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Technical Note

Applicability of traditional turbulent single-phase forced convection correlations to non-circular microchannels

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1. Introduction

Turbulent convection in microchannels has received much interest as an effective cooling mechanism for high power density systems. Among the diverse applications are the cooling of high power resistive magnets, accelerator targets, nuclear reactor cores, and high density multi-chip modules in modular electronics. Within the past few years several studies have shown the characteristics of forced convection in microchannels to be significantly different than large channels [1–5].

Recently, Adams et al. [1] showed that the turbulent heat transfer coefficients for water flowing in circular microchannels with diameters ranging from 0.76 to 1.09 mm were significantly higher than the values predicted by traditional Nusselt number correlations, e.g. the Gnielinski correlation, for the same Reynolds and Prandtl numbers. In some cases, the enhancement reached a value of nearly 100%. The data were in agreement with those previously reported by Yu et al. [3] for a diameter of 0.102 mm. Based on their experimental data, combined with the data of Yu et al. [3], Adams et al. [1] developed the following correlation to predict the enhanced Nusselt numbers for turbulent convection in microchannels:

$$Nu = Nu_{\mathrm{Gn}}(1 + F) \tag{1}$$

$$F = C Re\left(1 - \left(\frac{D}{D_0}\right)^2\right) \tag{2}$$

where Nu_{Gn} is Nusselt number predicted by the Gnielinski correlation [6]. That correlation is given by

$$Nu_{\mathrm{Gn}} = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{(1/2)}(Pr^{(2/3)} - 1)}. \tag{3}$$

The recommended friction factor correlation is that of Filonenko [7]:

$$f = (1.82\ \log(Re) - 1.64)^{-2}. \tag{4}$$

A least squares fit of all the data resulted in $C = 7.6 \times 10^{-5}$ and $D_0 = 1.167$ mm. For a confidence level of 95%, the developed correlation predicts the experimental heat transfer data within $\pm 18.6\%$. The range of validity for Eqs. (1) and (2) is $2.6 \times 10^3 \leq Re \leq 2.3 \times 10^4$, $1.53 \leq Pr \leq 6.43$ and $0.102 \leq D \leq 1.09$ mm.

The data used to develop Eqs. (1) and (2) show that the heat transfer augmentation due to a small hydraulic diameter increases with decreasing diameter, as can be inferred from the form of the correlation. Of particular interest is the constant D_0, which represents the smallest diameter for which traditional heat transfer correlations are valid. The purpose of the current investigation is to independently test the validity of this hydraulic diameter limit for turbulent convection in non-circular channels.

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Nomenclature

C	constant in heat transfer enhancement function
c	specific heat [kJ kg^{-1} K^{-1}]
D	inside diameter of microchannel [mm]
D_0	reference diameter in Eq. (2) [mm]
D_h	hydraulic diameter [mm]
f	friction factor
F	heat transfer enhancement function, Eq. (2)
k	thermal conductivity [W m^{-1} K^{-1}]
Nu	Nusselt number (hD/k_{fl})
Pr	Prandtl number ($\mu_{fl}c_{fl}/k_{fl}$)
Re	Reynolds number ($\rho_{fl}VD/\mu_{fl}$).

Greek symbols

μ	dynamic viscosity [kg m^{-1} s^{-1}]
ρ	density [kg m^{-3}].

Subscripts

fl	fluid
Gn	based on the Gnielinski correlation.

2. Experimental apparatus and procedure

An experimental facility was designed and constructed to measure the local heat transfer coefficient for the single-phase forced flow of water subject to a given flow rate and wall heat flux level. A schematic diagram of the test facility and a listing of its various hardware components and instrumentation are given in Fig. 1 and Table 1, respectively. A detailed description of the experimental facility is given in [8]. The facility's salient features are discussed in the following.

One of three pumps (A, B1 and B2 in Fig. 1) was used to provide a steady flow of water through the test section, depending on the flow rate. A counterflow heat exchanger upstream of the test section inlet allowed the test section inlet temperature to be maintained at $50 \pm 1°C$ while a nitrogen tank assembly (items J–L in Fig. 1) maintained exit pressure at a value of 1034 ± 8 kPa. Heat input to the test section was measured using a wattmeter (9 in Fig. 1). Experiments could be run using either water fully saturated with dissolved air or fully degassed water by employing the saturation loop (items X–AA in Fig. 1) or the degassing assembly (items R–U in Fig. 1), respectively. In order to eliminate any effect of the desorption of dissolved noncondensable gases, only fully degassed water was used for the present study.

