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Heat transfer characteristics of impinging steady and synthetic jets over vertical flat surface

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ABSTRACT

In this paper, heat transfer characteristics of single-slot impinging steady and synthetic jets on a 25.4-mm × 25.4-mm vertical surface were experimentally investigated. The experiments were conducted with a fixed nozzle width of 1 mm. For the steady jet study, the parameters varied in the testing were nozzle length (4 mm, 8 mm, 12 mm, 15 mm), Reynolds (*Re*) number (100–2500), and dimensionless nozzle-to-plate spacing (H/D_h = 5, 10, 15, 20). Correlations for average Nusselt (*Nu*) number were developed to accurately describe experimental data. The heat transfer coefficient over a vertical surface increases with increasing *Re* number. For a small nozzle-to-plate spacing ($H/D_h = 5$), the average *Nu* number is not only a function of the *Re* number, but also a function of nozzle length. For large nozzle-to-plate spacing ($H/D_h \geqslant 10$) and a nozzle length larger than 8 mm, the heat transfer coefficient is insensitive to H/D_h and nozzle length. An 8-mm × 1-mm synthetic jet was studied by varying the applied voltage (20–100 V), frequency (200–600 Hz), and dimensionless nozzle-to-plate spacing (H/D_h = 5, 10, 15, 20). Compared to the steady jet, the synthetic jet exhibited up to a 40% increase in the heat transfer coefficient. The dynamic *Re* number was introduced to correlate heat transfer characteristics between synthetic jets and steady jets. Using the dynamic *Re* number collapses the synthetic and steady jet data into a single *Nu* number curve.

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1. Introduction

Steady impinging jets have been used in a wide range of industrial processes, including steel annealing, drying of paper and textiles, and thermal management of electronic devices and power electronics. Many studies of steady jets impinging over flat and smooth surfaces have been conducted. Reviews of the experimental work on the effects of an impinging jet on heat and mass transfer characteristics were reported by Martin [1] and Jambunathan et al. [2].

Although numerous studies have been conducted to demonstrate the impact of Reynolds (*Re*) number and nozzle-to-plate spacing on local and average heat transfer characteristics of a slot nozzle impinging steady jet [3–11], limited information is available at low *Re* numbers (<2500). Choo et al. [9] studied the local and average heat transfer characteristics of a micro-scale slot jet impinging on a heated flat plate. The effects of *Re* number in the range of 150–5000 and nozzle-to-plate spacing of 0.5–10 were investigated. At *Re* < 2500, the heat transfer characteristics are similar for both micro-scale and macro-scale impinging slot jets, showing little sensitivity to nozzle-to-plate spacing. The study, however, was conducted using only one unconfined slot nozzle with a cross-section area of 10 mm × 0.25 mm. Previous studies also found that heat transfer characteristics of impinging jets also depend on the jet inlet geometry and aspect ratio [12–18]. One of the most comprehensive studies on the effect of the jet inlet geometry and aspect ratio on local and average heat transfer was conducted by Koseoglu et al. [15] using nine confined impinging jets under the same mass flow rate. The results showed heat transfer enhancement in the stagnation region when a higher aspect ratio nozzle was used. The study was conducted at a high *Re* number (*Re* = 10,000).

The need for characterizing heat transfer at low *Re* numbers is in part due to emerging advanced air-cooling technology using synthetic jets to cool electronics systems. Synthetic jets have zero net mass flux because they take in and eject a high-velocity working fluid from a single opening. They do, however, have a net momentum flux, which is directed outward from the synthetic jet. The synthetic jet devices used in this study, developed by

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Nomenclature

A	target heater surface area (m^2)
D_h	nozzle hydraulic diameter (m)
$\bar{h}$	average heat transfer coefficient on the target heater surface (W/m^2 K)
H	nozzle-to-plate distance (m)
k	air thermal conductivity (W/m K)
L	nozzle length (m)
Nu	Nusselt number
$\overline{Nu}$	average Nusselt number of the target heater surface
Nu_w	average Nusselt number defined based on nozzle width ($Nu_w = h_{Avg} * W/k$)
Q_{loss}	heat loss through the Teflon block (W)
$Q_{surface}$	heat transfer on the target heater surface (W)
Q_{total}	total target heater heat consumption (W)
Re	Reynolds number
Re_{dyn}	dynamic Reynolds number
$\overline{Re}_{Steady\,jet}$	steady jet Reynolds number based on the mean gas velocity
$Re_{SteadyJet\_Max}$	steady jet Reynolds number based on the maximum gas velocity
Re_{SynJet_Max}	synthetic jet Reynolds number based on the time-averaged maximum gas velocity
S	heater width (m)
T_{air}	air temperature (°C)
$t_{ejection}$	time duration of the ejection stroke (s)
T_s	target heater surface temperature (°C)
t_{total}	duration of full synthetic jet flow cycle
U_{dyn}	dynamic gas velocity (m/s)
V_{dyn}	dynamic velocity (m/s)
V_{max}	the maximum gas velocity (m/s)
$\overline{V}_{SteadyJet}$	mean gas velocity of steady jet (m/s)
$V_{SteadyJet\_Max}$	maximum gas velocity of steady jet (m/s)
V_{SynJet_Max}	time-averaged maximum gas velocity of synthetic jet
V_{SynJet_Peak}	instantaneous peak gas velocity of synthetic jet
W	nozzle width (m)
Greek symbol	
ν	gas kinetic viscosity (m^2/s)

