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Heat transfer and hydrodynamics of free water jet impingement at low nozzle-to-plate spacings

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In this study, heat transfer and hydrodynamics of a free water jet impinging a flat plate surface are experimentally investigated. The effects of the nozzle-to-plate spacing, which is equal to or less than one nozzle diameter ($H/d = 0.08–1$), on the Nusselt number, hydraulic jump diameter, and pressure at the stagnation point are considered. The results show that the normalized stagnation Nusselt number, pressure, and hydraulic jump diameter are divided into two regions: Region (I) jet deflection region ($H/d \leqslant 0.4$) and Region (II) inertia dominant region ($0.4 < H/d \leqslant 1$). In region I, the normalized stagnation Nusselt number and hydraulic jump diameter drastically increase with decreasing the nozzle-to-plate spacing, since the stagnation pressure increases due to the jet deflection effect. In region II, the effect of the nozzle-to-plate spacing is negligible on the normalized stagnation Nusselt number and hydraulic jump diameter since the average velocity of the jet is constant, which means the jet deflection effect disappears. Based on the experimental results, new correlations for the normalized hydraulic jump diameter, stagnation Nusselt number, and pressure are developed as a function of the nozzle-to-plate spacing alone.

1. Introduction

Impinging jets are widely used in many engineering applications for the heating, cooling, and drying of surfaces as they offer high rates of heating, cooling, and drying. Major industrial applications for impinging jets include turbine blade cooling, electronic equipment cooling, metal annealing, and textile drying. Due to this diverse range of uses, many investigations have examined the heat transfer characteristics of impinging jets in the past decades [1–7,18,19].

For free liquid impinging jets at high nozzle-to-plate spacings ($H/d > 1$), Elicson and Webb [1] studied local heat transfer of laminar, transitional, and turbulent regions at Reynolds numbers from 300 to 7000 and nozzle-to-plate spacing from 1.5 to 50. They showed that the stagnation Nusselt number is not influenced by the spacing between the nozzle exit and the heated plate. Stevens and Webb [2] investigated local heat transfer coefficients at Reynolds numbers from 9600 to 10,500 and nozzle-to-plate spacing from 1.7 to 34. They suggested an empirical correlation for the stagnation Nusselt number as a function of nozzle-to-plate spacing, $Nu_0 \sim (H/d)^{-0.032}$. The effect of the nozzle-to-plate spacing on the Nusselt number was slight. In addition, they found an empirical correlation for the hydraulic jump diameter as a function of the Reynolds number only. The correlation did not include the nozzle-to-plate spacing effect. Brechet and Neda [8] studied both experimentally and theoretically the circular hydraulic jump. They suggested a theoretical correlation for the hydraulic jump radius as a function of the nozzle-to-plate spacing, $R_{hj} \sim (H/d)^{-1/6}$.

For submerged impinging jets at low nozzle-to-plate spacings of less than one nozzle diameter ($H/d = 0.1–1.0$), Lytle and Webb [9] studied the local heat transfer characteristics using an infrared thermal imaging technique. They observed that the local Nusselt number increases as the nozzle-to-plate spacing decreases when the flow rate (or Reynolds number) is fixed. A power-law relationship between the stagnation Nusselt number and the nozzle-to-plate spacing was presented in the form of $Nu_0 \sim (H/d)^{-0.191}$. Choo et al. [10–12] investigated heat transfer characteristics of impinging jets under a fixed pumping power condition at low nozzle-to-plate spacing ($H/d = 0.125–1.0$). They show that the Nusselt number is independent of the nozzle-to-plate spacing under a fixed pumping power condition. Choo et al. [13] studied the relationship between the Nusselt number and stagnation pressure of the submerged impinging jets at a large range of nozzle-to-plate spacing ($H/d = 0.125–40$). They found that the Nusselt number is strongly dependent on the stagnation pressure variation.

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Nomenclature

d	nozzle diameter [m]	Re	jet Reynolds number [ud/ν]
D_{hj}	hydraulic jump diameter [m]	r	lateral distance from stagnation point [m]
D_{hj}/d	dimensionless hydraulic jump diameter [–]	R_{hj}	hydraulic jump radius [m]
g	gravitational acceleration		
H	nozzle-to-plate spacing	*Greek symbol*	
Nu_0	stagnation Nusselt number	ν	dynamic fluid viscosity [m^2/s]
Nu_0^*	normalized Nusselt number ($Nu_0/Nu_{0,H/d=1}$)	ρ	fluid density [kg/m^3]
P_0	pressure of jet at stagnation point		
P_0^*	normalized stagnation pressure ($P_0/P_{0,H/d=1}$)		
Q	water flow rate [m^3/s]		

Several empirical correlations were suggested for free liquid jets at high nozzle-to-plate spacings and submerged impinging jets at low nozzle-to-plate spacings, respectively. However, the understanding of a relationship for heat transfer and fluid flow characteristics of free liquid jets at low nozzle-to-plate spacings is still limited. The purpose of this study is to determine the heat transfer and fluid flow characteristics of free liquid impinging jets for low nozzle-to-plate spacing (H/d = 0.08–1). Stagnation pressure of the free liquid impinging jets were measured to understand how it affects the Nusselt number and the hydraulic jump diameter. Based on the experimental results, correlations for the normalized stagnation Nusselt number, hydraulic jump diameter, and pressure were also developed as a function of the nozzle-to-plate spacing alone.

