, the flow undergoes another smooth transition from rectangular to circular tubing and returns to the reservoir at atmospheric pressure. Additionally, a bypass loop around the channel was installed so that air pockets could be removed from the flow loop, and so the entire flow could become uniformly saturated with air prior to the surfaces being immersed in water.

The channel itself was designed to have an adjustable angle. Similar to the reservoir height, which can be adjusted to change the static pressure in the system, the angle of the channel can be used to adjust the relative change in pressure along the channel and thus maintain the superhydrophobic surface under near-uniform low pressure.
2.2.2 Particle Image Velocimetry Setup

This PIV experiment used a double-pulsed laser system for particle illumination. Two Q-switched Nd: YAG pulsed lasers fabricated by Big Sky were used for all experiments. The lasers produce light at a wavelength of 510 nm and have adjustable power settings. The DaVis software from LaVision was used to synchronize the camera and laser, which had a combined maximum operating frequency of 4.5 Hz. The laser beam traveled through a series of mirrors that brought the laser light down vertically onto the channel, where laser optics were used to focus the light and spread the laser beam into a narrow plane. Laser illuminated images were used to focus the beam. Adjustments targeted brighter particles, narrow wall illumination, and particle focus. Because of concerns about out of plane particle illumination for small turbulent scales, a plate with a hole was used to reduce the beam diameter before it passed through the focusing lens. Safety glasses were worn at all times working with the laser and a mechanical “stop” was installed so untrained persons would not encounter laser radiation when entering the lab.

The LaVision Imager Intense, a high intensity 12 bit CCD camera with 1376 x 1040 pixel resolution, was used to capture the images of the illuminated particles in the flow. In order to achieve high resolution a Navitar 100 mm lens, 78 mm of extension tubes, and a +4 closeup lens were used to achieve an image pixel size of 4.35 µm. An f-number of 5.6 was used. This ensured that particles remained in focus eliminated image aberrations near the image edges. The camera tripod was modified to provide six degrees of freedom, with translation and rotation in all three dimensions.

A typical PIV image is shown in Fig. 2.8 (A), where the flow direction is from left to right and the walls have been marked with a dotted line. High density seeding, as is shown in the figure, was shown to yield the best results; however, very sparse seeding had to be present for the results to be noticeably affected. The PIV domain, shown in Fig. 2.8 (B), was approximately 5.10 mm from wall to wall and 3.28 mm along the channel, where the 12 pixels (52 µm) nearest the wall were masked, to limit reflections. The portion of the image beyond the domain was not used to avoid spurious edge effects.

Before imaging, the camera was calibrated. The channel is designed so that it can translate forward and backward on a fixed track. This way the camera can be calibrated in the laser plane outside of the channel and the channel can be slid into view with no adjustment of the camera, thus
retaining the scale and calibration. This process used a miniature channel replica which was filled with water and resting atop the channel with a calibration plate inside. On the plate was a grid of 0.12 mm diameter dots spaced at 0.5 mm intervals.

### 2.2.3 PIV Seeding Particles

The flow was seeded with TiO$_2$ particles which are quite dense, but exhibit a strong light scattering capability. The particles underwent a settling process to target a particle size ranging between 1 to 6 µm, but there is a small distribution of larger particles in the flow. The particle settling time was estimated using Stokes Law, where the particle settling velocity, $u_{\text{part}}$ is defined in Eqn. 2.1,

$$u_{\text{part}} = \frac{2 \rho_p - \rho_f}{9 \mu} g a^2$$  \hspace{1cm} (2.1)

where $a$ is the radius, $g$ is gravitational acceleration, $\rho_p$ is the particle density, and $\rho_f$ is the fluid density. Given that the particles were allowed to settle 1.5 inch over 15 minutes, the particle
settling velocity of the seeding is less than 0.042 µm/µs. At this rate, the largest particles could
counter as much as 0.01 (1% of $\bar{u}_{avg}$) to $\bar{v}$. However, the flow is highly turbulent so some of the
settling effect is negated. Additionally, $\bar{v}$ is not used in any further calculations, so the only error
contribution is in the mean velocity.

The Stokes number, $St$, is the ratio of particle response time to the Kolmogorov time scale
and is a measure of how well particles track the flow, is shown in Eqn. 2.2,

$$St = \frac{\rho_p D_p^2 u}{18 \mu L}$$ (2.2)

where $\rho_p$ is the particle density, $D_p$ is the particle diameter, $u$ is the local flow velocity, $L$ is the
characteristic length $D_h$. Fundamentally the Stokes number is the ratio of particle response time to
the characteristic fluid time scale. For these experiments $St < 0.004 \ll 1$, indicating good particle
tracking of the flow.

When considering particle tracking it can also be useful to calculate the frequency at which
particles will respond. The characteristic frequency of the particles, $f_{char}$, was calculated using
Eqn. 2.3 [82]

$$f_{char} = \frac{18 \mu}{\rho_p d_p^2} = \frac{C_p}{2\pi}$$ (2.3)

where $C_p$ is the characteristic particle frequency in rad/s. This frequency can be used to estimate the
ability of a particle to track the flow, using Eqn. 2.4 which is the ratio of fluctuation energies written
in terms of the turbulent velocity fluctuation frequency, $\omega_c$ and the particle characteristic frequency,
$C_p$, as presented by Melling [82], using the characteristic frequency one can use Eqn. 2.4.

$$\frac{\bar{u}_p^2}{\bar{u}_j^2} = \left(1 + \frac{\omega_c}{C_p}\right)^{-1}$$ (2.4)

<table>
<thead>
<tr>
<th>Particle Size</th>
<th>.95 Response Ratio</th>
<th>.99 Response Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 µm</td>
<td>33.15 kHz</td>
<td>6.36 kHz</td>
</tr>
<tr>
<td>2 µm</td>
<td>8.29 kHz</td>
<td>1.59 kHz</td>
</tr>
<tr>
<td>5 µm</td>
<td>1.32 kHz</td>
<td>0.25 kHz</td>
</tr>
<tr>
<td>10 µm</td>
<td>0.33 kHz</td>
<td>0.06 kHz</td>
</tr>
</tbody>
</table>
Table 2.3: PIV processing parameters and settings.

<table>
<thead>
<tr>
<th>Process Parameter</th>
<th>Setting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Processing software</td>
<td>LaVision: DaVis PIV Software</td>
</tr>
<tr>
<td>Image processing</td>
<td>None</td>
</tr>
<tr>
<td>Masking</td>
<td>Masked Region:</td>
</tr>
<tr>
<td></td>
<td>11 px inside channel walls</td>
</tr>
<tr>
<td></td>
<td>10 px from entrance/exit</td>
</tr>
<tr>
<td>Correlation mode</td>
<td>Cross Correlation</td>
</tr>
<tr>
<td>Processing iteration</td>
<td>Multipass processing (decreasing window size)</td>
</tr>
<tr>
<td>Window size</td>
<td>128x128 → 64x64 → 32x32 (2 passes)</td>
</tr>
<tr>
<td>Window overlap</td>
<td>50% overlap</td>
</tr>
<tr>
<td>Image correction</td>
<td>None</td>
</tr>
<tr>
<td>Correlation function</td>
<td>Standard I1:I2 (FFT, no zero-padding)</td>
</tr>
<tr>
<td>Vector postprocessing</td>
<td>Enabled</td>
</tr>
<tr>
<td>Median filter</td>
<td>Strongly remove &amp; iteratively replace:</td>
</tr>
<tr>
<td></td>
<td>Remove is diff. to avg. &gt; 2×rms of neighbors</td>
</tr>
<tr>
<td></td>
<td>(Re)insert if diff. to avg. &lt; 2.5×rms of neighbors</td>
</tr>
<tr>
<td>Vector removal</td>
<td>Remove groups with &lt; 5 vectors</td>
</tr>
<tr>
<td>Interpolation of empty</td>
<td>None</td>
</tr>
<tr>
<td>spaces</td>
<td>None</td>
</tr>
<tr>
<td>Smoothing</td>
<td>None</td>
</tr>
<tr>
<td>Image pairs</td>
<td>4000</td>
</tr>
<tr>
<td>Averaging along channel</td>
<td>Rows: 4 - 60</td>
</tr>
</tbody>
</table>

A ratio of fluctuation energies near 1 corresponds with good frequency response and can be approximated in terms of the characteristic frequency of the particle using the relation Particle response frequencies were calculated for .95 and .99 response using Eqn. 2.4 and are listed in Table 2.2. The response times for different sizes of particle varies significantly, but they respond sufficiently quick to resolve the very fast Kolmogorov time scales which were estimated to be less than 0.030 kHz, based on the maximum turbulent energy dissipation, $\varepsilon$ (which is comparable to maximum turbulent kinetic energy production). The Kolmogorov time scale, $t_d$ can be found using Eqn. 2.5.

$$t_d = \left(\frac{\nu}{\varepsilon}\right)^{1/2}$$

(2.5)

2.2.4 Processing

The instantaneous vector fields were computed using a multipass processing algorithm, beginning with 128x128 pixel interrogation windows and ending with a double pass with 32x32
First and second order turbulent statistics are then computed for each location. These included the mean velocity of the flow and the fluctuating velocity statistics (the variances of $u$ and $v$ and the covariance of $u$ and $v$ which form the Reynolds stresses). For each run 4000 image pairs were collected. This proved to be sufficient for converged turbulent statistics and showed similar finding to that of Woolford [55]. No improvement was found by including more image pairs (6000 or 8000), but the higher order turbulent statistics showed more variation when less image pairs were used (1000 or 2000). The results from 60 interrogation windows along the 3.28 mm streamwise PIV domain were then averaged, leaving a single profile across the channel for each of the turbulent quantities from the test surface to the smooth top wall. Further processing was done to compute related quantities such as viscous stress, total shear stress, and turbulence production.

The raw PIV data, wall locations, and temperature information were then used to compute the normalized velocities and turbulent statistics, including Reynolds stress, total shear stress, and friction factor, as well as important channel and flow details such as $\bar{u}_{avg}$ and Reynolds number. These normalized quantities are shown in Table 2.4, listed by the property name, the symbol for the normalized value, and the normalization. Quantities include the time averaged streamwise velocity, $U$, the turbulence intensity or rms velocity in the streamwise, $U_{rms}$, and wall-normal directions, $V_{rms}$, the turbulent or Reynolds shear stress, $T_{turb}$, and the turbulence production, $P$. Y
Figure 2.9: Schematic showing the apparent origin of a riblet surface.

is the normalized wall-normal coordinate $Y$. The shear stress terms are normalized by the dynamic pressure, $\rho \bar{u}^2_{avg}$.

The wall location was identified from the reflection of particles at the wall. The location was selected by hand at the mid-location of the particle image using several images. The accuracy of this method was $\pm 2$ pixels (95% confidence) in most cases. The variation of the wall location itself over the whole image was $\pm 1.5$ pixels ($\pm 7$ µm). For superhydrophobic surfaces the flow was suspended above the microfeatures so the method of reflections was an accurate determination of the “wall” location. For the case of riblets, the wall location, in terms of the shear stress, is effectively where the apparent $u$ velocity profile goes to zero. This location is illustrated in Fig. 2.9, a figure similar to the one published by Bechert and Bartenwerfer [83]. The apparent origin of the velocity profile is measured using the protrusion height, $p_h$, the distance the riblets protrude into the flow beyond $u$ velocity origin. The flow velocity and momentum transfer are negligible for fluid between the riblets beyond this point. Bechert et al. [84] found that $p_h = 0.021 \cdot s$ for blade riblets, which corresponds to $p_h = 33.6$ µm in the current study.
2.3 Uncertainty Analysis

There are a number of sources of uncertainty in the measurements. This section evaluates the contributions of uncertainty and their propagation into the reported results. First the errors in the actual PIV measurement are found. Then the error in the quantities $\bar{u}$ and $\bar{v}$, $d\bar{u}/dy$, $\bar{u}_{avg}$, and $\bar{u'}v'$ is evaluated. The final discussion will cover the propagation of all errors into the friction factor computation, where a table will describe the contributions of each of these errors as a function of Reynolds number.

