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Radial heat transfer behavior of impinging submerged circular jets

D.W. Zhou *, C.F. Ma

Enhanced Heat Transfer and Energy Conservation, The Key Laboratory of Ministry of Education, China
Heat Transfer and Energy Conversion, The Key Laboratory of Beijing Municipality, College of Environmental and Energy Engineering, Beijing University of Technology, Beijing 100022, PR China

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Abstract

Experiments were performed to investigate the radial heat transfer behaviors of impinging submerged circular jets. Local heat transfer rate at several fixed radial locations and different nozzle-to-plate spacings were correlated and compared. Results reveal that with the jet being far from the stagnation point, the coefficient in the correlation $Nu \sim Re$ decreases while the exponent characterizing the flow pattern of the working liquid increases.

Keywords: Jet impingement; Convection heat transfer; Transition

1. Introduction

Due to a steady elevation in heat dissipation at the chip, module and system levels, considerable efforts have been devoted to the study of thermal problems encountered in the design of electronic and microelectronic equipment. Many efficient paths for heat transfer have been proposed as an alternate cooling mode. Among them, jet impingement cooling technique is still considered one of the best means of accommodating high heat fluxes while maintaining low wall temperature [1].

The common feature of impinging jets is the heat transfer enhancement in the stagnation region and a rapid decay of heat transfer in the wall jet region due to the boundary layer buildup on the target surface. Due to nonuniform heat transfer performance in a radial direction, considerable attention has been given to the effects of jet parameter on stagnation point heat transfer, producing many correlating expressions with different test conditions. In reviewing the existing literature [2,3], no previous study has been found addressing the local heat transfer behavior at the fixed radial locations. So the present study will help to fill this gap.

The purpose of the present study is twofold: to determine local heat transfer rate of impinging submerged jets and to interpret the exponent meaning of the jet Reynolds number of $Nu \sim Re$ expression in light of the differences of the flow fields.

2. Experimental apparatus and method

R113 was selected as the working fluids and circulated in a closed loop having provision for filtering, metering, preheating and cooling. Single flush-mounted heater of 5 mm × 5 mm was vertically fixed on one side of the chamber and nozzle with interior diameter of Ø1.01 mm was employed. The temperature at jet exit and the inner center surface of the heater were monitored with a 40-gage iron-constantan thermocouple of Ø0.08 mm, respectively. Further details about the experimental apparatus and procedure can be found in Ref. [4,5].

* Corresponding author. Address: Heat Transfer and Energy Conversion, The Key Laboratory of Beijing Municipality, College of Environmental and Energy Engineering, Beijing University of Technology, Beijing 100022, PR China. Tel./fax: +86 10 6739 2774.
E-mail address: drdwzhou@hotmail.com (D.W. Zhou).

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Nomenclature

C	empirical constant	u	jet exit velocity
d	nozzle inside diameter	z	nozzle-to-plate spacing
h	local heat transfer coefficient	z/d	dimensionless nozzle-to-plate spacing
k	thermal conductivity	v	kinematic viscosity
m	exponent used in equation	Nu	local Nusselt number
r	radial distance from the stagnation point	Re	jet Reynolds number, ud/v
r/d	dimensionless radial distance	Pr	Prandtl number, $c_p\mu/k$

The experimental data were finally reduced in terms of local Nusselt number and jet Reynolds number:

$$Nu = hd/k, \tag{1}$$

$$Re = ud/v, \tag{2}$$

where k and v are the conductivity coefficient and kinematic viscosity of the working fluid, respectively. The uncertainties in the Nusselt number and jet Reynolds number were determined to be less than ±4.5% and 5.0%.

3. Results and discussion

Measurements were made to investigate local heat transfer behaviors of impinging submerged circular jets at several fixed radial locations for jet Reynolds number in the range of 690–43568. The results for $z/d = 2$ and 5 in the form of $Nu/Pr^{1/3} \sim Re$ are depicted in Fig. 1(a,b), respectively. Stagnation point heat transfer are determined and compared with each other as a validation exercise in this study.

