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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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NOMENCLATURE

C_1, C_2	coefficients
C_p	specific heat at constant pressure
d	jet nozzle diameter
k	thermal conductivity
h	local heat transfer coefficient
m	exponent
Nu	local Nusselt number (hd/k)
n	exponent
p	exponent
Pr	Prandtl number $(C_p\mu/k)$
q	surface heat flux exponent
r	recovery factor
Re	Reynolds number (ud/ν)
T	temperature
u	mean liquid velocity at nozzle exit.

Greek symbols

μ	dynamic viscosity
ν	kinematic viscosity.

Subscripts

aw	adiabatic wall temperature
b	bulk temperature
f	film temperature
j	jet static temperature
w	wall temperature.

change in heat transfer performance in comparison with the case of constant properties. To take account of the effect of variation of fluid properties, two correlation approaches based on the constant property result have been in common use: the reference-temperature method and the property-ratio method [1]. Using the former, heat transfer coefficients can be obtained from constant property correlations or solutions if the fluid properties are evaluated at so-called reference-temperature in the heat transfer calculation. There is no general rule to determine the reference temperature, but the film temperature, average of bulk temperature and wall temperature, has been extensively used as the reference temperature. In the latter, the heat transfer coefficients are obtained with all the fluid properties evaluated at the bulk temperature, then corrected by a function of a ratio of some pertinent property evaluated at the wall temperature to the same property evaluated at the bulk temperature (or adiabatic wall temperature). For most liquids, their thermal conductivity and specific heat are relatively independent of temperature, but the viscosity decreases significantly with temperature. So, an excellent approximation can be used as follows:

$$Nu = Nu_b(\mu_b/\mu_w)^m \tag{1}$$

For laminar liquid flow in round tubes, Joshi and Bergles found $m = 0.14$ for constant-heat-flux wall both in the entry and developed regions [2], and $m = 0.11$ for constant-temperature wall in the developed region [3] both in the case of heating ($\eta_w < \mu_b$). These results are consistent with the analytical results by Deissler [4] and Yang [5]. For turbulent liquid flow in tubes, $m = 0.11$ was recommended by Petukhov [6] for the case of liquid heating. This value is close to 0.14 obtained in an early experimental study by Sieder and Tate [7].

Concerning heat transfer with impinging liquid jets, most experimental studies were performed with small temperature difference (usually 10 K or less) between the jets and the heated surfaces [8–17]. The effect of the fluid properties was insignificant in this case. To the best knowledge of the present authors, there are only two reports pertaining to this topic available in open literature. Using lubricating oil as the working fluid, Metzger *et al.* [17] studied the effect of fluid properties on impingement heat transfer with circular free-surface jets. Taking account of the recovery effect with large Prandtl number liquid jets, they evaluated all the fluid properties, except μ_w, at the adiabatic wall temperature instead of the bulk temperature in equation (1):

$$T_{aw} = T_j + r\cdot\frac{u^2}{2C_p}. \tag{2}$$

The results reported in ref. [17] were average heat transfer measurements which were mainly related to the flow and heat transfer in wall jet zone where the flow velocity was essentially parallel to the target surface. It is also noted that the measurements were made in a narrow range of μ_{aw}/μ_w between 1.29 and 1.68, and the exponent of the ratio (μ_{aw}/μ_w) was determined to be 0.37 which was very different from the values commonly used for liquid flow in pipes. More recently, Gu *et al.* [18] measured average heat transfer with PAO (poly alpha olefin, $Pr = 30$–55) coolant jets and determined $m = 0.25$ in the case of heating, but their research was conducted with two-dimensional jets, which was not directly related to the present work. Further research is still required to clarify the effect of the variation of fluid properties on impingement heat transfer, particularly in the stagnation zone, where the flow velocity is perpendicular to the heat transfer surface in contrast to the parallel flow in channels. The objective of this work is to investigate the fluid property effect on local heat transfer with impinging circular submerged liquid jets in experimental detail. Heat transfer coefficients at stagnation point were measured in wide ranges of experimental parameters and tested in correlating both by the two procedures.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

Transformer oil was chosen as the test liquid in this study for its good cooling performance and great sensitivity of thermal properties (μ and Pr) with temperature. The experimental method is the same as described in refs. [8–10, 12]. Only a brief description is given here. The working fluid was circulated in a closed loop which had provision for filtering, metering, preheating and cooling. The test section was vertically mounted on one side of the test chamber made of stainless steel. The main part of the test section was a strip of 10 μm thick constantan foil with a heated section of 5×5 mm (nominal) exposed to the coolant. This active part 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 heated surface was sustained at constant heat flux condition. The heat flux was calculated from the electrical power supplied to the heater.