2.3.1 Particle Displacement and Velocity Error

Each PIV displacement measurement will have a bias and a precision error \[85, 86\]. The bias toward integer values results from the subpixel displacement computation. It is assumed that the DaVis software uses an algorithm comparable error to the Gaussian fit. Forliti et al. found that, for a 128x128 domain using Gaussian fit, the bias, $\beta_x$, is 0.03 px and the precision error, $\sigma_x$, is 0.03 px \[85\]. The total error in image pixel displacement (for 95% confidence interval) is computed using a root sum squared method as shown in Eqn. 2.6,

$$\chi_\Delta x = \sqrt{\beta_x^2 + (2\sigma_x)^2} \quad (2.6)$$

where $\chi_\Delta x$ is the total position error. The $\pm 0.22$ deg error in alignment resulting from a typically 5 pixel misalignment over 1024 pixel range was not included since its effect on $\chi_\Delta x$ is negligible.

A strong bias toward lower displacements can occur when there is a strong velocity gradient. This gradient bias was investigated by Keane and Adrien who quantified the bias as a function of the displacement gradients \[87\]. They plotted variation in measured velocity, $M\sigma_u\Delta t/d_I$, in terms of the relative image displacement, $M\Delta u\Delta t/d_\tau$, where $M$ is the magnification, $\sigma_u$ is the velocity error, $\Delta t$ is the time step, $d_I$ is the image diameter (roughly the window size), $\Delta u$ is the change in displacement over half the window, and $d_\tau$ is the seed particle image diameter. For this paper, these terms can be written more familiarly in Eqn. 2.7 and 2.8 as:

$$\frac{M\sigma_u\Delta t}{d_I} = \frac{\sigma_u\Delta t}{C} \cdot \frac{1}{d_I} = \frac{\sigma_{\Delta x_{grad}}}{d_I} \quad (2.7)$$
Table 2.5: Velocity-reducing bias error for high gradient regions.

<table>
<thead>
<tr>
<th>( \frac{M \Delta u \Delta t}{d_\tau} )</th>
<th>( \frac{M \sigma_\Delta \Delta t}{d_\tau} )</th>
<th>( \delta x )</th>
<th>Gradient (px/px)</th>
<th>Bias Error (px)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>0.035</td>
<td>3</td>
<td>0.094</td>
<td>0.064</td>
</tr>
<tr>
<td>1</td>
<td>0.020</td>
<td>6</td>
<td>0.188</td>
<td>0.22</td>
</tr>
<tr>
<td>2</td>
<td>0.007</td>
<td>12</td>
<td>0.375</td>
<td>0.64</td>
</tr>
<tr>
<td>3</td>
<td>0.002</td>
<td>18</td>
<td>0.56</td>
<td>1.12</td>
</tr>
</tbody>
</table>

where \( \sigma_{\Delta x_{\text{grad}}} \) is the error in the displacement due to a velocity gradient, \( C \) is the calibration (px to mm), and \( \delta x \) is change in displacement across half the window. The seed particle image diameters for the PIV ranged from 3-4 px. The condensed results are shown in Table 2.5. The largest of the velocity gradients measured was \( \delta x \approx 3 \), corresponding to a bias of 0.064 px. The gradient bias therefore has little effect on the data. Additionally it acts on all data runs equally causing, at most, a 1% error in the velocity profile near the walls.

The velocity is computed using Eqn. 2.9 and 2.10

\[
\begin{align*}
    u &= C \frac{\Delta x}{\Delta t} \\
    v &= C \frac{\Delta y}{\Delta t}
\end{align*}
\]

where \( \Delta x \) and \( \Delta y \) are the window shifts in x and y, \( C \) is the calibration 4.350 \( \mu \)m/px, and \( \Delta t \) is the time between laser pulses which range between 16 and 80 \( \mu \)s. The uncertainty in calibration, \( \sigma_C \), is 0.0089 \( \mu \)m/px and the jitter in later pulses, \( \chi_{\Delta t} \), is 1 ns. The random and bias errors can be propagated as a velocity uncertainty value [88] as shown in Eqn. 2.11

\[
\sigma_u = \frac{1}{2} \sqrt{\left( \frac{\partial u}{\partial \Delta x} \right)^2 \chi_{\Delta x}^2 + \left( \frac{\partial u}{\partial C} \right)^2 \chi_C^2 + \left( \frac{\partial u}{\partial \Delta t} \right)^2 \chi_{\Delta t}^2}
\]

Substituting Eqns. 2.9 and 2.10 into Eqn. 2.11, the error can be written explicitly, as shown in Eqn. 2.12 for \( \sigma_u \).
\[
\sigma_u = \frac{1}{2} \sqrt{\left(\frac{C}{\Delta t}\right)^2 \chi_{u}^2 + \left(\frac{\Delta x}{\Delta t}\right)^2 (2\sigma_C)^2 + \left(-\frac{C\Delta x}{\Delta t^2}\right)^2 \chi_{\Delta t}^2}
\] (2.12)

The errors in \(u\) and \(v\) measurements are largely independent of \(Re\), with a total velocity error ranging from 1.0\% of \(\bar{u}_{avg}\) at lower \(Re\), up to 0.85\% of \(\bar{u}_{avg}\) for \(Re = 23,000\). The displacement error is the largest contributor to the error in \(u\), followed by the calibration error which is 30\% smaller. Laser timing error was negligible. The displacement error was the dominant error for \(v\), due to relatively small displacements.

### 2.3.2 Uncertainty in \(\bar{u}\) and \(\bar{v}\)

The average velocity at each location across the channel, \(\bar{u}\) and \(\bar{v}\), is computed by averaging first over 4000 image pairs and then over 56 longitudinal locations. The error in the average comes from 3 sources, actual fluctuations in the velocity \((\bar{u}'\bar{u}')\), propagated random errors \((\chi_u)\), and systematic bias errors \((\beta_u = C\beta_x/\Delta t)\). The standard deviation of the average velocity due to random error and fluctuation is shown in Eqn. 2.13

\[
\sigma_{\bar{u}} = \sqrt{\frac{\bar{u}'\bar{u}'}{n} + \frac{\sigma_u^2}{n}}
\] (2.13)

where \(n\) is the number of samples (240,000 from 60 samples of 4000 image pairs). In the interest of being thorough, a bias error is included in the velocity error, which would add error to \(\bar{u}\) that does not decrease with increasing sample size. This would be included in the total error in \(\bar{u}\) as shown in Eqn. 2.14 for 95\% confidence.

\[
\chi_{\bar{u}} = \sqrt{(2\sigma_{\bar{u}})^2 + \beta_u^2}
\] (2.14)

Note that \(\beta\), the bias, is a systematic error in PIV processing and is not reduced with increased sample size making it the largest source of error by an order of magnitude at 0.5\% of \(\bar{u}_{avg}\), followed by the velocity fluctuation \(\bar{u}'\bar{u}'\). This bias is visible in the \(v\) velocity profile, which generally shows a positive velocity of 0.005 \(\cdot\bar{u}_{avg}\) across the channel.
2.3.3 Uncertainty in $du/dy$ and $\bar{u}_{avg}$

Knowing the error in the average velocity, $\chi_{u}$ one can compute the error in the velocity gradient ($du/dy$) and the average velocity over the entire channel ($\bar{u}_{avg}$), which are important quantities relating to the accuracy of the friction factor. The velocity gradient is computed from the average slope, shown in Eqn. 2.15

$$\frac{d\bar{u}}{dy} \approx \frac{1}{2} \left( \frac{\bar{u}_{i+1} - \bar{u}_{i}}{\Delta y} + \frac{\bar{u}_{i} - \bar{u}_{i-1}}{\Delta y} \right) = \frac{\bar{u}_{i+1} - \bar{u}_{i-1}}{2\Delta y}$$  \hspace{1cm} (2.15)

where $\bar{u}_{i}$ the average velocity value at the evaluation location and $\Delta y$ is the distance between data points in the y-direction. The error in the velocity gradient due to the central difference computation is approximately $0.01 (d\bar{u}/dy)$ across most of the channel ($0.1 < y/H < 0.9$). The error in $\Delta y$ is very small and may be neglected. A bias error in $\bar{u}$ is not included, since any bias would cancel in the subtraction. Finally, the errors can be propagated and combined with a root mean squared method as shown in Eqn. 2.16.

$$\chi_{\frac{du}{dy}} = \sqrt{\frac{1}{2(\Delta y)^2} (2\sigma_{u})^2 + \left(0.01 \frac{d\bar{u}}{dy}\right)^2}$$  \hspace{1cm} (2.16)

From the numerical computations, it was observed that the error in $d\bar{u}/dy$ is small, less than 1.5% of $d\bar{u}/dy$, whereas the numerical error is generally larger than the velocity error. Regions very close to the wall errors as large as 7%, so the velocity gradient at the wall cannot be used as a method for determining shear stress.

The error in $\bar{u}_{avg}$ includes error from the numerical integration across the channel, $\chi_{fit}$. The integration was performed using a second-order fit to the data and the error was estimated to be less than 0.2% of the total, based on a numerical computation of an exact profile. Additional uncertainty in $\bar{u}_{avg}$ derives from the propagated error $\chi_{u}$ and any systematic errors. Combining the integration error and $\chi_{u}$ using a root mean square approach yields Eqn. 2.17.

$$\chi_{\bar{u}_{avg}} = \sqrt{(2\sigma_{\pi})^2 + \beta_{u}^2 + (0.002U_{avg})^2}$$  \hspace{1cm} (2.17)

The uncertainty in $\bar{u}_{avg}$ is dominated by $\beta_{u}$, although the numerical integration has some contribution. The error in the value of $\pi_{avg}$ is approximately 0.5%.
2.3.4 Uncertainty in $u'v'$

The final velocity error quantity to be addressed is the error in the Reynolds stress, $\chi_{u'v'}$. There are three primary sources of error, the error in the velocity measurement, $\chi_u$, the actual fluctuation in the Reynolds stress, and error from insufficient spatial resolution. There is not enough vector information preserved to accurately assess the error in the Reynolds stress, but the effect of PIV velocity error on the Reynolds stress must be small. The variation in $u'v'$ from natural fluctuations is reduced by a factor of $\sqrt{n}$ and will be relatively small.

As $Re$ increases, the velocity fluctuations become comparable to the window size and can be averaged across the laser beam width, causing $u'v'$ to be under-represented. This is evidenced by noted decrease and nonlinearity in the total shear stress at higher $Re$, due to the overly low Reynolds shear stress that is computed. This will be discussed later in the results section.

2.3.5 Friction factor uncertainty

The friction factor is calculated by fitting a linear profile to $\tau_w$, which is defined again for convenience in Eqn. 2.18

$$\tau_{tot} = \mu \frac{d\bar{u}}{dy} - \rho \bar{u}'v'$$

(2.18)

Sources of error in $\tau_{tot}$ include $\mu$, which is affected by temperature measurement error, $\chi_T$, $\chi_{d\bar{u}/dy}$, and error in the Reynolds stress, $\chi_{u'v'}$. These errors can be directly substituted into Eqn. 2.18 and propagated to find the error in $f$. Once $\tau_{tot}$ is known, the profile is fit to a line and the intercept at the wall is used to find $\tau_w$. Errors in both the linear regression fit of the $\tau_{tot}$, $\chi_{fit}$ and the wall location, $\chi_{wall}$, directly affect the error in $\tau_w$. Once $\tau_w$ is determined, $f$ can be computed as shown in Eqn. 2.19.

$$f = \frac{8\tau_w}{\rho \bar{u}_{avg}^2}$$

(2.19)

The calibrated thermocouple reading has an error $\chi_{therm}$ of 0.5°C, but the actual temperature will also change over the course of a test. The total variation in temperature can be estimated by Eqn. 2.20.
\[ \chi T = \sqrt{\chi_{therm}^2 + \left(\frac{\Delta T}{2}\right)^2} \]  

(2.20)

where \( \Delta T \) is the change in temperature over an experimental run at a single \( Re \). This error propagates as an error in \( Re \) as well as the viscous stress. An error where temperature is underpredicted leads to higher viscosity value, meaning lower \( Re \) but increased \( f \). These two error sources cancel each other to some degree, as they are not independent.