Fig. 1(a) presented the effect of jet Reynolds number on stagnation point (i.e., $r/d = 0$) Nusselt number. Stagnation point Nusselt number increases remarkably with jet Reynolds number. For comparison, Fig. 1(a) also presents the experimental data of Ma et al. [6] for an R113 submerged jet and Lienhard et al. [7] for a water free surface jet. As illustrated by squares in the figure, stagnation point heat transfer in this study was enhanced slightly. Stagnation point heat transfer data within the potential core could be well expressed by the correlation:

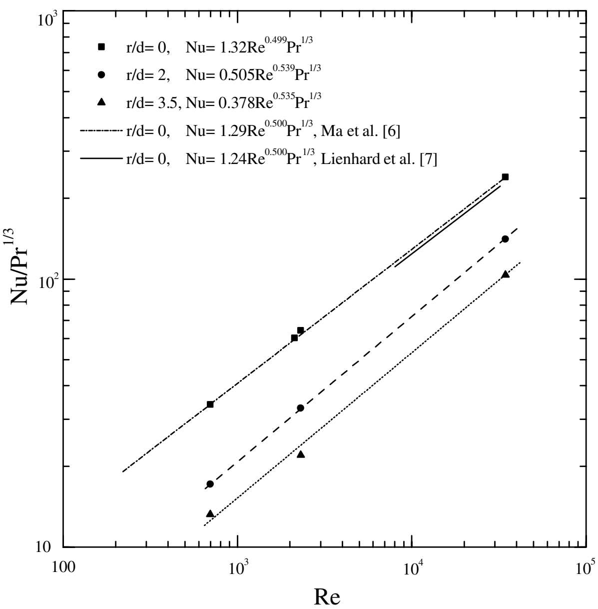
$$Nu = CRe^m Pr^{1/3}, \tag{3}$$

where the standard Prandtl number exponent of 1/3 is adopted from the recommendation of Ref. [6], and the coefficients C and exponent m were determined from experimental data: $C = 1.32$, and $m = 0.499$. These empirical constants are very close to the values of $C = 1.29$ and $m = 0.5$ reported in Ref. [6]. The exponent of jet Reynolds number clearly indicates the laminar characteristic of the impingement flow at the stagnation point, where the favorable pressure gradient parallel to the target surface tends to laminarize the jet flow. This fluid flow characteristic has been verified with the available exact analytical solutions [8] and experimental results [2,9] of jet impingement heat transfer. Independent of jet Reynolds number and jet type, the present data agree well with the previous experimental results of Refs. [6,7]. Comparisons show that very small differences are perceived for a submerged jet of 0.28% and a free surface jet of 4.33% and these differences are mainly attributed to different initial turbulence intensity at the nozzle exit [10] and radial velocity gradient between them, respectively.

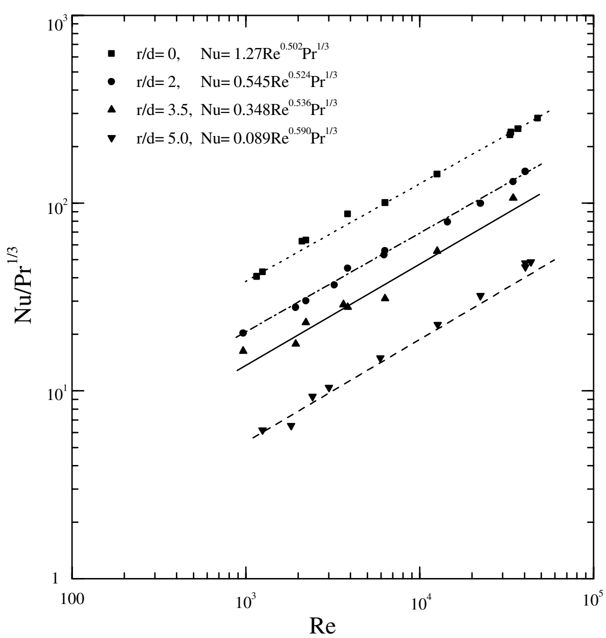
Fig. 1(a) also presented variation of local Nusselt number at $r/d = 2$ and 3.5 as a function of jet Reynolds number. These data exhibit the same trend as that obtained at the stagnation point although limited data were taken there. However, local Nusselt number decreases notably with the increase of radial distance from the stagnation point because of reduced radial velocity. The effects of jet Reynolds number and radial distance on local Nusselt number remain unchanged as nozzle-to-plate spacing is increased from $z/d = 2$ to 5, as shown in Fig. 1(b). Since the plate was held within the potential core, all the experimental data taken from submerged circular R113 jets were collected and also correlated with Eq. (3) at $z/d = 2$ and 5.

Table 1 presented variations of coefficient and exponent of $Nu \sim Re$ expression at different nozzle-to-plate spacing and radial locations. At fixed nozzle-to-plate spacing, inspection of Table 1 indicates that the exponent m increases with radial distance while the coefficient C decreases sharply. For example, as the radial distance is increased from $r/d = 0$ to 5, the exponent at $z/d = 5$ increases from $m = 0.502$ to 0.590 and yet the coefficient decreases from $C = 1.27$ to 0.089. For given radial location, as nozzle-to-plate spacing is increased from $z/d = 2$ to 5, the exponent basically increases while the coefficient C decreases, implying a low heat transfer rate there.

