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Thermal transport due to liquid jet impingement on superhydrophobic surfaces with isotropic slip

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ABSTRACT

This paper presents an analytical investigation of thermal transport due to a steady, laminar, axisymmetric liquid jet impinging normally on a superhydrophobic (SHPo) surface maintained at constant surface temperature. At the liquid-surface boundary of the spreading thin film, an isotropic hydrodynamic slip and temperature jump are imposed to approximate the SHPo surface boundary condition. Applying an integral analysis within the thin film results in a system of differential equations which are solved numerically to obtain local hydrodynamic and thermal boundary layer thicknesses, thin film height, and local and radially averaged heat flux. The classical smooth hydrophobic scenario with no-slip and no-temperature jump showed excellent agreement with previous differential analysis of the same problem. The influence of varying temperature jump length on the local Nusselt number was obtained over a range of Reynolds and Prandtl numbers. Increasing temperature jump length results in a dramatic decrease in the local thermal transport near the impingement point. The greatest decrease occurs at small temperature jump lengths. Further, local and average Nusselt numbers are less influenced by the Reynolds and Prandtl numbers as temperature jump length increases. Overall, variations in the temperature jump length exert much more influence than variations in the hydrodynamic slip length.

1. Introduction

Numerous studies have demonstrated drag reduction and self-cleaning behavior on superhydrophobic (SHPo) surfaces [1–7]. More recently, thermal transport physics have been explored as well. For a liquid flowing over a SHPo surface, thermal transport is inhibited due to insulating air-filled cavities at the wall. This has been observed in forced internal flow [8–11], natural convection in microchannels [12,13], and Marangoni convection in droplets [14]. Boiling on SHPo surfaces is inhibited primarily due to an early transition from nucleate to film boiling resulting from the higher liquid-solid contact angle and altered surface geometry [15,16]. Conversely, condensation heat transfer is enhanced due to increased vapor-surface contact area [17,18]. The focus of this paper is to explore the thermal transport for a liquid flow scenario which has not received attention: a laminar, liquid, axisymmetric jet impinging normally on a SHPo surface with isotropic slip.

The hydrodynamics of a jet impinging on a no-slip surface were modeled analytically by Watson [19]. These results specifically capture the development of the hydrodynamic boundary layer within the spreading thin film, the radial variation of thin film height, and estimate the hydrodynamic jump location. The model compares well with experimental data and has been improved to include surface tension in subsequent studies [20–22].

Thermal transport on a surface due to the spreading thin film of an impinging jet has also received significant prior attention. This interest has been motivated by industrial applications including cooling of electronic systems and quenching of metals and molded plastics [23,24]. Analytical models for scenarios of constant wall temperature [25], constant wall heat flux [26], and varying wall temperature or heat flux [27,28] have been accomplished. Additionally, numerical [29–33] and experimental studies [24,34,35] have been performed. In all studies, it was observed that the greatest thermal transport occurs near the impingement point and decreases asymptotically with increasing radial position. Additionally, the local and average Nusselt numbers increase with increasing Prandtl number and jet Reynolds number.

SHPo surfaces differ significantly from smooth hydrophobic or hydrophilic surfaces. They are created by combining micro/nano-scale roughness with hydrophobic surface chemistry. Due to the surface hydrophobicity, water does not wet the cavities between roughness features if the Laplace pressure is not exceeded. Random or patterned microfeatures are commonly employed. Important geometric parameters of SHPo surfaces with repeating features

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Nomenclature

a	jet radius
c	liquid specific heat
h	liquid film height
$\hat{h}$	normalized film height $\hat{h} = h/a$
$\hat{h}_0$	normalized liquid film height at end of region I
$\hat{h}_1$	normalized film height at end of region II
H	control volume height
k	liquid thermal conductivity
$\dot{m}$	mass flow rate leaving top control volume surface
Nu	Nusselt number $Nu = q''_w a\pi/(k(T_w - T_j))$
$\overline{Nu}$	radially-averaged Nusselt number
$\overline{Nu}_0$	average Nusselt number for no-slip and no temperature jump case
Pr	Prandtl number $Pr = \nu/\alpha$
Q	jet volumetric flow rate
q''_w	wall heat flux
r	radial coordinate
$\hat{r}$	normalized radial coordinate $\hat{r} = r/a$
$\hat{r}_0$	normalized radial coordinate at end of region I
$\hat{r}_1$	normalized radial coordinate at end of region II
Re	Reynolds number $Re = Q/(a\nu) = \pi aV/\nu$
T	local liquid temperature
T_{fs}	free surface temperature
T_j	initial jet temperature
T_w	wall temperature
u	radial velocity
$\hat{u}$	normalized velocity $\hat{u} = u/V$
U	free surface velocity
$\widehat{U}$	normalized free surface velocity $\widehat{U} = U/V$
V	jet velocity
z	axial coordinate
$\hat{z}$	normalized axial coordinate $\hat{z} = z/a$