With the shear stress profile error accounted for, \( \tau_w \) can be determined. \( \tau_w \) will include error from both the linear fit and wall location, in addition to the shear stress errors already accounted for. The error in the least squares fit of the total shear stress can be calculated using standard linear regression uncertainty analysis [89] with Eqn. 2.21, which states

\[ \chi_{fit} = 2 \sqrt{\frac{SSE}{n-2}} \sqrt{\frac{1}{n} + \frac{(X_i - \overline{X})^2}{SS_x}} \]  

(2.21)

where \( SSE \) is the sum of the squared errors, \( n \) is the number points, \( X_i \) is location of interest, \( \overline{X} \) is mean value of \( X \) for the point distribution, and \( SS_x \) is the sum of the squares of \( X \). The error in wall location \( \chi_{wall} \) is approximately 2 pixels or about 9 \( \mu m \) and is based on the ability of the examiner to locate the wall, by finding the midpoint between the particle and reflected image. The error is linear and directly affects the intercept value of the shear stress fit.

These can all be propagated through direct substitution to find their respective influence on the friction factor. Since it was observed that the error is largely independent of surface type, an average error is reported for all the surfaces here. The error magnitudes at several Reynolds numbers are shown in Table 2.6. The error in the least squares fit to the total stress is generally the largest source of error, especially at higher flow rates. This comes from non-linearity of the total stress that propagates from errors in the velocity variance, \( \overline{u'v'} \). At low flow rates where the flow is laminar and/or transitional the velocity gradient error is very large, propagated by the dominant viscous stress. The wall location error provides a small consistent source of error, that is only minimally affected by Reynolds number. Temperature error is more important at lower flow rates where viscous forces dominate the flow. Curiously there is a minimum temperature error in the low turbulent Reynolds numbers, where the change in friction factor due to temperature is countered by the error in Reynolds number due to temperature.
Table 2.6: Average friction factor error contributions as a function of flow rate (95% confidence).

<table>
<thead>
<tr>
<th>Flow Rate (LPM)</th>
<th>Reynolds Number</th>
<th>Least Squares Fit Error (%)</th>
<th>Velocity Grad. Error (%)</th>
<th>Wall Location Error (%)</th>
<th>Temp. Error (%)</th>
<th>Total Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>1800</td>
<td>1.04</td>
<td>1.50</td>
<td>0.24</td>
<td>0.92</td>
<td>2.33</td>
</tr>
<tr>
<td>4</td>
<td>3600</td>
<td>0.39</td>
<td>0.59</td>
<td>0.19</td>
<td>0.20</td>
<td>1.06</td>
</tr>
<tr>
<td>6</td>
<td>5500</td>
<td>0.38</td>
<td>0.22</td>
<td>0.17</td>
<td>0.11</td>
<td>0.70</td>
</tr>
<tr>
<td>8</td>
<td>7600</td>
<td>0.42</td>
<td>0.14</td>
<td>0.17</td>
<td>0.17</td>
<td>0.70</td>
</tr>
<tr>
<td>10</td>
<td>9500</td>
<td>0.53</td>
<td>0.11</td>
<td>0.17</td>
<td>0.23</td>
<td>0.84</td>
</tr>
<tr>
<td>12</td>
<td>11700</td>
<td>0.63</td>
<td>0.09</td>
<td>0.17</td>
<td>0.26</td>
<td>0.98</td>
</tr>
<tr>
<td>14</td>
<td>13500</td>
<td>0.69</td>
<td>0.08</td>
<td>0.17</td>
<td>0.25</td>
<td>1.05</td>
</tr>
<tr>
<td>16</td>
<td>15500</td>
<td>0.89</td>
<td>0.08</td>
<td>0.17</td>
<td>0.26</td>
<td>1.31</td>
</tr>
<tr>
<td>18</td>
<td>17700</td>
<td>1.01</td>
<td>0.07</td>
<td>0.16</td>
<td>0.26</td>
<td>1.48</td>
</tr>
<tr>
<td>20</td>
<td>20000</td>
<td>1.06</td>
<td>0.07</td>
<td>0.16</td>
<td>0.29</td>
<td>1.55</td>
</tr>
<tr>
<td>22</td>
<td>22500</td>
<td>1.09</td>
<td>0.06</td>
<td>0.16</td>
<td>0.27</td>
<td>1.58</td>
</tr>
</tbody>
</table>

It should be noted that not only is the least squares fit error the largest of the direct errors; however, there is another large source of error. The under-representation of $u'v'$ near the wall, means that the nearer points are to the wall, the less accurate they will be, thus the selection of how close to the wall to include points adds a substantial amount of variability, and any choice could be seen as subjective. Points closer to the wall will under-estimate the friction factor, but points in the channel center are far from the wall, requiring a large degree of extrapolation. For consistency, the 12 PIV windows nearest the wall, corresponding to $Y < 0.15$ and $Y > 0.85$, were not included in the linear fit. Since the profiles were all taken with the same setup at the same location, the relative differences between test surfaces should hold relatively constant. It is the case that many of the surface types were tested multiple times. The averaged values for the runs and the respective standard deviations will be reported, as needed.
CHAPTER 3. RESULTS

In this section the experimental results are presented. First, qualitative results are discussed relating to the behavior and conditions of the superhydrophobic and hybrid surfaces. Wetting behaviors, pressure response, and surface appearance are described in this section. Then, the PIV velocity results are presented, beginning with the velocity profiles across the channel. The averaged and fluctuating profiles are explored and differences between surfaces types are discussed. Next, the shear stress across the channel is presented, followed by a section where the friction factor results are plotted against Reynolds numbers and the overall drag reduction is characterized. Lastly, the changes in turbulence kinetic energy production are discussed for the different surface types.

3.1 Qualitative Results for Superhydrophobic and Hybrid Surfaces

Both the superhydrophobic and hybrid surface utilize the apparent slip achieved by the air-water interface in an effort to achieve a greater drag reduction. One of the more difficult aspects of research with superhydrophobic surfaces is the delicate and variable nature of the surfaces. Not only are they susceptible to contamination, friction, gas diffusion, and pressure, but the meniscus and plastron appear to have some elastic effects. Most of the time observers are only able to discern the current state of the surface through variations in the way light reflects off the surface at different angles and the way the surface responds to stimuli such as pressure pulses, slow pressure swings, and changing saturation of air in the water. All of these change the state of the surfaces and can be harmful to the superhydrophobic suspension of the meniscus above the cavities. In addition to the limited evaluations that can be made, many experiments are performed in a closed and opaque apparatus, further limiting knowledge concerning the actual surface working state.

The observations and investigation of the plastron state of the superhydrophobic and hybrid surfaces that will follow hereafter are included to shed some light, however rudimentary, on the dynamics and behavior of superhydrophobic surfaces in turbulent conditions. To this end, pho-
Figure 3.1: Overhead (A) and side (B) images of a superhydrophobic surface in turbulent flow under vacuum conditions. Surfaces with and without corrals are shown.

Photographs of the surfaces during testing are included and specific conditions at the time of testing are discussed. Some implications based on the observation for this test and research in general are mentioned.
3.1.1 Superhydrophobic Surfaces Observations

In the initial testing of the superhydrophobic surfaces for this effort, the effectiveness of breaker ridges/corrals was tested. As discussed previously, breaker ridges are sparsely spaced transverse ridges that effectively divide the cavity regions into cells. On the test surfaces, breaker ridges were spaced every 2500 µm, as was shown in Fig. 2.1. For this setup, in which surfaces with 32 µm cavities underwent turbulent flow saturated with air, the surfaces began to flood near a pressure of 2.5 kPa (10 inH₂O), just over the Laplace pressure. This was observed visually as the shimmering mirror layer atop the cavities vanished leaving a very dark and opaque non-reflective surface over the course of 20 minutes. As further evidence of complete wetting, once the pressure was lowered to vacuum pressures, there appeared to be no recovery of the superhydrophobic sheen nor any formation of bubbles.

Figure 3.1 (A) shows an overhead view of two superhydrophobic surfaces aligned in sequence along the flow direction when the flow is subjected to vacuum pressure. The upstream surface (bottom left) has standard rib and cavity structures aligned parallel to the flow direction. Streaks are visible indicating non-uniform pressure caused by flow vibration and air loss. The downstream surface (top right) has visible corrals crossing the surface, effectively breaking the surface into closed cells. The greater uniformity of the surface indicates more stable menisci, subject to less vibration and air loss. The majority of the bubbles that are propagating along the surface with breaker ridges (draining air) clearly originate from the superhydrophobic surface without corrals. Additionally, the surface without corrals appeared to have visible surface oscillations of a lower frequency due to turbulence. Surfaces with corrals appeared to be much more robust in both positive and negative pressures and less likely to wet or bleed off air through the end of the surface. The breaker ridge surfaces showed more limited oscillation, but with high Re flows, they showed higher frequency oscillations.

Figure 3.1 (B) shows a side view of the channel, with the corralled surface upstream now, in which the superhydrophobic surfaces are experiencing vacuum pressures. Bubbles that formed are observed on the surfaces with a concentration of bubbles at the interface between them. The signature superhydrophobic sheen, indicating a plastron presence, can be seen more strongly on the upstream surface with corrals (left).
The transient effect of vacuum pressures above the superhydrophobic surfaces is worth presenting. A one-hour timelapse video was taken to evaluate the transient effects of exposure to vacuum pressure at $Re \approx 1.6 \times 10^4$. Stills from the video are shown in Fig. 3.2 from left to right. During startup (upper left), the surface is initially maintained in the Cassie-Baxter state during startup. From above, the surface appears dull (from the side the familiar sheen is present). The surface then undergoes a test pressure reduction (top center) causing the menisci to bow outward and reflect a more light. Vibrations of the menisci due to turbulent flow are observed along the surface. At that point the pressure is returned to positive pressure and the original state appears to be recovered (top right).

The pressure is once again lowered to 0.25 kPa vacuum pressure (second row). After several minutes bubbles begin to form along the surface and travel downward leaving darkened streaks, ostensibly a patch of cavities that have partially evacuated and returned to a flatter meniscus. Over time more bubbles form and the streaks begin to cover the entire surface (third row). The streaks also begin to diminish in sharpness, indicating some sharing of the air between the closed cells formed by cavities and corrals (fourth row). A decrease in overall brightness is observed indicating flatter menisci and an overall loss in air from the cavities. Recovery to the Cassie-Baxter state was not observed upon restoration of positive pressures (not pictured).

For many of the results that will be reported later in this thesis, the $F_c = 0.8$ superhydrophobic surfaces used in testing showed little change from control surfaces. In terms of turbulence production and shear stress they actually show worse performance. Repeated testing confirmed that this was consistently the case. Tests were done with fully wetting superhydrophobic surfaces and the performance was nearly identical to those in the Cassie state. These interesting and somewhat unexpected results may be due to the presence of breaker ridges on the surface, causing the flow above the cavity to start moving and stop suddenly, periodically. Data was not taken for surfaces in the vacuum state with protruding menisci, but may be interesting for future study.