2. Experimental procedures

Fig. 1 shows a schematic diagram of the experimental apparatus. The liquid flow was supplied by a water reservoir to furnish a steady flow. A gear pump (Micropump) is used to supply water to the test section. The pumped water passed through a flexible tube before entering the test section. A positive-displacement-type flowmeter is used during the experiment. A heat exchanger is connected to a constant temperature bath to control the jet temperature. Three K-type thermocouples were located directly upstream and downstream of the flowmeters and downstream of the heat exchanger to monitor temperatures.

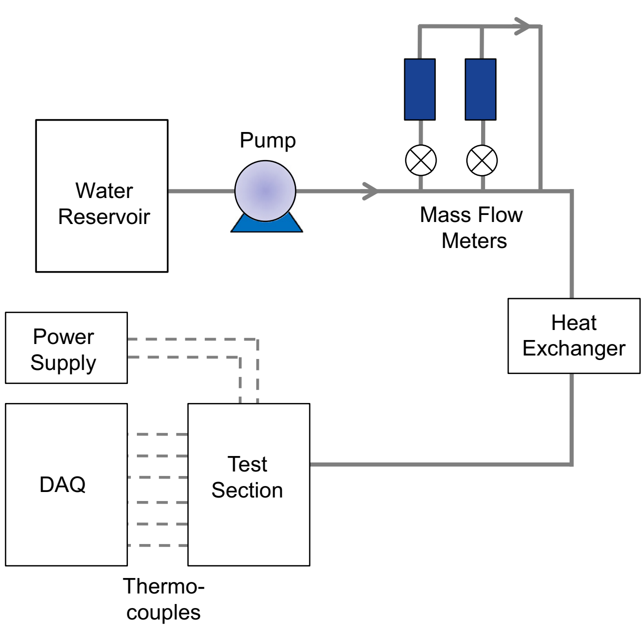

Fig. 1. Schematic diagram of the experimental set-up.

A circular nozzle was used in the experiment. It has a 6.65 mm inner diameter and is 420 mm long. The circular pipe was fixed on a 3-axis (x-y-z) stage with a 10 μm resolution made by Thorlabs, Inc. Thus, the nozzle could be moved either parallel or perpendicular to the direction of the jet. A flat acrylic plate with a diameter of 100 mm and thickness of 10 mm was used for pressure measurement. A pressure tap with a diameter of 0.1 mm was drilled and placed at the center of the flat acrylic plate. The pressure tap was connected to a micronanometer (Meriam M200-DI001). The manometers have a range of 0–6.89 kPa with accuracies of ±0.05%. The stagnation pressure of the impinging jet was measured at the center of impinging plate.

A schematic of the test section is presented in Fig. 2(a). The test section was constructed out of a clear acrylic sheet. The impingement surface was designed to be at a greater elevation than the pool bottom so the impinged liquid would fall off the impinging plate and into the pool. The edge of the impingement plate was chamfered to ensure smooth drainage of liquid. This relieves the impinging fluid from downstream influences. The circular impinging plate was constructed from 0.5 in thick PTFE Teflon disk with a 216 mm diameter and a 1 mm diameter orifice in the center. The orifice was connected to the manometer using flexible tubing. The hydraulic jump created on the impinging plate was measured using a digital camera (Nikon, D50) and a pulse generator (Fovitec - Speedlight flash, KD560) [14,15]. For accuracy and repeatability 5 min were given between each flow rate change to allow the system to reach a steady state for the different volumetric qualities. Both hydraulic jump and stagnation pressure were measured with minimum, actual, and maximum values at the steady state time.

Fig. 2(b) shows a schematic of the test section for local temperature measurement. The DC power supply was connected to the bus bar soldered to the heater at the center of the impingement surface. The heater is made of stainless steel that is 0.0508 mm thick, 12.5 mm wide and 67.8 mm long. The heater was connected to a high voltage DC power supply (Agilent 6651A #J03) in series with a shunt, rated 0–6 V and 0–60 A, allowing adjustable DC voltage to the electrodes. With DC electric current applied to the heater, a nearly uniform wall heat flux boundary condition was established. The amount of heat generation was obtained under the steady state condition. First, impinging fluid was introduced to the unheated heater, and then heat was applied to the heater. Values were recorded once the variation of the temperature difference between the heater and the nozzle exit was within 0.2 °C for 10 min. The voltage and the resistance across the heater were then measured in order to obtain the electrical energy input accurately with a multimeter.

A 0.5 in thick PTFE Teflon disk was used to mount the heater, thermocouples, and copper bus. The Teflon disk also provides insulation to minimize heat loss through the dry side of the heater. Five