Transformer oil jets issued from a horizontal jet-tube of 0.987 mm inside diameter and 35 mm length. 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 static temperature was measured also by a 40 gage iron–constantan thermocouple upstream the jet-tube.

The present experiment was composed of five test runs in the range of $Pr = 134$–348 and $Re = 1.69 \times 10^2$–2.07×10^3. In each test run, the jet velocity was maintained to be a constant between 3.71 and 20.5 m s^{-1} within $\pm 5\%$, and the adiabatic wall temperature was adjusted to a constant around 18°C to better than ± 0.3°C. During the test runs, several steady-states were achieved and maintained for heat transfer measurement by varying the heat flux from 1.55×10^5 to 1.99×10^6 w m^{-2}. Heater temperature at stagnation point was recorded from 29.2 to 68.5°C at these steady-states to calculate the heat transfer coefficient by the following formula:

$$h = \frac{q}{T_w - T_{aw}} \qquad (3)$$

where the recovery factor was obtained from ref. [12].

In this study, the ratio of μ_{aw}/μ_w ranges from 1.68 to 6.61, corresponding to the heater wall temperature from 29.2 to 68.5°C. The maximum overall uncertainties in Nusselt number and Reynolds number were determined to be ± 5 and $\pm 5.5\%$, respectively. More details of the experimental apparatus, procedure and data uncertainty were presented in refs. [8–10, 12].

3. RESULTS AND DISCUSSION

The measured heat transfer coefficients at stagnation point collected in this work were presented in terms of the Nusselt number:

$$Nu = h \cdot d/k. \qquad (4)$$

The experimental results were tested to be correlated both by the property-ratio method and the reference-temperature method.

3.1. *Property-ratio method*

When this correlation procedure is used, the fluid properties in the dimensionless groups should be evaluated at the adiabatic wall temperature defined by equation (2). Based on the results from refs. [8–16], the following equation is proposed to correlate the experimental data:

$$Nu_{aw} = C_1 \cdot Pr_{aw}^p Re_{aw}^q (\mu_{aw}/\mu_w)^m \qquad (5)$$

where the exponent p has been determined to be 1/3 in the previous investigations [9, 12, 15]. With p set equal to 1/3, the values of C_1, m and q were determined by a least-squares technique from the experimental data: $C_1 = 0.89$, $q = 0.547$, and $m = 0.16$. With these constants, equation (5) correlates all the experimental data within $\pm 10\%$. The average error and standard deviation of this correlation are ± 4.73 and 5.57%, respectively. The Reynolds number dependence of local heat transfer is illustrated in Fig. 1. Good agreement is seen between the correlation curve and the present data. The power 0.547 of Reynolds number determined in this work is very close to the value of 0.5 obtained from previous studies [8–10, 12], which clearly indicates the laminar characteristic of the impingement heat transfer within the stagnation zone where the favorable pressure gradient tends to laminarize the liquid flow. The data presented in Fig. 1 are plotted again in Fig. 2 to highlight the effect of variable fluid properties on local impingement heat transfer. It is seen from Fig. 2 that all the data of the five sets fall along a single straight line by equation (5). The success in collapsing the experimental data of very different thermal properties is due to the use of the ordinate variable, which fairly accounts for the difference in fluid properties. It is observed from the figure that the heat transfer coefficients increase slightly with the ratio of μ_{aw}/μ_w.

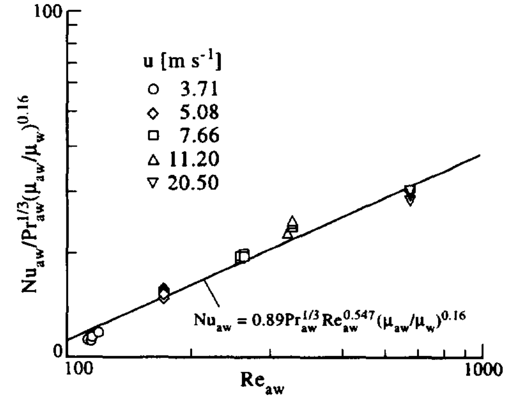

Fig. 1. Data correlation by property-ratio method.

