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Technical Note

Effect of nozzle geometry on pressure drop and heat transfer in submerged jet arrays

Anja Royne, Christopher J. Dey *

School of Physics A28, University of Sydney, Sydney, NSW 2006, Australia

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1. Introduction

In the area of concentrating photovoltaics, where replacing expensive solar cell area by less expensive concentrator material is used as a way of reducing costs, one of the principal concerns is excessive heating of the photovoltaic cells. If the cells are not cooled efficiently, an elevated cell temperature can dramatically reduce their efficiency and can also lead to irreversible cell damage. In addition, there is a need for a cooling system which delivers a high rate of cooling at a minimum pumping power requirement. Liquid jet impingement is a promising method for this purpose because of its potential for very high heat transfer coefficients [1]. This technology is widely used in areas such as the thermal treatment of metals, cooling of internal combustion engines and thermal control of high power density electronic devices [2]. Various aspects of jet impingement such as the effect of Reynolds number, Prandtl number, nozzle-to-plate spacing, nozzle pitch and nozzle geometry have been studied extensively. Comprehensive literature reviews on jet impingement in generalare given by Martin [3], Webb and Ma [2] and Han and Goldstein [4].

This technical note deals specifically with the effect of nozzle configuration in submerged, confined jet arrays. Several other studies have investigated the effect of nozzle geometry on the Nusselt number of impinging jets. Lee and Lee [5] compared the heat transfer characteristics of sharp-edged, square-edged and an intermediate case of nozzle geometries and found the sharp-edged orifice to yield the highest local and average Nusselt number because of its more vigorous turbulence behaviour. The sensitivity to nozzle configuration was found to be stronger at low z/d. This finding was supported by Garimella and Nenaydykh [6], who attributed this phenomenon to interaction with ambient fluid downstream from the jet exit which tends to smooth out differences in the original flow structure. Garimella and Nenaydykh [6] also found the developing length of the nozzle (l/d) to be a major influence on the heat transfer under liquid jets. It was found that short developing lengths ($l/d < 1$) yielded the highest stagnation point heat transfer coefficients.

One aspect of nozzle configuration effects which has not received much attention is the tradeoff between heat transfer and pressure drop for a given nozzle geometry. Because the pumping power required for the cooling device is proportional to the product of flow rate and pressure drop, this is an important issue for the design of cooling devices, especially for those used in power producing systems. One study which looks into this problem is that of Brignoni and Garimella [7], who compared the heat transfer and pressure drop characteristics for orifice nozzles countersunk at two different angles with a regular square-edged orifice nozzle. Previous studies have found that countersunk orifices yield lower heat transfer coefficients when compared with square or sharp-edged orifices. However, Brignoni and Garimella showed that countersinking the nozzle significantly reduced the pressure drop while only slightly lowering the heat transfer coefficient. A countersunk angle of about 30°

* Corresponding author. Tel.: +61 2 9351 5979; fax: +61 2 9351 7726.
E-mail address: c.dey@physics.usyd.edu.au (C.J. Dey).

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Nomenclature

A	area
C_c	contraction coefficient
C_d	discharge coefficient
C_v	velocity coefficient
d	nozzle diameter
g	acceleration due to gravity
h	heat transfer coefficient
l	thickness of orifice plate
N	number of nozzles
Nu	Nusselt number, hd/k
Pr	Prandtl number, υ/κ
p	pressure
$\dot{q}$	power dissipated in foil
Q	volume flow rate
s	nozzle pitch
Re	nozzle diameter Reynolds number, $V_n d/\upsilon$
T	temperature
V	mean fluid velocity
z	orifice plate to impingement plate spacing

Greek symbols

κ	thermal diffusivity
ρ	liquid density
υ	kinematic viscosity

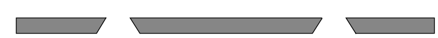
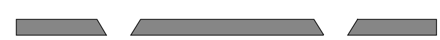
Subscripts

avg	average
c	vena contracta
n	nozzle
0	stagnation point
1	nozzle inlet
2	nozzle exit

(angle to normal) seems to yield the best result. At higher angles, the nozzle again starts to resemble a sharp corner which increases the pressure drop through it. The difference in heat transfer between different nozzles was found to become more significant with increasing Reynolds number.

Lee et al. [8] found that the nozzle diameter, with all dimensionless parameters held constant, had an influence on the Nusselt number in the impingement zone out to $r/d \sim 0.5$. In this region, the local Nusselt number was found to increase by about 10% from the smallest to the largest nozzle. This was explained by a higher velocity and turbulence intensity for the larger nozzles. Only relatively large nozzles of $d = 13.6$–34.0 mm were used. Garimella and Rice [9] also found the stagnation point Nusselt number to be dependent on the nozzle diameter but could find no systematic relationship.

Table 1
Overview of orifice plates used in the study

Device	Nozzle configuration
Short/straight	
Long/straight	
Sharp-edged	
Countersunk	

2. Experimental design and procedure

As the experimental set-up is described in detail by Royne and Dey [10] and Royne [11], only a brief overview will be given here. The water flows through the inlet into a plenum chamber manufactured from a 20 mm wide square stainless steel tube and is forced through an orifice plate. After impinging onto the heated surface, it is drained through the return flow chamber. This geometric arrangement was not chosen primarily for the nozzle geometry study, but because this is part of a larger study investigating a prototype jet impingement cooling device where the spend liquid is drained in a direction normal to the heated surface. The heated surface consists of a 29 mm × 25 mm, 0.05 mm thick stainless steel foil which was clamped and stretched tightly between two aluminium bus–bars. The foil temperature distribution was recorded using thermographic liquid crystals and a digital camera. The spatial resolution of the temperature measurements was found to be less than 0.1 mm, but because of the finite power intervals the resulting temperature maps sometimes had slightly poorer spatial resolution.

Four different orifice plates were tested, as listed in Table 1. All of the arrays had four-nozzles with the same nozzle diameter $d = 1.4$ mm and nozzle pitch to diameter ratio $s/d = 7.14$. The nozzle-to-plate spacing to diameter ratio was set to $z/d = 3.57$ because several studies have shown that the maximum average heat transfer coefficient for submerged jets occurs at a nozzle-to-plate spacing $z/d \approx 3$–4 [9,12]. The thickness of the orifice plates was 1 mm for all plates except for the one called ‘long/straight’ which was 2 mm thick. The sharp-edged and countersunk nozzles were made using a conventional 30° countersinking tool.