3.1.2 Hybrid Surfaces Observations

The hybrid surfaces offer a unique set of advantages, namely, the ability to perform well regardless of the state of the air in the cavities, potential for giant slip when the riblets act superhydrophobically, and multi-tier superhydrophobic robustness. These benefits can be enhanced with
Figure 3.2: Timelapse stills recording the superhydrophobic surface reaction to 0.25 kPa vacuum pressure. Images are spaced in time fairly evenly over the course of one hour. Camera is situated above the channel and ambient room lighting is used.
further improvements such as more structurally robust features, the addition of smaller micro- and nano-scale roughness, and improved surface coatings, all of which are beyond the scope of this thesis.

The surfaces do have one major challenge which is ascertaining the current surface state. Much like the superhydrophobic surfaces, where our understanding of the exact meniscus shape and position is fairly limited, the state of hybrid riblet-superhydrophobic surfaces is difficult to determine. In fact the state is substantially more complicated, owing to the fact that there are more possible states for the surface, but also due to the tendency toward partial wetting in localized regions.

Fig. 3.3 is a set of illustrations showing some of the possible wetting states of the hybrid riblet-superhydrophobic surfaces. Fig. 3.3 (A-C) are states that may occur when the meniscus is pinned at the top of the hydrophobic riblets, creating a superhydrophobic surface with potential for enormous slip. This type of state is highly susceptible to small pressure variations, which can lead to wetting or bubble formation or both. Figure 3.3 (D) illustrates partial riblet wetting and (E) illustrates full riblet wetting. These scenarios create a riblet type surface with low-friction interfaces near the wall. This state is expected to have better performance than riblets alone due to the slip interfaces. Figure 3.3 (F) shows complete surface wetting. Given the loss in slip surfaces this state is expected to perform like standard riblets. Figure 3.3 (G) shows a surface with a non-uniform wetting condition which contains a mixture of several wetting states and the possible presence of bubbles. This state is difficult to deal with analytically. Because of the small pressure necessary...
Figure 3.4: Side view of hybrid surfaces in a fully superhydrophobic state for (A) the final wafer which above which the flow is imaged and (B) the initial wafer in the development section over the test surfaces.

to initiate wetting, partial wetting will begin but not necessarily continue. The contraction of the plastron due to wetting leaves unwetted cavity regions and bubbles at higher pressures that are more stable.

When a test begins, the channel is gradually filled with water in a low pressure state. As the water moves along the channel it is suspended on top of the riblets in a superhydrophobic state. Figure 3.4 (A) and (B) are pictures taken through the channel sidewall of the superhydrophobic state prior to the first hybrid test run. The mirror finish is present on the surface, indicating the largescale riblets are in the Cassie-Baxter state. Notably the glassy surface appears to have some texture, not acting as a perfect mirror. This in due to the uneven SU-8 height due to the processing method, rather than an indicator of wetting. This superhydrophobic state on top of the riblets can be maintained indefinitely in a very narrow pressure window. Figure 3.4 (A) shows the last test wafer in the channel. The PIV measurements are taken over the center of this wafer. Figure 3.4 (B) shows the first wafer in the test section. A bubble is seen attached to this surface, as can easily occur. Once the turbulent flow in the channel began, the bubble translated downstream and was gradually removed from the surface.

When the surface is immersed in water, different wetting conditions can occur. Figure 3.5 shows immersed test surfaces in the assembled channel. Wetting is observed over part of the surface due to an increase in static pressure. Figure 3.5 (A) shows the initial wetting with less than 1 kPa of pressure. The riblets in the surface regions which appear dark are clearly wetting,
Figure 3.5: Flooding of the final wafer due to a strong pressure spike. (A) Shows initial wetting for a pressure of less than 1 kPa, and (B) and (C) show top and side view of wetting resulting from a large pressure increase (more than 5 kPa).

but it is unclear if the superhydrophobic microribs below are wetting. In an effort to achieve a condition of fully wetted riblets the pressure was increased to 5 kPa, above the Laplace pressure for the microribs. Figure 3.5 (B) and (C) show the additional wetting that occurred at the higher pressure. No further wetting occurred at this elevated pressure. This seems to indicate that the remaining superhydrophobic riblet regions may have absorbed the air from the rest of the surface to match the pressure across the meniscus, effectively creating bubbles between the riblets.

A different wetting phenomenon was observed when the pressure over the surface was maintained in the narrow pressure bounds of 0-1 kPa in turbulent flow. This can be seen in Fig. 3.6, which shows before and after results of the third hybrid surface test. Before flow begins in Fig. 3.6
(A) and (B) the water is suspended on top of the superhydrophobic riblets. There is a reflective plastron visible in Fig. 3.6 (A) and trace deposits of particle seeding visible in Fig. 3.6 (B). After running through the flow rates of the test, noticeable wetting has occurred. Wetting locations can be seen in the sharp contrasting patches of Fig. 3.6 (C). The darker regions are indicators of wetting whereas the lighter portions reflect greater light due to a bulging meniscus and disconnected plastron regions. Figure 3.6 (D) shows white streaks along the surface. These streaks appear to be a buildup of seeding particles where the riblets are behaving in a superhydrophobic manner. The particle streaks are the most dense at the downstream edge of the plastron, which may be explained by a recirculation or slowing due to a sudden reduction in slip on top of the riblets, which would facilitating particle settling.

Additionally, air bubble shedding was observed during the test using PIV in the region near the wall of $y < 300$ µm. Typically a bubble would gradually emerge from the meniscus. Under the turbulent conditions the bubble would oscillate and become more distinct. Lastly the bubble breaks free of the surface and is carried down the channel, setting up conditions for another bubble to form, contributing to a gradual loss in air from the cavities.

It is most interesting that, despite the varied surface wetting conditions, the PIV results for each $Re$ and even between tests were very similar and showed significant flow effects and drag reduction (as will be seen hereafter). Perhaps the slip benefits are greater than the detriment of the mixed wetting condition. More work with hydrophobic riblets and large module width superhydrophobic surfaces may shed more light on this phenomenon. Additionally, in further work it would be interesting to test hybrid surfaces at vacuum pressures or with riblet wetting but not microrib wetting (possibly achieved through pressure pulsing while filling the channel).

### 3.2 PIV Quantitative Results

The discussion will now turn to the quantitative results acquired from the PIV measurements. Hereafter the SH will refer to the superhydrophobic microrib surfaces. The steady and fluctuating velocity profiles in the channel will be discussed as well as any other related changes in the flow field. Then the shear stress will be discussed and drag reduction will be reported. Lastly, the turbulent kinetic energy production will be discussed.
3.2.1 Channel Velocity Profile

The smooth control surface velocity profiles obtained from the measurements will be considered first. Fig. 3.7 shows the profiles of $U$, the normalized time-averaged velocity across the channel for several Reynolds numbers. The profiles progress from the familiar parabolic laminar profile at $Re = 1800$ (black line and square markers) and become increasingly more turbulent. As $Re$ increases the profiles become flatter across in the center with lower maximum velocities and develop steeper velocity gradients near the wall. This behavior matches the expected classical turbulent behavior and verifies that the screen set up to trip turbulence has been successful, even at low $Re$. Additionally, all turbulent velocity profiles for the control surfaces follow the law of the wall that was presented earlier in Fig. 1.1.

Having verified the flow is fully turbulent and the PIV setup is correctly measuring average velocities, the effect of each surface type on the average velocity profile across the channel can be
Figure 3.7: Normalized $\eta$ profiles plotted across the channel for various select Reynolds numbers for the control case.

The velocity profiles for the four surface types are shown in Fig. 3.8 for a representative case of $Re = 12.3 \times 10^4$. Additionally, there is an inset which shows a more magnified velocity profile over half the channel. The superhydrophobic (blue dots) and control surfaces (black line) track identically indicating that the superhydrophobic surface has minimal effect on the average streamwise velocity. The riblet (green triangles) and hybrid surfaces (red squares) profiles show that there is a distinct shift in the flow profile as a noticeable portion of the flow has shifted toward the riblet surface, causing the point of maximum velocity to move towards the test surface, indicative of a greater slip or lower turbulence over the hybrid surfaces. This shift is more pronounced with the hybrid surface.

The magnitude of the velocity profile shift was evaluated by finding the increase in volumetric flow rate on the lower half of the channel as a percentage of the total flow volumetric flow.
Figure 3.8: Normalized velocity profiles, $U$, plotted across the channel for the four test surfaces for $Re = 1.22 \times 10^4$. This quantity is called the percent flow shift, $\%FS$. The results are plotted in Fig. 3.9. All raw data is shown in Fig. 3.9 (A) in order to show the number of data sets involved and their spread, where each point represents the PIV results for 4000 image pairs. Hereafter figures will show the mean values and standard error of the mean as in Fig. 3.9 (B). The control surfaces (black squares) have no significant flow shift, as expected. The SH surfaces (blue dots) also have no significant flow shift, reflecting the symmetric velocity profile seen in Fig. 3.8. The scatter of the control and SH data indicates the amount of variation in a single measurement of $\%FS$ is up to 0.1%. The riblet surfaces (green triangles) begin with a negative $\%FS$, indicative of an increase in friction at low $Re$ and show an increasing shift near their ideal operating range at $s^+ \approx 16$ which occurs at $Re = 1.6 \times 10^4$, where riblets spacing optimally damps the turbulence formation, giving a maximum shift of 0.2%. The hybrid surfaces (red x’s) show a similar, but much larger $\%FS$, indicating more
Figure 3.9: The percent of the volumetric flow rate shifted to the test surface half of the channel plotted against $Re$. 
substantial drag reduction than the riblet surfaces, but following the same general shape, with a maximum shift of 1%. The shift in flow correlates well with drag reduction, which is reasonable, considering that a shift in the flow towards one wall represents less resistance to flow in a region.

Figure 3.10: $Y$ location of maximum velocity, $U_{max}$, as a function of $Re$ for each surface type.

Another metric of the shift of the average velocity profile is the location of the normalized velocity maximum, $U_{max}$. Figure 3.10 shows $U_{max}$ location as a function of $Re$ for each of the surface types. The control (open squares) and superhydrophoic (shaded circles) surfaces occur right on $Y = 0.5$, and thus directly in the center of the channel, indicating a symmetric velocity profile. The velocity maximum for riblet surfaces (green triangles) occurs in the channel center for laminar flows, but as $Re$ approaches $1.6 \times 10^4$ a maximum $U_{max}$ shift occurs at $Y = 0.484$, a 1.6% shift towards the riblet surface. The hybrid surface results (red x’s) show a more substantial shift in the $U_{max}$ location with a max shift to $Y = 0.464$, a 3.6% shift towards the hybrid surface. The shift in $U_{max}$ of the hybrid surfaces has the same $Re$ dependent behavior as the riblet surfaces, but more
Figure 3.11: The normalized $U_{rms}$ profiles across the channel for the control surfaces.

exaggerated. Additionally, there appears to be a much larger variation in the hybrid tests, which is most likely a response to different wetting conditions on the surfaces.

### 3.2.2 $U_{RMS}$ Velocity Profile

$U_{rms}$ is the normalized root mean squared fluctuating streamwise velocity computed as $U_{rms} = \sqrt{\bar{u'^2}/\bar{u_{avg}}}$.

It is a measure of the fluctuations in the streamwise velocity in the channel and an indicator the overall turbulence for a given location. Several profiles for the smooth surface control data are shown in Fig. 3.11 to illustrate the effect of $Re$ on $U_{rms}$. Initially, though not shown because of low resolution near the wall, there is a steep linear slope from $y^+ = 0.1$ to $y^+ = 5$ [14]. $U_{rms}$ increases to a maximum at $y^+ \approx 15$. This is true for all surface types and all $Re$. Then $U_{rms}$ decreases from this maximum down to a minimum near the center of the channel. At lower $Re$ the
values between the peaks appear parabolic in nature (black squares, blue triangles), but this is not true at higher Re where the profile descends from a sharp peak, levels off, then drops again to a minimum in the channel center (green pluses, orange diamonds). The leveling off of the profile is expected and is related to the turbulent transport of kinetic energy. The overall drop in the center indicates a more steady velocity in the channel center and dissipation of the turbulence generated near the wall as it moves toward the channel center.