Fig. 2 shows the design of the test section. The test section was machined from a single piece of cylindrical free machining copper. The flow passage of the test section was created using the electrode discharge machining (EDM) technique, and was designed to simulate the interior subchannels of a triangularly-arranged rod bundle with rod diameters of 3.2 mm and a pitch to diameter ratio of 1.15. The dimensions were selected to produce a hydraulic diameter of the flow passage nearly identical to the diameter limit (D_0) in the correlation of Adams et al. [1], Eq. (2). Exact measurements of the channel dimensions resulted in a hydraulic diameter of 1.13 mm. Heat input to the test section was provided by an insulated NiChrome wire wrapped in a helical fashion around the outer surface. High temperature insulation (calcium silicate) surrounded the entire test section.

Three 0.25 mm diameter Type-E thermocouple probes were located 120° apart at each of two different planes located 1.25 cm from the respective ends of the heated section; the thermocouple junctions were located 0.25 mm from the inner surface of the flow channel. Extrapolations of the thermocouple measurements based on one-dimensional radial conduction allowed the inner wall temperature of the channel to be determined. The local fluid temperatures were calculated based on a macroscopic energy balance applied to the fluid between the test section inlet and the axial locations of the thermocouple probes. The wall-fluid temperature differences and the measured heat fluxes were used to calculate local heat transfer coefficients.

Table 1
Major component in experimental test facility

A	Centrifugal pump	R	Relief valve
B1, B2	PD pumps	S	Feed water tank
C	Ball valves	T	Gas catching column
D1	Flow control valve	U	Degassing tank relief
D2	Metering valve	V	Saturation tank
E	Bypass control valve	W	Saturation tank relief
F	In-line heater	X	Vertical column
G	208 VAC power	Y	Compressed air tank
H	Variac	Z	Recirculation pump
I1, I2	Heat exchangers	1, 2	Rotameters
J, K	Relief valves	3,4	Flow transducers
L	Nitrogen cylinder	5	Pressure transducer
M	Bellows accumulator	6	DP transducer
N	O2 sensor housing	7, 8	Thermocouples
O	Three way valves	9	Wattmeter
P	Isolation valves	10	Dissolved O_2 probe
Q	Supply water inlet		

