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Prandtl number dependence of impingement heat transfer with circular free-surface liquid jets

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1. INTRODUCTION

Forced convection with liquid jets is usually associated with extremely high heat/mass transfer coefficients [1]. With liquid as the working fluid, two operation modes are possible: free-surface jets and submerged jets. In the former case, a liquid jet is exposed to a gaseous environment. In the latter, a jet is discharged into stagnant fluid of the same type. Impinging liquid jets have been employed in some technical processes of heating or cooling, including oil jets in oil-cooled internal combustion engines [2], water and fluorocarbon liquid jets in electronic devices [3–5], and water jets in hot rolling process of steel [6]. The Prandtl number of the above coolant liquids ranges from about 3 to several hundreds, covering three orders of magnitude. However, the influence of Prandtl number on convective heat transfer is still one of the least understood aspects for impinging jets. Usually, the Prandtl number dependence is expressed with a power function of Prandtl number itself: $Nu \sim Pr^m$. The values of exponent m are different with each other from available literature. From analytical studies [1, 7–9] the power value of 1/3 was obtained, being consistent with the experimental results of water and FC72 [10], transformer oil [11, 12] and electrochemical liquid [13, 14]. However, this value was also determined or adopted as 0.4 for water [8, 15–18] and FC77 [16]. Based on their average heat transfer measurements, Metzger *et al.* [19] proposed the exponents to be 0.24 and 0.487 for lubricating oil and water jets respectively. The objective of this work was to conduct an extensive and consistent study using three test liquids to testify the Prandtl number dependence of local heat transfer for free-surface circular jets in the range of Prandtl number between 7 and 262. Empirical formulas were presented to correlate all the heat transfer data at stagnation point. Good agreement was observed between the correlations and the experimental data from the present work and some other resources [14, 20].

2. EXPERIMENTAL APPARATUS AND METHOD

The experimental Apparatus and method in this work are similar with those described in Refs [11, 12, 21]. Only a brief description of the experimental apparatus and method is given here. The test liquid is circulated in a closed loop which had provision for filtering, metering, preheating and cooling. The test section assembly was vertically fixed on one side of the chamber made of stainless steel. The main part of the test section was a strip of 10 μm thickness of constantan foil with a heated section of 5×5 mm exposed to the liquid jets. This active section of the foil was used as an electrically heating element as well as a heat transfer surface. The temperature of the center of the inner surface of the heater was measured by a 40 gage iron–constantan thermocouple which was electrically insulated from the foil yet in close thermal contact. The heat transfer surface was sustained at constant heat flux condition. The surface heat flux was calculated from the electrical power supply to the heater.

Three liquids, R113 (Pr = 7.2–8.9), kerosene (Pr = 20.9–21.2) and transformer oil (Pr = 134–262), were chosen as working fluid in the present study. Kerosene is certainly not a good coolant because of its flammability, but the Prandtl number of kerosene is similar with that of the dielectric liquid commonly used for immersion cooling of microelectronic devices [3]. The liquid jets issued from a horizontal jet-tube of 0.987 mm of inside diameter and 35 mm in length. The length-diameter ratio is sufficient to give reasonable fully developed profiles of mean velocity and turbulence [22]. The jet-tube assembly was fixed on a three-dimensional coordinate rack, and could be adjusted with respect to the test section within 0.01 mm. The jet temperature was also measured by a 40 gage iron–constantan thermocouple upstream the jet-tube. By recording the center temperature of the test section for various locations of the jet tube the horizontal temperature distributions could be obtained for given jet conditions and surface heat fluxes. Then the radial profiles of the local heat transfer coefficient could be calculated. All the properties of the working fluid were evaluated at the film temperature by averaging the wall and jet temperature. The uncertainty in Nusselt number and Reynolds number did not exceed 5 and 5.5%, respectively. The detail of the experimental apparatus, procedure and data uncertainty were presented in refs. [21, 23].