Greek symbols

α	thermal diffusivity
δ	hydrodynamic boundary layer thickness
$\hat{\delta}$	normalized hydrodynamic boundary layer thickness $\hat{\delta} = \delta/a$
δ_T	thermal boundary layer thickness
$\hat{\delta}_T$	normalized thermal boundary layer thickness $\hat{\delta}_T = \delta_T/a$
$\hat{\delta}_{T0}$	normalized thermal boundary layer thickness at end of region I
Δr	control volume radial thickness
ΔT	wall temperature jump $\Delta T = T_w - T(r, z = 0)$
$\Delta\widehat{T}$	non-dimensional wall temperature jump $\Delta\widehat{T} = \Delta T/(T_w - T_j)$
θ	non-dimensional temperature $\theta = (T - T_w)/(T_j - T_w)$
θ_{fs}	non-dimensional free surface temperature $\theta_{fs} = (T_{fs} - T_w)/(T_j - T_w)$
λ	hydrodynamic slip length
$\hat{\lambda}$	normalized hydrodynamic slip length $\hat{\lambda} = \lambda/a$
λ_T	temperature jump length
$\hat{\lambda}_T$	normalized temperature jump length $\hat{\lambda}_T = \lambda_T/a$
ν	kinematic viscosity
ρ	fluid density

are the pitch, w, (the distance between microscale features) and cavity fraction, F_c (ratio of the cavity area projected onto the interface to the total interface area).

If the air-filled cavities are on the micron scale, the hydrodynamic and thermal boundary conditions at a surface can be significantly altered [17,36]. Liquid near the wall encounters a no-slip boundary condition at the liquid-solid microfeature interface and a nearly shear-free boundary condition at the liquid-air interface between microfeatures. For macroscopic flows, it is beneficial to define an aggregate slip accounting for the alternating slip and no-slip boundary conditions to obtain a uniform boundary condition. This condition allows definition of a local slip velocity, u_s, which is proportional to the wall shear stress, τ_w, and is given by, $u_s = \lambda\tau_w/\mu$, where μ is liquid dynamic viscosity, and λ is the hydrodynamic slip length [36]. λ is a property of the wall microstructure and can be defined in the Stokes flow regime by the surface parameters, cavity fraction and pitch [36].

The jet impingement problem on SHPo surfaces with isotropic [37] and anisotropic slip lengths [38] has previously been modeled analytically and validated experimentally [39]. Increasing λ at fixed Reynolds number was observed to have a similar effect on the hydrodynamics as increasing the Reynolds number. The hydrodynamic boundary layer developed more slowly, the film thickness decreased, and the hydrodynamic jump radius increased.

The composite liquid-solid and liquid-gas interface also alters the thermal boundary conditions. The liquid-solid interface yields a conventional convection boundary condition, while at the liquid-gas interface the heat flux is greatly reduced. Assuming metallic microfeatures (high thermal conductivity), the thermal conductivity of the gas filling each cavity is several orders of magnitude less than that of the microfeatures, rendering the liquid-gas interface nominally adiabatic. Again for macroscopic scale flows, an aggregate boundary condition can be defined in the form of a wall temperature jump, ΔT_w. This temperature jump is proportional to the wall heat flux, q''_w, as given by the relationship, $\Delta T_w = \lambda_T q''_w/k$ [11], where k is the liquid thermal conductivity and λ_T the temperature jump length. The temperature jump length is the thermal analog to the hydrodynamic slip length [9,11,40] and is a function of microscale feature geometry.

The relationship between λ_T and λ is an area of ongoing research. For Stokes flow between parallel plates with ribs aligned with the flow, λ_T is very nearly equal to λ [11]. A study of thermal transport in a channel with ribs oriented perpendicular to the flow observed a more complex relationship between λ and λ_T [8]. In this study, the ratio of λ_T to λ was shown to depend on Peclet number, relative module width (ratio of pitch to channel hydraulic diameter), and cavity fraction. At low Peclet number and small relative module width, the ratio of λ_T to λ is nominally 2. However, increasing the Peclet number to the order of 1000 and setting the relative module width to 1 causes the ratio to decrease.

The effect of λ_T on heat transfer in an impinging jet has not been previously explained, but is of interest as it represents a common cooling scenario for self-cleaning surfaces. Section 2 of this paper presents an integral analysis to quantify the effect of isotropic slip and temperature jump (λ and λ_T) on the local and average thermal transport. Results from the model are presented and discussed in Section 3 and conclusions are given in Section 4.

2. Analysis

2.1. Model description

Shown in Fig. 1 is a schematic illustration of a vertical liquid jet impinging on a horizontal SHPo surface. A non-submerged, steady