We now turn our attention to the exponent meaning of jet Reynolds number of Eq. (3). A numerical study carried out by Zhou et al. [11] indicated that pressure gradient controlled the flow pattern of the working fluid close to the impingement plate, which in turn determined the profile of radial heat transfer rate distribution. The pressure gradient distribution on the impingement plate is believed to be irrelevant to nozzle diameter [12], but strongly depends on nozzle exit configuration [13], nozzle-to-plate spacing [14], jet Reynolds number and jet type. The decrease and disappearance of pressure gradient usually implies an increase of



(a) z/d= 2



(b) z/d= 5

Fig. 1. (a,b) Correlation and comparison of local Nusselt number at fixed radial locations.

turbulence intensity there. For given jet type and nozzle exit configuration, as the impingement plate is held within the potential core, jet Reynolds number has a weak influence on the pressure gradient.

As mentioned above, the jet Reynolds number dependence $Re^{0.5}$ suggests laminar characteristics of impingement heat transfer at the stagnation point. The pressure gradient decreases rapidly with radial distance from the stagnant point within the stagnation zone. Thereby, the exponent of jet Reynolds number increases gradually. This has been verified by the data shown in Fig. 1(a,b).

Table 1
Correlation of local Nusselt number at fixed radial locations

z/d	r/d	C	m	Range of Re number	rms
2	0	1.32	0.499	690–34466	0.99
2	2	0.505	0.539	690–34466	1.00
2	3.5	0.378	0.535	690–34466	0.99
5	0	1.27	0.502	1153–47634	0.99
5	2	0.545	0.524	965–40440	0.99
5	3.5	0.348	0.536	965–34451	0.96
5	5	0.089	0.590	1247–43568	0.99

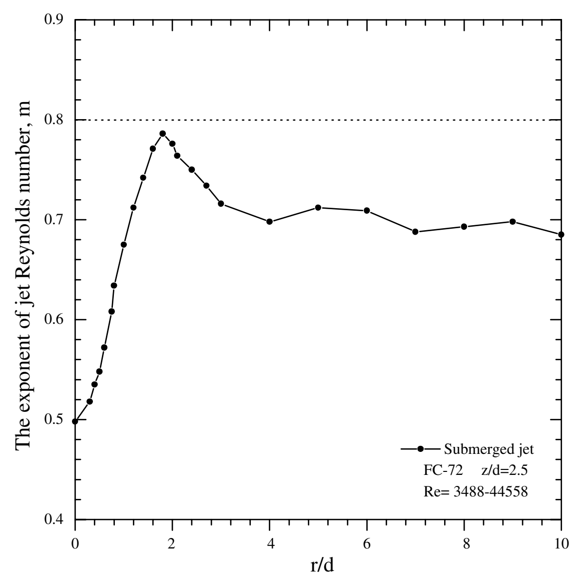


Fig. 2. Variation of jet Reynolds number exponent with respect to radial distance.

Based on the experimental data reported by Qin et al. [9], the present authors obtained the variation of the exponent m of jet Reynolds number with respective of radial distance. The result for a FC-72 submerged jet at $z/d = 2.5$ and $Re = 3488–44558$ was plotted in Fig. 2. As can be seen in the figure, the exponent m at the stagnation point approaches 0.5. It increases sharply with initial radial distance from the stagnation point and has a local peak at $r/d = 1.8$, where the second peak of local heat transfer rate occurs [9,15]. Elison and Webb [16] confirmed that the maximum value of jet Reynolds number exponent was 0.8, which characterizes a turbulent flow of liquid on the impingement plate. It implies that the transition from laminar to turbulent occurs due to disappearance of negative pressure gradient. Thereafter, the exponent m decreases moderately. As the radial distance is beyond $r/d = 4$, the exponent m has a weak dependence on fluid flow, indicating a fully developed wall jet there. Since the exponent m is closely relevant to turbulence intensity of working fluid, it can be concluded from the correlating expression of $Nu \sim Re$ that the exponent of jet Reynolds number implies the flow pattern of the working fluid on the impingement plate, irrespective of jet types.

4. Conclusions

The most important findings of this experimental study are:

(1) For R113 submerged jets, local heat transfer rates at given radial locations were correlated and compared.
(2) The exponent of jet Reynolds number of $Nu \sim Re$ expression implies a flow pattern of the working fluid regardless of jet type. Laminar and turbulent flow correspond to $m = 0.5$ and 0.8, respectively.

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