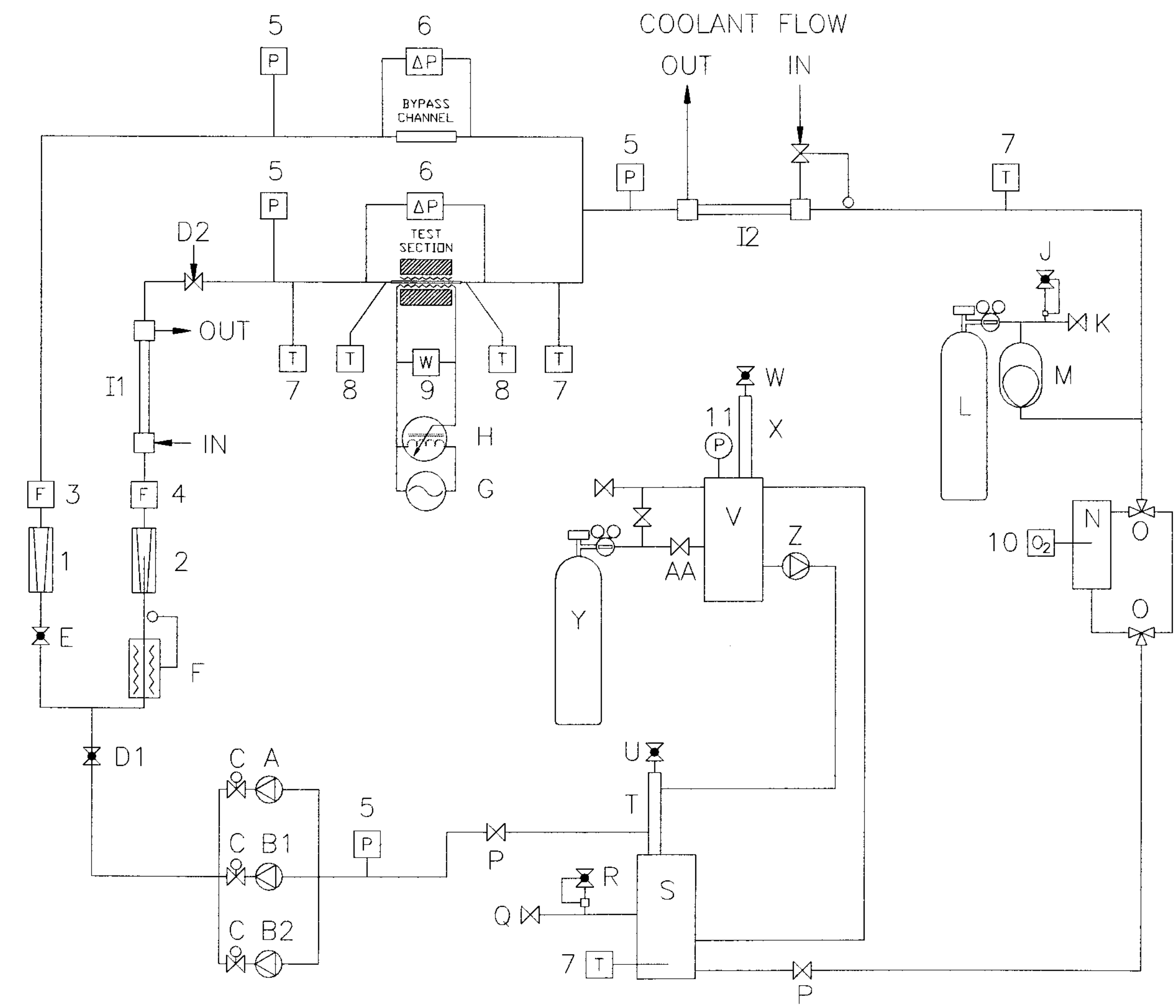

Fig. 1. Experimental test facility.

3. Results

The resulting hydraulic diameter of the test section was 1.13 mm which is within 2.5% of D_0 in Eq. (2). If this size limit for the applicability of traditional Nusselt-type correlations is valid for non-circular geometries, very little, if any, deviation from the Gnielinski correlation (with D replaced by D_0) would be expected for the measured Nusselt number in this non-circular channel.

Fig. 3 shows the variation of the experimentally obtained Nusselt numbers as a function of Reynolds numbers. Also shown are curves corresponding to predictions made by the Gnielinski correlation for the lowest and highest Prandtl numbers covered by the data, namely, 1.22 and 3.02. For all heat flux levels, the experimental Nusselt numbers are within the limits predicted by the Gnielinski correlation.

In order to account for the different Prandtl number ranges realized by the different heat flux levels, it was desired to make a direct comparison of the experimentally obtained Nusselt numbers to the values predicted by the Gnielinski correlation for the same Reynolds and Prandtl numbers. Fig. 4 gives such a comparison. Clearly, the data follow the Gnielinski correlation closely, all the data falling within $\pm 10\%$ of the predicted values. Also evident from Fig. 4 is that there seems to be no effect of heat flux as to whether the Gnielinski correlation applies; i.e. the correlation accurately predicts the data over the entire ranges of Reynolds and Prandtl numbers used, namely, 3.9×10^3–2.14×10^4 and 1.22–3.02, respectively. These