Figure 3.12 shows the $U_{rms}$ profiles for each surface type at $Re = 1.35 \times 10^4$. The plot fails to capture the $U_{rms}$ behavior near the wall, which is generally true for $Re > 1.0 \times 10^4$. The control (black line) and SH (blue dots) data follow nearly the same profile, indicating that the SH surface has little effect on $U_{rms}$. The riblet (green triangle) and hybrid (red squares) data both show a shift in the local minimum. The shift in the local minimum is the same shift as the location of the

Figure 3.12: The normalized $U_{RMS}$ profile across the channel for each test surface for $Re = 1.22 \times 10^4$. 
velocity maximum that was presented in Fig. 3.10. Additionally, the riblet data (green triangle) show a reduction in $U_{rms}$ in the half of the channel nearest the riblet test surface, indicating that the riblet surface has reduced the amount of turbulence generated on that side of the channel. The hybrid data (red squares) shows even lower velocity fluctuation on the test half of the channel than the riblet data, indicating a substantial reduction in turbulence. The reduction is more than 10% at $Re > 1.0 \times 10^4$. The hybrid surface profile does show a small increase in $U_{rms}$ near the smooth wall. It is unclear what could be causing this effect, although it is likely related to the shift in the average flow velocity away from that wall.

Another indicator of the surface effects is the average value of $U_{rms}$. The average streamwise RMS fluctuating velocity, $U_{rms}^*$, is computed as $U_{rms}^* = \frac{1}{2} \int_0^{0.5} U_{rms} dY$. Figure 3.13 shows $U_{rms}^*$ plotted against $Re$ for each of the surfaces. All the profiles rapidly decrease with $Re$ and then level off a little after $Re > 1.0 \times 10^4$. The change in behavior most likely indicates a failure to capture some of the small velocity fluctuations due to the PIV setup. The control (black squares) and SH
(blue circles) results again closely match one another. The riblet data (green triangles) show a reduction in $U_{rms}^*$ for all $Re$, indicating that the riblets are reducing the streamwise turbulent fluctuations. The reduction is smaller at low $Re$, close to 5%, but it become more pronounced for higher $Re$ where a 13% reduction is observed. The hybrid data (red x’s) shows a slightly larger reduction than the riblets for all $Re$, indicating a slight reduction in the streamwise fluctuations. At $Re \approx 3.5 \times 10^3$ the hybrid data are very low and the SH data are reduced. At this point the channel is transitional with reduced fluctuation near the walls and low level isotropic fluctuations are occurring in the channel center for $0.3 < Y < 0.7$.

### 3.2.3 $V_{rms}$ Velocity Profile

$V_{rms}$ is the normalized RMS fluctuating wall-normal velocity computed as $V_{rms} = \sqrt{\langle v'^2 \rangle / \bar{u}_{avg}}$. It is a measure of the turbulent transport normal to the wall. Figure 3.14 shows $V_{rms}$ plotted across the channel for the control surfaces at several $Re$. In Fig. 3.14 (A) $V_{rms}$ profiles are shown for several $Re$ less than $1.0 \times 10^4$. For the lowest $Re$ explored, $Re = 3.5 \times 10^3$, the profile rises a plateau in the channel center indicating the flow may still be developing. As $Re$ increases the normalized center velocity fluctuation is no longer a function of $Re$, becoming constant at $V_{rms} = 0.045$. However, nearer to the walls distinct local maximums develop at $y^+ \approx 55$ that become higher with increasing $Re$. The wall-normal component of turbulence appears to be rapidly increasing near the wall for $Re < 1.0 \times 10^4$.

Figure 3.14 (B) shows $V_{rms}$ profiles for $Re > 1.0 \times 10^4$. There is a distinct change in behavior from the lower $Re$, in that the $V_{rms}$ maximum values near the wall are no longer increasing with $Re$. This change in behavior is most likely due to the PIV setup, which will average velocity fluctuations over the laser plane as the velocity scales become smaller. The locations of the maximum regions do continue to move closer to the wall with increasing $Re$ and the peak still located in the same relative location in wall units at $y^+ \approx 55$. The channel center region ($0.2 < Y < 0.8$) shows a small decrease in $V_{rms}$ throughout and appears to have a slightly larger radius of curvature; however, the profile as a whole shows little change with $Re$ for $Re > 1.0 \times 10^4$. Rather than the more complex behavior of $U_{rms}$, the profile is similar, with an increase in turbulence near the wall and then a slight decrease as dissipation occurs nearer the center. In contrast to the $U_{rms}$ profile, the $V_{rms}$ profile has maximums further from the wall, indicating that the streamwise fluctuations
Figure 3.14: $V_{rms}$ plotted across the channel for the control data. Part (A) shows the profiles for $Re < 1.0 \times 10^4$ and part (B) shows the behavior for $Re > 1.0 \times 10^4$.

occur nearer to the wall and later develop into wall-normal fluctuations. At $Re$ increases the RMS velocities near the center of the channel, for both $U_{rms}$ and $V_{rms}$, approach each other, indicating a region of more homogeneous turbulence.
Figure 3.15: The normalized $V_{rms}$ profile across the channel for each test surface for $Re = 1.22 \times 10^4$.

The $V_{rms}$ profiles, averaged over all tests, for the four surface types are shown in Fig. 3.15 for the representative case of $Re = 1.35 \times 10^4$. The SH data (blue dots) show $V_{rms}$ values greater than the control data (black line) over most of the channel. This increase is most pronounced at the maximum near the smooth wall at $Y = 0.85$. The increase is consistent with an increase in the wall-normal turbulent fluctuations of the channel. It is possible that this increased fluctuation across the channel is derived from the elastic nature of the water-air boundary of the superhydrophobic test surfaces, perhaps explaining the increase in friction on the wall opposite the test surface. The riblet data (green triangles) are markedly lower for $Y < 0.5$ and appear to line up with the control data well near the smooth wall. Again, a shift in can be seen, similar to that of the overall velocity profile $U$, evidenced by the shifted center minimum and the shifted maximum near the smooth wall. This is attributed to a reduction in the vorticity generation over the riblet wall, due to the reduction of spanwise ejection events by limiting streamwise vortice translation and due to a shift of the shear
stress. The hybrid data (red squares) behavior reflects characteristics of both the superhydrophobic and riblet profiles. There is a distinct shift in the center minimum, a strong decrease in $V_{rms}$ for $Y < 0.5$, and a significant increase near the smooth wall opposite the test surface. This indicates that the hybrid surfaces reduced the turbulence generation to an even greater degree than for a plain riblet surface. There is an increase in wall normal fluctuations near the smooth wall, indicating an increase in fluctuation on the wall opposite the hybrid surface. This may be caused by the shifted flow maximum or perhaps by the superhydrophobic characteristics of the surface.

As before, $V_{rms}$ can be integrated across half the channel as $V_{rms}^* = \frac{1}{2} \int_0^{0.5} V_{rms} dY$ to find the average wall-normal rms fluctuating velocity. $V_{rms}^*$ is shown in Fig. 3.16 against $Re$ for each of the surface types. As observed in the profiles there is a substantial increase in $V_{rms}^*$ with $Re$ up to $Re \approx 1.0\times10^4$, at which point the $V_{rms}^*$ levels off. At higher $Re$ there is a slight decrease in the average rms wall-normal velocity fluctuation. The superhydrophobic data (blue circles) track closely with the control data (black squares), showing little difference between the two surface types when $V_{rms}$
is averaged. The riblet (green triangle) and hybrid data (red x’s) have nearly identical behavior and show a relatively constant and substantial reduction in $V_{rms}^*$ for $Re > 4.0 \times 10^3$, indicating an overall reduction in turbulent motion throughout.

### 3.2.4 Shear Stress Profiles

Since one of the aims of this research is to measure the drag reduction, which is computed from the total shear stress at the wall, the shear stress profiles across the channel are next considered. First the total shear stress profile will be presented and then comparisons will be made between surface types. Dimensional shear stresses are notated using $\tau$. Shear stresses normalized by dynamic pressure, $\rho \bar{u}^2$ are notated with $T$. Shear stresses normalized by the wall shear stress, $\tau_{wall}$ are notated with $\tau^+$.

Figure 3.17 shows $\tau_{tot}^+$ plotted against $y^+$ from the test surface to the channel center for the control surfaces. The wall-normalized shear stress, defined as $\tau_{tot}^+ = \tau_{tot} / \tau_{wall}$, shows the fraction of the total shear stress to the wall shear stress. $\tau_{tot}^+$ ranges from 1 at the wall ($y^+ = 0$) to 0 at the channel center ($y^+ = Re \tau$), for the case of a symmetric channel profile. Here the actual profiles obtained are shown with markers and the expected linear profiles are shown as lines. For the low $Re$ case (black squares), the shear stress is linear up to $\tau_{tot}^+ = 0.8$ corresponding to $Y = 0.1$. After the very clear linear portion, smaller structures begin to occur in the region near the wall, corresponding to $y^+ < 20$. This is the region where the velocity fluctuations become smaller, causing the PIV measurements to underestimate $u'v'$ as a result of averaging across the laser width. The underestimated turbulent stress strongly affects $\tau_{turb}$, causing the data to be reduced from the expected linear shear stress profile in the near wall region. This effect becomes more dramatic at higher $Re$ where smaller turbulent structures form, such that shear stress data for $Y < 0.85$ (corresponding to $\tau^+ > 0.7$) is not considered valid.

Figure 3.18 shows the averaged shear stress profiles for all the tests across the channel for each of the surface types at $Re = 1.56 \times 10^4$. Assuming the profiles are fully developed, the shear stress should be linear, going from the positive, normalized $\tau_w$ at the test wall to negative at the smooth wall. There are non-linear regions for all profiles near the wall ($Y < 0.25$ and $Y > 0.75$), where the total shear stress in underpredicted, confirming that the non-linearity observed in the control surfaces applies to the other surfaces as well. The control (black squares) and SH (blue
Figure 3.17: Actual profiles and expected profiles of $\tau_{+\text{tot}}$ plotted against $y^+$ for the control case at several Re.

dots) profiles pass though $T_{\text{tot}} = 0$ at the center line ($Y = 0.5$), as is expected for channel flows. The slope of the SH profile is steeper with the shear stress passing through zero in the channel center, indicating an increase in friction at both walls. The riblet (green triangle) and hybrid (red x's) profiles both run nearly parallel to the control surfaces, indicating comparable levels of total friction; however, the profiles are distinctly shifted. This shift indicates lower shear stress near the test surface and an increase in friction on the smooth surface. This behavior was not entirely anticipated, as it was thought the smooth wall shear stress would remain constant.

It may be helpful to consider this behavior in terms of two different half channels, with average velocities and channel sizes that extend from the wall to the zero shear stress point. This additional analysis will be undertaken as time permits.
Figure 3.18: $T_{tot}$ plotted against $Y$ for each surface type at $Re = 1.56 \times 10^4$

The linear region of $T_{tot}$ in the channel center plays a large role in determining the friction factor, since the near-wall shear data is not reliable. Figure 3.19 shows an enlarged view of Fig. 3.18 for the region of $0.25 < Y < 0.75$. In addition to the shear stress profiles (markers as above), the linear fits to the $T_{tot}$ are shown as lines. Next to the lines one can see the non-linearity of the shear stress profile even applies in the center region, although to a much lesser extent. The linear fit to the SH profile (blue dotted line) is clearly much steeper than the other lines, which run approximately parallel each other, indicating a uniform increase in wall shear stress for both the test and smooth surface.
Figure 3.19: A closer look at the linear region of $T_{tot}$ from Fig. 3.18, plotted across the middle region of the channel from $0.25 < Y < 0.75$ for each surface type at $Re = 1.56 \times 10^4$. Lines show the linear fit to each profile.