3. RESULT AND DISCUSSION

Presented in the paper is the local heat transfer result expressed in the terms of the Nusselt number

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

where the local heat transfer coefficient is defined by

$$h = q/(T_w - T_{aw}). \tag{2}$$

In the range of jet exit velocity in the present work ($u < 20$ m s^{-1}), the difference between T_{aw} and T_j can be neglected for R113 and kerosene. For transformer oil, the result of measured adiabatic wall temperature was expressed by the recovery factor

$$r = \frac{T_{aw} - T_j}{u^2/2C_p}. \tag{3}$$

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NOMENCLATURE

C	coefficient	Re	ud/ν, jet Reynolds number
C_p	special heat at constant	r	recovery factor, radial distance from stagnation point
d	jet nozzle diameter	T_{aw}	adiabatic wall temperature
h	local heat transfer coefficient	T_j	jet static temperature
k	thermal conductivity of fluid	T_w	wall temperature
m	exponent	u	jet exit velocity
Nu	hd/k, local Nusselt number	Z	nozzle-to-plate spacing.
Nu_0	local Nusselt number at stagnation point		
n	exponent	Greek symbols	
q	surface heat flux	μ	dynamic viscosity
Pr	$C_p\mu/k$, Prandtl number	ν	kinematic viscosity.

The recovery factor of transformer oil used in this work was obtained from ref. [12]. Measurements were first made to determine the local heat transfer profiles with the three liquids. For each working fluid the profiles were measured at two constant jet velocities. Since the profiles were symmetric about the stagnation point, only half-profiles were measured. All the distribution curves are of bell shape with the maximum appearing at the stagnation point. After normalization to the maximum at stagnation point, the profiles are presented in Fig. 1. The local heat transfer coefficients diminish remarkably from the stagnation point along the radial direction. Presented in the figure is also the predicted curves by the integral solution [1, 9]:

$$Nu/Nu_0 = 0.777(r/d)^{-0.5}$$

$$\text{for } 1 < r/d < r_\nu/d$$

$$r_\nu/d = 0.177Re$$

$$Nu/Nu_0 = 1.85Re^{-1/6}\left[\frac{25.7}{Re}(r/d)^3 + 0.857\right]^{-2/3}$$

$$\text{for } r/d > r_\nu/d. \qquad (4)$$

Good agreement is observed between the data and the predictions.

It is seen that all the curves tend to collapse to a single profile, nearly independent of Reynolds number in the range of $r/d < 3$ in the present study. It is also found that the shape of the heat transfer profiles for the three liquids is not affected by the fluid Prandtl number which ranges from 7 to 262 in this work.

Variation of Nusselt number at stagnation point was examined with the nozzle-to-plate spacing using the three test liquids at constant jet velocities. The experimental data are plotted in Fig. 2 after normalization to the maximum at $z/d = 2$. An empirical formula was developed to correlate all the experimental data:

$$Nu_0/Nu_{0,\max} = e^{-7.89\times10^{-4}(z/d)^{1.85}}. \qquad (5)$$

Equation (5) presents all the data within $\pm 10\%$, with an average error of $\pm 1.5\%$. Good agreement is seen from the figure between the data and the correlation. It is noted that the stagnation heat transfer rate slightly declines with increasing of z/d. This trend is consistent with the result reported by Stevens and Webb for water jets [13].

The measured Nusselt numbers at stagnation point are presented in Fig. 3 against the Reynolds number. It has been well established by analytical [7–9] and experimental [8, 10–18, 20] studies that the heat transfer coefficient at stagnation point can be expressed by

$$Nu_0 = CPr^mRe^n. \qquad (6)$$

Using a least-squares technique, the coefficients in eqn (6) were obtained: $C = 1.38$, $m = 0.329$ and $n = 0.489$. With this set of constants, eqn (6) correlates 91.3% of the present data within $\pm 15\%$. The average error and standard deviation are ± 7 and 8.2%, respectively. Good agreement of the correlation is observed from Fig. 3(a) between the correlation and the experimental data. The exponents of Reynolds number and Prandtl number are almost identical with