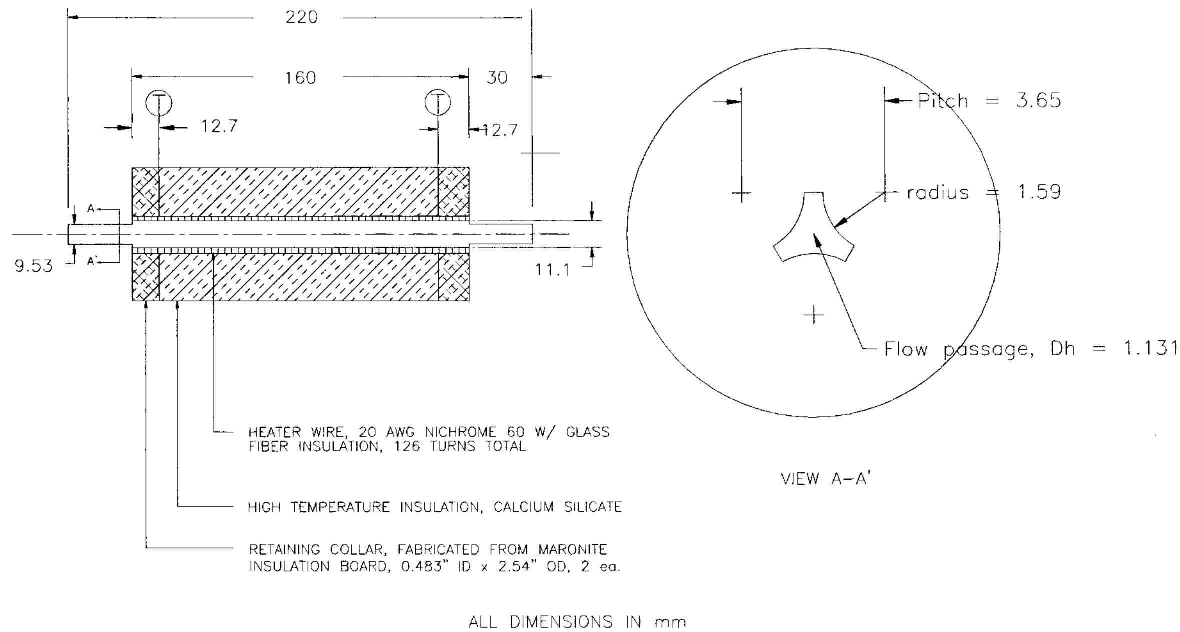

Fig. 2. Test section design and geometry.

results suggest that the size limit reported by Adams et al. [1] for the applicability of traditional Nusselt numbers for turbulent convection in small channels is also valid as a hydraulic diameter limit for non-circular channels.

4. Conclusion

Heat transfer experiments were performed for the single-phase forced convective flow of water in a non-circular microchannel having a hydraulic diameter of 1.13 mm. The hydraulic diameter of the microchannel was selected to match the diameter limit estimated by Adams et al. [1] below which heat transfer enhancement over values predicted for large channels can be expected. Within $\pm 10\%$, all the heat transfer data were well predicted by the Gnielinski correlation, providing strong evidence that a hydraulic diameter of approximately 1.2 mm is a reasonable lower limit for the applicability of standard turbulent single-phase Nusselt-type correlations to non-circular channels.

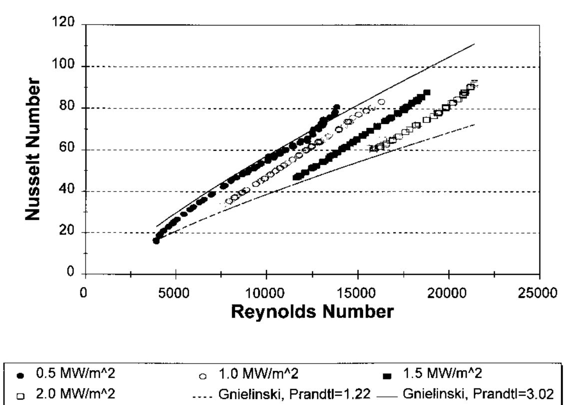

Fig. 3. Comparison of experimental Nusselt numbers to the Gnielinski correlation.

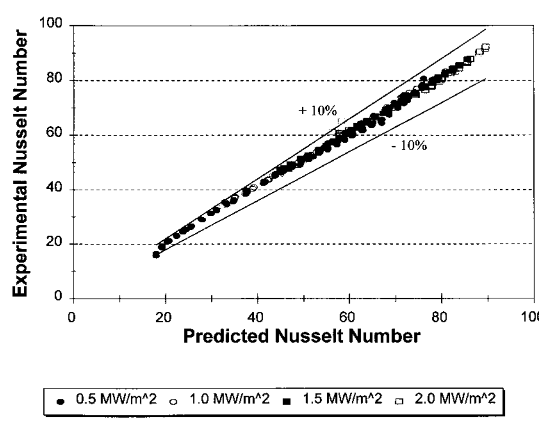

Fig. 4. Comparison of experimental to predicted Nusselt numbers based on the Gnielinski correlation.

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