### 3.2.5 Friction Factors and Drag Reduction

Measurement of the friction on the walls can be made based on the total shear stress profile. The wall shear stress is determined by applying a least squares fit to the total shear stress and extrapolating to the wall. The worst non-linear sections for $Y < 0.15$ and $Y > 0.85$ were not used in the friction factor computation. The friction factor is related to the wall shear stress by Eqn. 3.1.

$$f = \frac{8\tau_w}{\rho U^2} = 8T_{tot}|_{wall} \tag{3.1}$$

Figure 3.20 shows averaged friction factors for the control surfaces (shown as black circles) plotted against the $Re$. The 95% confidence intervals for the mean values are shown as errorbars.
Figure 3.20: The averaged friction factors for the control surfaces plotted against $Re$. (- -) is the correlation proposed by Beavers et al. and (blue line) is the classic Colebrook equation for turbulent flow in a smooth pipe.

The correlations proposed by Colebrook (blue solid line) and Beavers et al. (black dashed line) are also shown [16, 17]. The Colebrook correlation [16] is for pipe flow and is corrected for a rectangular channel by the hydraulic diameter and the Beavers et al. correlation [17] is for low $Re$ turbulent flow in large aspect ratio rectangular ducts. The control data fall within the expected range and clearly follow the slope of Beavers correlation, just slightly below. This makes sense, considering the non-linear shear stress behavior. The data seem to line up very well. There is greater error in lower $Re$ measurements as expected from the error analysis results that were presented in Table 2.6.

The averaged friction factor results for all the test surfaces are presented in Fig. 3.21. For the SH surfaces (blue circle), $f$ is substantially higher for nearly every case, with the exception of
Figure 3.21: The averaged friction factors for the control surfaces plotted against $Re$.

$Re = 3.5 \times 10^3$, where the flow where the flow is transitional and not fully turbulent. This increase in $f$ means that the surfaces are actually adding more resistance, beyond that of a smooth surface, and not producing the expected drag reduction. This could a result of the breaker ridges or indicative of some degree of wetting. The riblet surfaces (green triangles) exhibit behavior in line with what was expected. At $Re = 1.0 \times 10^4$ there appears to be an increase in friction. Then as the turbulence scales near the surface decrease, toward the riblet design case, a drag reduction is observed. Once $Re$ exceeds $2.2 \times 10^4$ the riblets once again lose their drag reduction capabilities. There are two points below $Re = 1.0 \times 10^4$ that also show substantial drag reduction. The cause is unknown, but could be either measurement error or possibly a harmonic of the ideal riblet size, since the spacing is approximately half of the optimal spacing, where $s^+ \approx 7.5$. The hybrid surfaces (red x’s) perform like the riblets, but show even greater drag reduction, especially for $Re > 1.0 \times 10^4$. In
Table 3.1: Friction factors on the test surface and opposite smooth wall for the different surface types as a function of flow rate.

<table>
<thead>
<tr>
<th>Flow Rate (LPM)</th>
<th>Reynolds Number</th>
<th>Control Avg. ( f )</th>
<th>SHS Test ( f )</th>
<th>Riblet Test ( f )</th>
<th>Hybrid Test ( f )</th>
<th>Control Top ( f )</th>
<th>SHS Top ( f )</th>
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<td>–</td>
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</table>

In most cases the drag reduction relative to the control surfaces is twice as much. There is a single point that drops very low at \( Re = 3.5 \times 10^3 \), where the flow was shown earlier to be in a developing or relaminarization state. The point immediately after it shows no drag reduction. Indicating that the surface is not providing a friction reduction.

The numeric values of \( f \) for the test surface and smooth top wall for each surface type are listed in Table 3.1. They are sorted by the nominal flow rate and resulting average \( Re \) of the channel. Data that was not taken is indicated by (–). "Avg." indicates a the average of the control top and bottom surfaces, "Test" indicates the test surface, and "Top" indicates smooth wall opposite the test surface. Interestingly, there is a strong increase in \( f \) for the smooth top wall, values shown in the last three columns. This indicates the presence of a special test surface alters the friction on the wall opposite the test surface, causing an increase in friction.

The SH surfaces warrant additional investigation because of the unexpected results. To that end, Fig. 3.22 shows the individual results for the superhydrophobic surfaces taken during different data runs. The S# refers the manufactured set of surfaces (two different sets were tested). The A# refers for a given assembly of the test channel, surfaces that were re-used were re-coated between assemblies. The data appear widely scattered, although generally still show an increase in friction. A notable exception is S1 A1 (red triangles), the first assembly with the first set of superhydrophobic surfaces, for which extreme results were obtained, the largest drag increase and...
Figure 3.22: The friction factors for the SH surfaces plotted against $Re$. S# refers to the set of surfaces and A# refers to occasion the surfaces were coated and assembled in the channel.

the largest drag reductions. In the case of drag increase, notable wetting and bubble formation were observed on the surface. It is not clear why the large drag reduction occurred during the second test run on that assembly. These were some of the earlier tests it is possible that the PIV setup was not working properly, averaging over a greater channel region and further under-predicting shear stress more. Regardless of the cause, the results were not able to be duplicated in later runs. The fully wetting cases for S1 (gray crosses) and S2 (black crosses) are also worth noting. These were runs where the set of SH surfaces was intentionally flooded by running the flow over the surfaces at high pressures and flow rates for a long duration, such that the surfaces appeared completely flooded in the cavity region. These sets of data track well with most of the data, indicating that generally SH surfaces showed no benefit over the wetted surfaces.
Figure 3.23 shows $f$ for both the bottom riblet wall (hollow shapes) and the smooth top wall (x’s) of the same test plotted against $Re$ for each riblet run (set of tests run on the same day). The control data are also included (black circles). Each run seems to follow the same shape and have nearly the same spacing between the riblet wall and the smooth wall. However for some reason Run 1 (red triangles and x’s) appears to have high friction factors and Run 3 (blue squares and x’s) appears to have very low friction factors. This may be related to the temperature measurement of the surfaces, since Run 1 (red markers) generally shows lower $Re$ at each location, which would indicate a higher viscosity causing higher shear stress results due to an underpredicted temperature reading. The opposite would be true with Run 3 (blue markers), where a higher $Re$ reading indicates lower viscosity causing lower shear stress from an overpredicted temperature.
With better thermocouple measurements this could have likely been avoided to some degree, but still does not account for the substantial total shear stress reduction near the wall.

Examining the friction factor profiles further, Fig. 3.24 shows the test surface and smooth top wall \( f \) for both the riblets and hybrid surfaces plotted against \( Re \). For most of the operating range the hybrid test surfaces (red squares) show a large decrease in \( f \), especially in the optimal riblet range, whereas hybrid smooth surfaces (red pluses) show a substantial increase in friction. This agrees well with the strong decrease in \( V_{rms} \) near the test wall and increase in \( V_{rms} \) near the top wall, but from drag reduction standpoint is somewhat unexpected. Since the total shear stress profiles for the control, riblet, and hybrid surfaces in Fig. 3.19 run nearly parallel, whenever there is a decrease one wall, the other wall compensates with increased friction, related to the flow shift. The riblet (green triangles and x’s) \( f \) behavior is nearly identical to the hybrid behavior, except it
Figure 3.25: The raw friction factors for hybrid surfaces plotted against $Re$ on a log plot for both the test surface on the bottom wall (squares) and the smooth top wall (pluses).

is closer to the control data and less extreme on either wall. The low hybrid point at $Re = 3.5 \times 10^3$ is transitional, as was seen in earlier sections.

As a confirmation that the PIV setup is working and to verify the reality of the measurements, laminar measurements were taken that verify the viscous stress was being correctly determined. Figure 3.25 shows the riblet data plotted on a log plot. The laminar $f$ for a two-dimensional channel (blue dotted line) descends at a steep rate from the top left corner following the curve, shown in Eqn. 3.2

$$f = \frac{96}{Re}$$ (3.2)
Figure 3.26: The percent change in drag plotted against $s^+$ for the riblet (open triangle) and hybrid data (open square). The data obtained by Prince and by Bechert et al. are included for comparison.

The hybrid $f$, for both top and bottom surface in the laminar region beginning at $Re = 720$, match the expected values to a high degree of accuracy. Then around $Re = 3.5 \times 10^3$ the flow transitions into the turbulent regime, where the hybrid surfaces show a significant reduction in $f$ on the test surface and increase in $f$ at the smooth top wall.

The percent drag reduction of the test surfaces relative to the control surface, $DR$, can be computed as shown in Eqn. 3.3

$$DR = \frac{f_{\text{control}} - f_{\text{test}}}{f_{\text{control}}} \times 100$$

where $f_{\text{control}}$ is the averaged friction factor for the control surface and $f_{\text{test}}$ is the friction factor for the test surface. $DR$ for the riblet and hybrid surfaces, plotted as % change in friction
Table 3.2: *DR* of the test surface relative to the control and *SD* for the different surface types as a function of flow rate.

<table>
<thead>
<tr>
<th>Flow Rate (LPM)</th>
<th>Reynolds Number</th>
<th>$s^+$ Value</th>
<th>SHS %DR</th>
<th>Riblet %DR</th>
<th>Hybrid %DR</th>
<th>SHS %SD</th>
<th>Riblet %SD</th>
<th>Hybrid %SD</th>
</tr>
</thead>
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<tr>
<td>6</td>
<td>5500</td>
<td>6.1</td>
<td>-6.4</td>
<td>3.8</td>
<td>0.4</td>
<td>3.9</td>
<td>5.0</td>
<td>10.8</td>
</tr>
<tr>
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<td>7600</td>
<td>7.8</td>
<td>-7.5</td>
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<td>2.6</td>
<td>5.2</td>
<td>13.5</td>
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<tr>
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<td>9.6</td>
<td>-11.3</td>
<td>-0.4</td>
<td>0.2</td>
<td>2.5</td>
<td>6.0</td>
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<tr>
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<td>11.4</td>
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<td>0.7</td>
<td>5.9</td>
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<td>1.1</td>
<td>3.9</td>
<td>1.4</td>
<td>3.6</td>
<td>13.8</td>
</tr>
</tbody>
</table>

factor is shown in Fig. 3.26 plotted against $s^+$, for both this test and the tests of Prince [54] and Bechert et al. [74]. The riblet data (open green triangles) shows a strong increase in drag reduction as $s^+ \to 15$, then *DR* decreases as the riblets depart from optimal spacing. The riblet data appears to track the data obtained by Prince (solid blue squares) which had a comparable thickness to spacing ratio, $t/s$. The data by Bechert et al. [74] (purple circles, black diamond) showed greater drag reduction, which makes sense given the thinner blade geometry with a more optimal $t/s$ ratio. The hybrid data (open red squares) shows a strong increase in drag reduction at $s^+ = 13$ which gradually decreases down to the riblet range. It appears to outperform the riblet surface for all $s^+$. The *DR* results are summarized numerically in Table 3.2 where *DR* results are plotted for each of the surfaces against the flow rate, *Re*, and $s^+$ value. The superhydrophobic surface show no drag reduction for any flow rate. It is also valuable to examine the difference between friction factors of the test surface and the smooth top wall, since the data was taken simultaneously for the exact same flow rate and temperature. The table also includes this value, the percent surface difference, *SD*, which is defined in Eqn. 3.4

$$SD = \frac{f_{smooth} - f_{test}}{f_{control}} \times 100 \quad (3.4)$$

where $f_{test}$ is the friction factor for the test surface and $f_{smooth}$ is the friction factor for the smooth top surface of the channel during the same test. The *SD* is particularly good measure of surface performance, since the test surface relative to the simultaneous measurement of the opposite smooth
surface. $SD$ is always positive, indicating the test surface always had a lower friction factor than the opposite smooth surface.