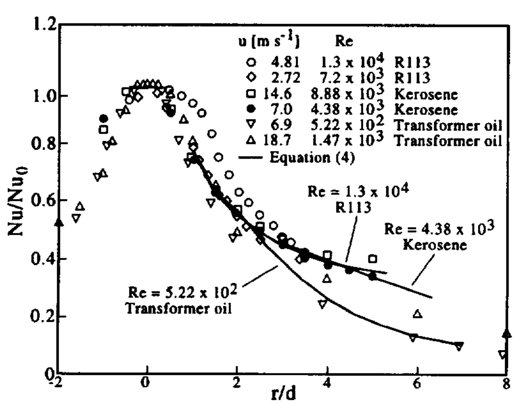

Fig. 1. Radial profiles of local heat transfer coefficients.

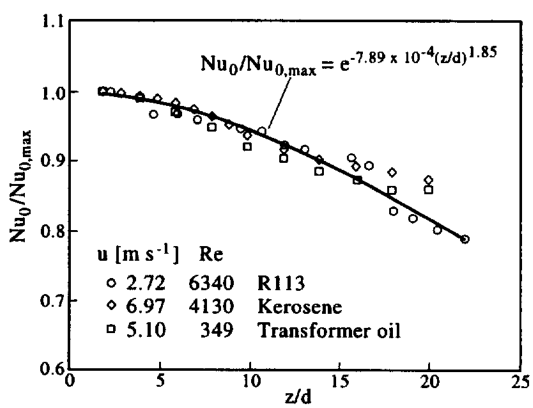

Fig. 2. Variation of stagnation point Nusselt number with nozzle-to-plate spacing.

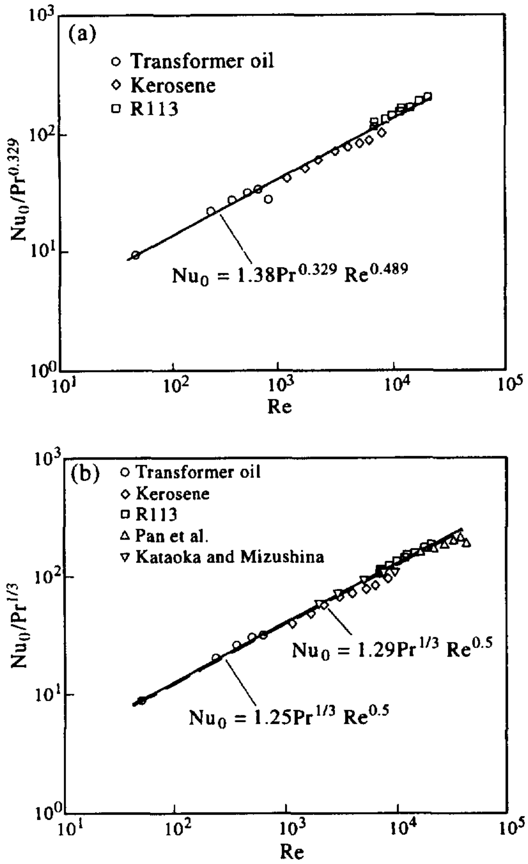

Fig. 3. Heat transfer at stagnation point.