Good agreement is found between the data and the correlation. It is noteworthy that the exponent 0.16 of ratio (μ_{aw}/μ_w) in equation (5) is almost identical with 0.14 determined for laminar pipe flow [2, 4, 7].

3.2. *Reference-temperature method*

Reference-temperature method has been successfully used to correlate the experimental data of submerged circular jets of water [8, 13–15], R113 [11], ethylene glycol [9], tranformer oil [9, 12], kerosene [10], and Fluorinert liquid [14] in the case of lower temperature difference (about 10°C or less) between the jets and the heated surfaces. Film temperature, the average of wall and jet temperature, was chosen as the characteristic temperature. It was tested in this study to correlate the transformer oil data of ratio (μ_{aw}/μ_w) up to 6.6 using the following formula:

$$Nu_f = C_2 \cdot Pr_f^p \cdot Re_f^n \qquad (6)$$

where the exponent p was set equal to 1/3 as in equation (5), the other constants were obtained from the experimental data by a least-squares technique: $C_2 = 0.843$, and $n = 0.570$. With these empirical constants, equation (6) presents 90% of the present data within $\pm 10\%$. The average error and standard deviation of the correlation are ± 5.57 and 6.64%, respectively. Experimental data are presented in

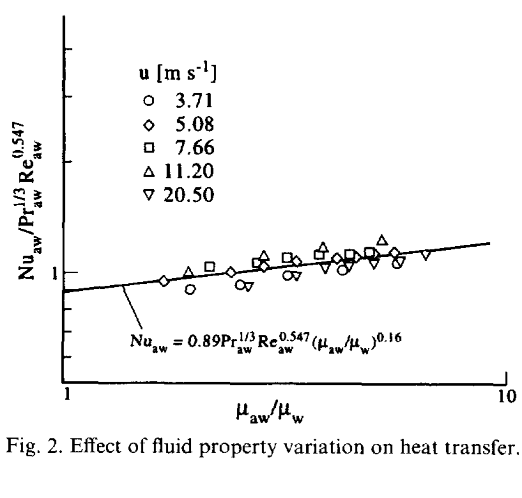

Fig. 2. Effect of fluid property variation on heat transfer.

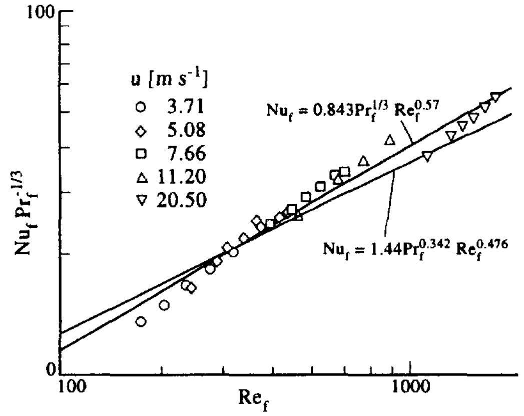

Fig. 3. Data correlation by reference temperature method.

Fig. 3 together with the correlation curve by equation (6). Good agreement is seen from the figure between the data and the curve. Presented in Fig. 3 is also a correlation based on experimental data of seven working fluids with Prandtl number between 0.7 and 348 [19]:

$$Nu_f = 1.44Pr_f^{0.342} \cdot Re_f^{0.476}. \tag{7}$$

The two correlation curves agree well with each other, as shown in the figure. Figure 3 indicates the fact that the characteristic temperature method is effective in correlating the impingement heat transfer data with variable fluid properties, but the equation (5) based on property-ratio method seems slightly more precise than equation (6).

4. CONCLUSIONS

Local measurements were made to determine the heat transfer coefficients at stagnation point with submerged circular jets of transformer oil in the range of $\mu_{aw}/\mu_w = 1.7$–6.6. The effect of variation of fluid properties on impingement heat transfer was examined in experimental detail. Two formulae, based on property-ratio method and reference-temperature method respectively, were developed to correlate the experimental result. Both the two correlation procedures were found to be quite effective for large Prandtl number liquid jets. The power of the ratio μ_{aw}/μ_w in the viscosity correction term in the property-ratio method was determined to be 0.16 which was very close to 0.14 proposed for laminar liquid flow in pipes. This method seems slightly more precise than the reference temperature method.

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