This can be seen in Fig. 3.27, where $SD$ is plotted against $Re$ for each of the surface types with the standard error included. The control case (black squares) tracks close to zero, for $Re > 8.0 \times 10^3$, but there is an increase in $SD$ at low $Re$, which may be the a result of processing error. In general the error in the control data is much larger than the SH error, despite having a similar number of data points, indicating that some of the early control runs may not have been as accurate. The SH data (blue dots) is just above the control data, indicating a small increase in $SD$, meaning some drag reduction may be occurring. Most surprising is how small the error is, despite the surface variability seen earlier. This indicates the surfaces themselves perform fairly consistently relative to the smooth wall. The riblet surfaces (green triangles) show a more substantial difference between sides of the channel, that matches closely with the expected drag reduction of the surface, reaching a maximum of 7%. The error in the surface difference is small, despite having only four test runs, indicating very reliable results. $SD$ for the hybrid surfaces (red x’s) is the largest, as expected, showing a substantial variation between surfaces up to 16% at $Re = 1.32 \times 10^4$. The trend matches the riblet trend closely, with about twice the $SD$. The error in the hybrid measurements is small, considering only 3-4 data points are included, indicating consistent surface performance. There is an abrupt rise and dip in the data around $Re = 1.5 \times 10^4$ for all the data sets. It is unclear what may be causing this behavior.

### 3.2.6 Turbulence Production

One of the important equations for quantifying turbulence is the turbulent kinetic energy (TKE) equation, which governs production, transport, and dissipation of turbulent kinetic energy, $k$. For a two dimensional channel, the equation simplifies to the form shown in Eqn. 3.5 [14]

$$0 = -\frac{\langle u'v' \rangle}{\partial y} - \varepsilon - \frac{1}{\rho} \frac{d \langle p' \varepsilon \rangle}{\partial y} + \nu \frac{d^2 k}{d y^2} - \frac{d \langle v' u'^2 / 2 \rangle}{d y} \quad (3.5)$$

where $\varepsilon$ is the TKE dissipation, and $p'$ is a local pressure fluctuation. The terms on the right-hand side are the production, dissipation, pressure work, viscous transport and turbulent transport terms for $k$. Of particular interest are the production term (where TKE is created) and the dissipation
Figure 3.27: The friction factor difference between surfaces, $SD$, is plotted against $Re$ for each of the test surfaces.

term (where the TKE is destroyed). It is useful to examine their profile in wall units. $k$ production in wall units is defined in Eqn.3.6

$$P^+ = \frac{u'v'}{\nu} \frac{dv}{dy} \frac{u_\tau^4}{u_\tau^3}$$

and the dissipation of $k$ in wall units is defined in Eqn. 3.7.

$$\varepsilon^+ = \varepsilon \frac{v}{u_\tau^3}$$

Figure 3.28 shows the production and dissipation terms plotted in wall units for a typical channel flow at $Re_\tau = 590$. The data was from Bernard and Wallace [14] who reported results of a DNS simulation at $Re_\tau = 590$ by Moser et al. [90]. Turbulence production is zero at the wall and
Figure 3.28: Typical turbulent kinetic energy production and dissipation terms plotted in wall units from a DNS simulation at $Re_\tau = 590$

rapidly increases at edge of the viscous sublayer to be greatest in the buffer layer, near $y^+ = 12$. Since the turbulence production is a product of $u'v'$ and $d\bar{u}/dv$, the maximum value should occur according to Eqn. 3.8.

$$\tau_{visc} = \tau_{turb} = \frac{1}{2} \tau_{tot} \tag{3.8}$$

At higher $Re$ when $y^+$ is very near the wall, this equation can be normalized by the wall shear stress as shown in Eqn. 3.9.

$$\tau_{visc}^+ = \tau_{turb}^+ = \frac{1}{2} \tag{3.9}$$
Figure 3.29: Plots of the normalized turbulence production, $P$, across the channel for each of the test surfaces. The plots correspond to $Re$ as follows: (left) $Re = 1.04 \times 10^4$, (center) $Re = 1.41 \times 10^4$, and (right) $Re = 1.79 \times 10^4$.

TKE production in wall units, $P^+$, is simply the product of the wall viscous and turbulent shear stresses. This means that as the buffer layer becomes closer to the wall due to increasing $Re$, $P^+$ approaches 0.25 asymptotically, forming the upper bound of the turbulence production near $y^+ = 0.25$. In Fig. 3.28, $\varepsilon^+$ is at a maximum near the wall and decreases rapidly down to a plateau that begins at $y^+ \approx 8$. This plateau region occurs because of pressure work and viscous and turbulent transport of $k$. The profile then decreases tracking $P^+$ closely out from $y^+ \approx 40$ out into the channel center.

When the production term and the ratio of $P^+/\varepsilon^+$ are known, $\varepsilon$ can be computed along the channel, allowing one to compute the Kolmogorov length scales along the channel to give a sense of the size of the turbulent scales.

Turbulence production in wall units will be self-similar for all profiles. Although this is useful, the actual turbulence production highlights differences between surfaces. To preserve the
unique features of the TKE production at either of the non-uniform walls, the data will be normalized by the dynamic pressure [14], as shown in Eqn. 3.10.

\[ P = \frac{-\overline{u'v'}H \frac{dn}{dy}}{\overline{n^3}} \]  

(3.10)

In Figure 3.29, \( P \) is plotted for the control, superhydrophobic, riblet, and hybrid surfaces at three \( Re \). As \( Re \) increase, the turbulent production draws nearer to the wall and increases in magnitude. It should be noted that the turbulence production values are approximate. A negative consequence of the under-resolved velocity covariance, \( \overline{u'v'} \), is that the turbulent production peaks will scale with both the error in the turbulent stress and the friction factor. The superhydrophobic surfaces (blue dots) show an increase in \( P \), matching the increase in friction factor. The riblets (green triangles) show a significant reduction in turbulent production for \( 8.6 \times 10^3 < Re < 1.85 \times 10^4 \) \( (8 < s^+ < 18) \), indicative of the reduction of turbulence creation near the wall and combined with turbulent stress error. The interesting case is the hybrid surfaces (red squares) where the maximum value matches well with the control case, but the values are clearly lower for \( Y > 0.1 \). Since the hybrid surfaces exhibited the lowest friction factor of all the surfaces, the other surfaces must all under-report \( P \) to a greater degree.

In an effort to better understand the \( P \) results, the results from both sides of the channel are plotted together in Fig. 3.30 at \( Re = 1.57 \times 10^4 \). The test surface data is shown with dashed lines and the opposite smooth top surface data is shown with solid lines. The most dynamic range at \( Y < 0.2 \) will have low accuracy, as it is a product of the viscous and turbulent shear stress in the least accurately resolved region. Figure 3.30 (A) shows the control data (black) and the SH data (blue) across the channel. Interestingly, the control surfaces show a difference in turbulence production. This difference was observed at all \( Re \). It may be that the shear stress is less resolved near the test wall, due to camera angle and focus. The SH surface does not show the same difference, likely a result of the increase in \( f \) over the surface. Figure 3.30 (B) shows the riblet (green) and hybrid (red) data plotted with the control data (black). \( P \) of the riblets appears greatly reduced from the control data, but it still shows a similar gap. The low values are not the case for all \( Re \) and great variability is observed. The hybrid turbulence production data (red) are worthy of note. In most cases the hybrid production bounds the control data, with lower production over the test surface.
Figure 3.30: Plot of the normalized turbulence production, $P$, which both sides of the channel displayed, for the region $0 < Y < 0.5$ at $Re = 1.57 \times 10^4$.

and higher production near the smooth wall. The difference between the surfaces is more apparent, given the larger gap that is visible.

Given the significant variability in the region near the wall, a better region for comparison in nearer the channel center in the region $0.2 < Y < 0.5$, where the turbulent stress is fully resolved.
Figure 3.31: Plot of the normalized turbulence production, $P$, which both sides of the channel displayed, for the region $0.2 < Y < 0.5$ at $Re = 1.57 \times 10^4$.

In Figure 3.31 $P$ is shown for the surfaces for both the test surfaces (dashed lines) and opposite smooth surface (solid lines) plotted together. Figure 3.31 (A) shows $P$ for the control data (black) and SH data (blue). The control data clearly follow the same profile, indicating identical turbulence production on either side of the channel. The SH data both are higher than than the control data,
manifesting the drag increase in the form of increased turbulence. $P$ near the SH wall (dash-dotted line) is lower than the opposite smooth surface (solid line), showing the reduced friction on the test wall, relative to the smooth wall, that was seen in $SD$. Figure 3.31 (B) shows the $P$ for the riblet (green) and hybrid (red) data, with the control data is included for convenience (black dashed line). The results are similar to the friction factor data, with both the riblet and hybrid data bounding the control line. The riblet data shows a decrease in turbulence formation over the test surface, and an increase over the smooth opposite wall. $P$ for the hybrid surfaces is much lower over the test wall (red dotted line) and much greater near the opposite smooth wall (red solid line). Additionally, the zero shear stress point can be clearly seen for the hybrid surfaces at $Y = 0.465$, where the turbulence production goes to zero. The riblet zero shear stress point is less clear, but can be found at $Y = 0.485$. 
CHAPTER 4. CONCLUSIONS AND FUTURE WORK

4.1 Conclusions

The following section describes the conclusions that have been reached based on this research. The first conclusions relate to the observations of phenomena during testing. Superhydrophobic surfaces were shown to be more stable and less resistant to flooding with breaker ridges. When exposed to vacuum pressure a complicated bubble and flow interaction takes place, where air is extracted from the cavities and carried downstream by shear forces, eventually draining a surface of its plastron. The hybrid surfaces were shown to initially be superhydrophobic on top of the riblets. With a small applied pressure (<1 kPa), the hybrid surfaces would partially wet; however, once partial wetting occurred hybrid surfaces formed more concentrated and robust plastron/bubble features between the larger riblet structures in the remaining unwetted regions. Hybrid behavior wetting behavior is a strong function of the initial pressure and flow conditions.

PIV proved an excellent method of measuring the average velocity profiles. The SH and control surfaces were shown to have symmetric and nearly identical average velocity profiles. Riblet and hybrid surfaces, on the other hand, caused a distinct shift in the average flow towards the structured test surface, with the hybrid surface showing the greatest effect on the flow. The shift in flow caused by the riblet and hybrid surfaces was apparent in nearly every metric of surface behavior, i.e. $U, U_{rms}, V_{rms}, T_{tot}, f$, and can be thought of as the shift in zero shear stress location, or the effective channel center, in terms of turbulence.

Superhydrophobic surfaces behaved similar to the smooth control surfaces in terms of the streamwise average and fluctuating velocity profiles. In contrast, the shear stress and friction factor showed clear increase in friction. This is contrary to predicted results and may be a result of the breaker ridges. Additionally, significant variability between test runs was observed, the cause of which has not been fully determined.
It was observed that for this channel with a test surface on one wall, any decrease in friction over the test wall corresponded with an increase in friction on the smooth wall opposite the test surface, maintaining an equilibrium and causing the shift in zero stress location and velocity profiles. This behavior is logical, since a decrease in friction lowers the resistance to the flow allowing the zero-shear-stress maximum flow point to move nearer to the low-friction wall. This behavior was not entirely expected based on other PIV studies for similar surface types.

Significant error was present in the turbulent shear stress near the walls. This error propagates into the total shear stress. This was most likely caused by the imaging setup of the experiment. The laser plane illuminating the particles was comparable to 1/10 of the channel height, causing turbulent fluctuations to be averaged across the laser plane, resulting in an under-representation of the velocity fluctuations. This effect was more pronounced near the wall, where smaller structures structures form.