1/2 and 1/3, respectively, from the previous results of analytical [1, 7, 9] and experimental [4, 11–13] studies. With the two exponents set equal to 1/2 and 1/3 respectively, the coefficient C was determined from the present data to be 1.25 which is very close to 1.29 proposed in refs. [4, 11]. With this set of coefficients, eqn (6) presents 91.3% of the present data within $\pm 15\%$, with almost same average error and standard deviation as the equation with the former empirical constants. The comparison of the correlation and the data was given in Fig. 3(b). The Reynolds number dependence $Re^{1/2}$ suggests the laminar characteristics of the impingement heat transfer at stagnation zone where the strong favorable pressure gradient tends to laminarize the liquid flow. The Prandtl number dependence $Pr^{1/3}$ testified with impinging circular jets of R113, kerosene and transformer oil in the present work, is consistent with the results of analytical [7, 9] and experimental [10–14] studies, covering a wide range of Prandtl number from about 7 to 262. To highlight the influence of Prandtl number the experimental result given in Fig. 3(b) are plotted again in Fig. 4. It is seen from Fig. 4 that eqn (6) with $m = 1/3$ and $n = 0.5$ well correlates all the experimental data in the wide range of Prandtl number. The Prandtl number dependence of $Pr^{0.4}$, which is extensively adopted for liquids with medium Prandtl number [5, 15–18, 20, 21], is also presented in Fig. 4. As shown in the figure, the correlation with $Pr^{0.4}$ much overestimates the heat transfer rates in the range of large Pr number. In the case of $Pr = 262$ the deviation may be as much as 45%, but for lower Prandtl number, the two power values of Prandtl number don't make significant difference in heat transfer calculation.

Most recently, local heat transfer at stagnation zone was

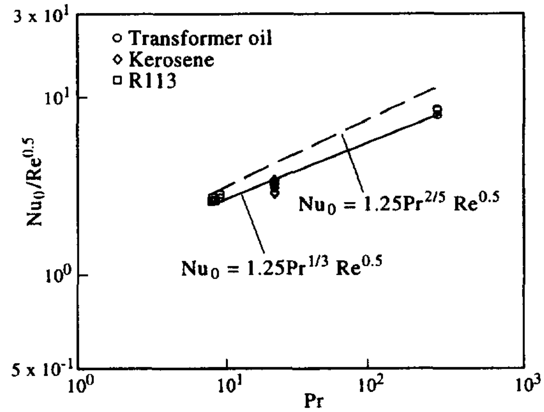

Fig. 4. Effect of Prandtl number on stagnation point heat transfer.

studied in detail by Pan *et al.* [20] for impinging circular free-surface water jets. Some experimental data with similar nozzle geometry to our investigation (fully-developed pipe flow) from ref. [20] are plotted in Fig. 3(b) for comparison. These data are in good agreement with the predicted curve by eqn (6), as shown in the figure. Presented in Fig. 4 are also the experimental data of mass transfer with circular free-surface jets of heavy electrochemical liquid ($Pr = 2420$–3300) obtained by Kataoka and Mizushina [14]. Considering the large difference in test liquid and experimental arrangement between the study of ref. [14] and this work, the agreement of the data with the correlation should be satisfied. The agreement between eqn (6) and the data from refs. [14, 20] further verified the validity of the correlation presented in this paper, and extended its applicability to the range of even higher Prandtl number.

4. CONCLUSIONS

Local measurements were made to investigate the characteristics of convective heat transfer from small heaters to impinging free-surface circular jets of three liquids with Prandtl number between 7 and 262. Local heat transfer profiles were obtained and compared with integral solutions [1, 9]. Good agreement indicates their independence on Prandtl number and insensitivity on Reynolds number. Variation of stagnation point heat transfer was studied as a function of Prandtl number, Reynolds number and nozzle-to-plate spacing. Empirical formulas were developed to correlate all the heat transfer data at stagnation point. The Prandtl number dependence of $Pr^{1/3}$ was testified by the present experimental data, as well as the data of water and heavy electrochemical liquid from other resources.

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Effect of variable fluid properties on impingement heat transfer with submerged circular jets

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1. INTRODUCTION

Forced convection problems can be simplified with an assumption that the fluid properties remain constant throughout the flow field. It has been found convenient to obtain the constant property analytical solutions, and to collect experimental data with small temperature differences between the fluid flow and the heat transfer surface. However, this assumption is obviously an idealization in engineering practice as the thermal properties of almost all the working fluids used in industry vary with temperature. The effect of variable fluid properties is to yield distorted velocity and temperature fields, resulting in considerable

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