General Electric, use meso-scale piezoelectric actuators with a size on the order of centimeters to millimeters. Their characteristic *Re* number is dependent on the geometry and driving conditions of the synthetic jet. These synthetic jet devices are particularly well suited for small-scale electronics, such as printed circuit board components, where natural convection is not sufficient and other cooling devices, such as fans, are not practical due to space limitations. Compared to natural air convection, synthetic jets have been shown to enhance the local heat transfer coefficient by up to 15 times on a 25.4-mm vertical square surface [19–22]. Compared to the steady jet, up to 40% enhancement of heat transfer coefficients has been reported with a synthetic jet [19]. However, limited analysis has been conducted to correlate impingement heat transfer characteristics between steady jets and synthetic jets.

Motivated, in particular, by the dearth of heat transfer data at low *Re* numbers, a high-fidelity test rig was developed at the National Renewable Energy Laboratory to study the impingement heat transfer on a vertical surface for both steady jets and synthetic jets. The experimental parameters include *Re* number ($Re = 100$–2000), nozzle-to-plate spacing ($H/D_h = 5$–20), and nozzle aspect ratio ($L/W = 4$–15) at a fixed nozzle width of 1 mm, where W is the nozzle width (m), and L is the nozzle length (m). A synthetic jet with a cross-section area of 8 mm × 1 mm was tested to develop the correlation between synthetic jet and steady jet impingement heat transfer in the low *Re* number range.

2. Experimental study

2.1. Steady jets

Experiments were performed using an air flow test bench that both controlled and accurately measured flow rate. Fig. 1 shows the schematic of the steady jet experimental setup used for this study.

Compressed air was supplied to a desiccant dryer to remove moisture. The air was dried to a dew point of −20 °C or lower. It was then passed through a 5-μm particulate filter and regulated to a constant pressure within the range of 68 to 137 kPa. This regulated pressure served as the source air for a mass flow controller, Sierra model C100L, which provided a range of flow from 3.3 cm^3/s to 166 cm^3/s. The air was then passed through a plate heat exchanger for optional temperature control. Next, a laminar flow element, CME model 10 (0–166 cm^3/s), was used for an accurate measurement of the actual air flow rate. This was used to adjust the upstream mass flow controller set point. The air was then supplied to a settling chamber, where a honeycomb structure followed by two spaced screens straightened the flow and reduced turbulence prior to the air entering the nozzle. A calibrated T-type thermocouple was placed in the settling chamber to measure the jet air temperature. Air then exited the nozzle and impinged on the heat transfer target. The nozzle and target were placed within an acrylic plastic enclosure to minimize the effect of ambient air motion in the laboratory on the experiment. Heat transfer measurements were fully automated and controlled by a computer and a National Instruments data acquisition system.

Fig. 2 shows a close-up view and schematic of the steady jet nozzle and heater target. Steady jet nozzles were fabricated using a rapid prototyping process. Four nozzles were tested in this study, which had the same exiting width (W) of 1 mm but different lengths (L) of 4 mm, 8 mm, 12 mm, and 15 mm. Fig. 3 shows the steady jet nozzle design with exit size of 8 mm × 1 mm [20]. The nozzles are designed to attach to the settling chamber base plate, using an O-ring to prevent air leakage.

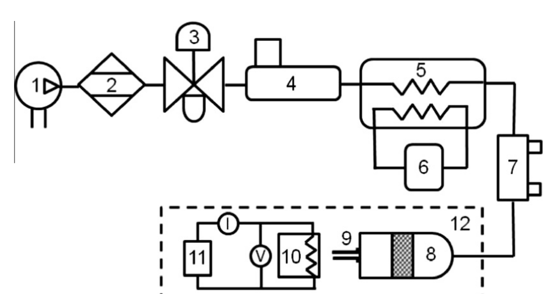

Fig. 1. Schematic of the steady jet experimental setup. 1 – Compressed air, 2 – desiccant dryer, 3 – filter/regulator, 4 – mass flow controller, 5 – plate heat exchanger, 6 – recirculating bath, 7 – laminar flow element, 8 – settling chamber, 9 – nozzle, 10 – heater target, 11 – heater power supply, 12 – isolation box.

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