The riblet surfaces showed measurable drag reduction when compared with the control surfaces. The drag reduction followed the trends that other riblets have shown, having a maximum of 4.3% near $s^+ = 13$. The magnitude was a little lower than predicted. When comparing the difference between the test riblet surface and the opposite smooth wall for the same test, the corresponding drag reduction was on par with the expected riblet performance with a maximum of 7.4% at $s^+ = 13$.

The hybrid surfaces showed significant drag reduction relative to the control surfaces. Maximum drag reduction of 8.8% occurred at $s^+ = 13$. The drag reduction was larger than was expected for a blade riblets of this geometry, indicating some superhydrophobic behavior was present. When considering the difference between the test hybrid surface and the opposite smooth wall, a maximum drag reduction of 16.6% occurred at $s^+ = 13$, with comparable drag reduction in the region $9,500 < Re < 22,500$.

4.2 Future Work

A great deal of information has been presented and perhaps many more questions have been raised during the presentation of the data. It is the author’s opinion that there is a great more research that can be done towards PIV measurements in turbulent flow over these very unique surfaces.
In order for this research to be more effective there will need to be research done on surfaces with higher $F_c$ and larger $s$ surfaces. The current setup could even be used for additional surface types, but could be improved with a closed development length section and an improved test section designed for easier access and installation of the test surfaces. The method proposed by Shirtcliffe et al. [91] which uses identical test sections may be a good method of resolving the actual change in skin friction of the various surface types, since it gives a more direct measurement of the flow resistance whereas data from PIV and pressure drop measurements often show inconsistent friction factor results.

There is a need for further investigation of the interactions between the surface, the plastron, bubbles, and the turbulent liquid flow over the surface. A superhydrophobic surface in turbulent flow is subject to a complex set of motions, some of which are not fully understood. Numerical models do not consider this complex behavior, modeling only the resulting apparent slip boundary conditions. With better understanding of the surface scale effects, better design could be performed.

Currently there is no experimental work characterizing the wall friction as a function of the meniscus contact angle. There are several numerical works to this end, and some measure of agreement, but the effects of meniscus angle have not been measured directly in any experiments. Additionally there could be more investigation into the friction results of an immiscible liquid adhering to the micro-geometry of a hydrophilic surface and filling the cavities, which may be contrasted with the pressure-sensitive nature of the plastron air layer currently encountered.

There is also room to refine the manufacturing processes for multi-tier surface structures. That was a major shortcoming in this research, as the surfaces where highly susceptible to mechanical failure during manufacturing. Additionally so many steps were required to make the hybrid surfaces, that uniform surfaces are nearly impossible to achieve and manufacture became prohibitively difficult and time-consuming.

There is also room for improved hydrophobic coating methods. Coating techniques that generate nano-scale roughness would reduce wetting and improve corrosion resistance, leading to an overall improved surface. Coatings that do not require a metal base coat for adherence would have simplified the manufacturing process.
Finally there is room for more characterization of the wetting states that are encountered with a multi-tier superhydrophobic surface. Full investigation of the flow and pressure conditions that lead to given wetting behaviors could be useful and invaluable if more hybrid type superhydrophobic surfaces are pursued in the future.
REFERENCES


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APPENDIX A. SURFACE MANUFACTURE AND WAFER PROCESSING

A.1 Control Surface

- A virgin wafer is diced to the dimensions 79.8 mm x 46.4 mm using a dicing saw

A.2 Superhydrophobic Surface

- Rinse wafer with acetone and isopropanol and then place in oven for dehydration bake
- Spin AZ 2020 photoresist onto cooled wafer:
  1. 2750 rpm, 60 s, 300 rpm/s
- Softbake on hotplate: 1 min at 110°C
- Exposure on aligner: 80 mJ/cm² (soft contact, first mask, go longer w/filter)*
- Post Exposure Bake on hotplate: 1 min at 110°C
- Develop in AZ300 MIF developer: 1 min
- Rinse off excess developer
- Dry wafer with air
- Deep Reactive Ion Etch (DRIE) on the ICP RIE: SF₆ etch, C₄F₈ passivation, ≈ 60 cycles
- Place wafer in nanostrip to remove photoresist: 90°C for several hours
- Deposit chromium on wafer using the thermal evaporator: 3 rods
- Spin 0.2% Teflon AF solution in FC-75:
  1. 1000 rpm, 20 s, 300 rpm/s
- Bake on hotplate: ramp to 330°C then bake for 20 min
- Dice wafer to 79.8 mm x 46.4 mm using dicing saw
- Apply second coat of Teflon®AF as before

*Optional: may be exposed additional time with breaker ridge mask

A.3 Riblet Surface

- Rinse wafer with acetone and isopropanol and then place in oven for dehydration bake
- Spin Microchem Omnicoat™onto cooled wafer:
1. 500 rpm, 5 s, 100 rpm/s
2. 2250 rpm, 30 s, 300 rpm/s

- Bake on hotplate: 2 min at 200 C
- Spin Microchem Omnicoat onto cooled wafer as before
- Bake on hotplate as before
- Spin Microchem SU-8 2075 onto cooled wafer:
  1. 500 rpm, 5 s, 100 rpm/s
  2. 2250 rpm, 30 s, 300 rpm/s
  3. 6100 rpm, 2 s, max. acc.

- Soft bake on hotplate:
  1. 65 C for 5 min
  2. 95 C for 10 min
  3. Allow to cool for 30 min on hotplate
- Remove edgebead by placing wafer on spinner and using acetone-soaked "Q-tip"
- Exposure on aligner: 650 mJ/cm² (Soft Contact, First Mask)
- Post exposure bake on hotplate:
  1. 45 C for 5 min
  2. 55 C for 5 min
  3. 60 C for 60 min
  4. Allow to cool for 30 min on hotplate
- Place wafer in container of SU-8 developer and agitate: 10 min
- Place wafer in container isopropanol to rinse*
- Dry wafer with air
- Hardbake on hotplate:
  1. 60 C for 5 min
  2. 90 C for 5 min
  3. 120 C for 5 min
  4. 165 C for 5 min
  5. 200 C for 30 min
  6. Allow to cool for 40 min on hotplate
- Deposit aluminum on wafer using the thermal evaporator: 1 µm thick (3 boats with 3 pellets)
- Spin on SU-8, but stop when it reaches edges
  1. 500 rpm, 5 s, 100 rpm/s
  2. 2000 rpm, 30 s, 300 rpm/s
- Place wafer in sealed container with parafilm and bake at 55 C fro 90 min
- Spin off SU-8 using most recent spin settings
- Softbake on hotplate:
  1. 55 C for 5 min
2. 65 C for 5 min
3. 75 C for 5 min
4. 85 C for 5 min
5. 90 C for 10 min
6. Allow to cool for 30 min on hotplate

- Dice wafer to 79.8 mm x 46.4 mm using dicing saw
- Place wafer in SU-8 developer to remove SU-8 2025 photoresist from riblets: 5 min
- Rinse in bath of isopropanol (place in SU-8 for 2 minutes if it turns white)

*Note: if surface turns white return to SU-8 developer for 2 min then repeat this step as needed

A.4 Hybrid Surface

- Rinse wafer with acetone and isopropanol and then place in oven for dehydration bake
- Spin AZ 2020 photoresist onto cooled wafer: 2750 rpm, 60 s, 300 rpm/s
- Softbake on hotplate: 1 min at 110 C
- Exposure on aligner: 80 mJ/cm$^2$ (go longer w/filter)
- Post Exposure Bake on hotplate: 1 min at 110 C
- Develop in AZ300 MIF Developer: 1 min
- Rinse off excess developer
- Deep Reactive Ion Etch (DRIE) on the ICP RIE: SF$_6$ etch, C$_4$F$_8$ passivation, approx 60 cycles
- Place wafer in nanostrip to remove photoresist: 90 C for several hours
- Rinse wafer with acetone and isopropanol and then place in oven for dehydration bake
- Spin Microchem Omnicoat$^{TM}$ onto cooled wafer:
  1. 500 rpm, 5 s, 100 rpm/s
  2. 2250 rpm, 30 s, 300 rpm/s
- Bake on hotplate: 2 min at 200 C
- Spin Microchem Omnicoat onto cooled wafer as before
- Bake on hotplate as before
- Spin Microchem SU-8 2075 onto cooled wafer:
  1. 500 rpm, 5 s, 100 rpm/s
  2. 2250 rpm, 30 s, 300 rpm/s
  3. 6100 rpm, 2 s, max. acc.
- Soft bake on hotplate:
  1. 65 C for 5 min
  2. 95 C for 10 min
  3. Allow to cool for 30 min on hotplate
• Remove edgebead by placing wafer on spinner and using acetone-soaked ”Q-tip”
• Remove photoresist over alignment pattern
• Expose on aligner, lining up ”+” marks: 650 mJ/cm² (soft contact, 100 µm alignment gap)
• Post exposure bake on hotplate:
  1. 45 C for 5 min
  2. 55 C for 5 min
  3. 60 C fro 60 min
  4. Allow to cool for 30 min on hotplate
• Place wafer in container of SU-8 developer and agitate: 10 min
• Place wafer in container isopropanol to rinse*
• Dry wafer with air
• Hardbake on hotplate:
  1. 60 C for 5 min
  2. 90 C for 5 min
  3. 120 C for 5 min
  4. 165 C for 5 min
  5. 200 C for 30 min
  6. Allow to cool for 40 min on hotplate
• Deposit aluminum on wafer using the thermal evaporator: 1 µm thick (3 boats with 3 pellets)
• Deposit chromium on wafer using the thermal evaporator: 3 rods
• Spin on SU-8, but stop when it reaches edges
  1. 500 rpm, 5 s, 100 rpm/s
  2. 2000 rpm, 30 s, 300 rpm/s
• Place wafer in sealed container with parafilm and bake at 55 C fro 90 min
• Spin off SU-8 using most recent spin settings
• Softbake on hotplate:
  1. 55 C for 5 min
  2. 65 C for 5 min
  3. 75 C for 5 min
  4. 85 C for 5 min
  5. 90 C for 10 min
  6. Allow to cool for 30 min on hotplate
• Dice wafer to 79.8 mm x 46.4 mm using dicing saw
• Place wafer in SU-8 developer to remove SU-8 2025 photoresist from riblets: 5 min
• Rinse in bath of isopropanol (place in SU-8 for 2 minutes if it turns white)
• Spin 0.2% Teflon®AF solution in FC-75: 1000 rpm, 20 s, 300 rpm/s
• Bake on hotplate:
  1. 60 C for 5 min
  2. 90 C for 5 min
3. 120°C for 5 min
4. 165°C for 5 min
5. 200°C for 20 min
6. Allow to cool for 40 min on hotplate

- Apply second coat of Teflon AF as before

*Note: if surface turns white return to SU-8 developer for 2 min then repeat this step as needed
APPENDIX B. CHANNEL DIMENSIONAL DRAWINGS

B.1 Drawings

This section contains the black and white dimensional drawings for the four acrylic parts of the final channel. They are presented the following order: (1) Base Plate, (2) Top Plate, (3) Thin Spacer, (4) Thick Spacer (for camera optics).
The surface must have a fine surface finish and be milled using a sharp tool and cutting fluid to prevent damage due to heat/warping.

Leave factory finish on 3/16" sides (camera view window)

There are two spacers and they are identical.

The inner edge of the spacer needs to be clamped to lay straight when the holes are drilled. The distance to the holes should be guided off this edge. That way the inner dimensions of the channel are left unchanged by material variation.
The 0.708 STOCK dimension is not machined and is the default material thickness. This face must be kept smooth and unmarred and retain the factory finish.