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# Turbulent heat transfer with FC-72 in small diameter tubes

Yantao Li, Katsuya Fukuda *, Qiusheng Liu, Makoto Shibahara

*Department of Marine Engineering, Kobe Univ., 5-1-1, Fukaeminami, Higashinada, Kobe 658-0022, Japan*

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ABSTRACT

The heat transfer process of turbulent, single-phase forced convection of FC-72 through small diameter tubes with 1 and 1.8 mm inner diameters was experimentally investigated. The influence of Reynolds number ($Re_d$), Prandtl number ($Pr$), viscosity ratio ($\mu/\mu_w$) and ratios of heated length to inner diameter ($L/d$) on turbulent heat transfer was studied in detail. The experimental data were also compared with the values calculated by classical correlations for conventional sized channels. The results indicated that the classical heat transfer correlations are not adequate for calculation of the heat transfer coefficient in small diameter tubes. The Nusselt numbers ($Nu_d$) for 1 and 1.8 mm depend on $Re_d$ in a different manner compared to classical correlations. The deviation from classical heat transfer correlations increased as the $Re_d$ increase. The turbulent heat transfer correlation for FC-72 flow in tubes with diameters of 1 and 1.8 mm has been developed based on the experiment data. The differences between experimental and predicted $Nu_d$ are within ±15%.

## 1. Introduction

Thermal issues are increasingly affecting the performance of high density multi-chip modules in super computers, high power resistive magnets and other electronical devices due to the high heat dissipation rates requirements in a relatively small space. Forced convection of liquid through mini or micro channels is an effective cooling mechanism to meet those requirements. Kandlikar [1] classified channels with hydraulic diameters of 0.01–0.2 mm as micro-channels, hydraulic diameters of 0.2–3 mm as mini-channels, and hydraulic diameters greater than 3 mm as conventional sized channels. Whether the classical theory and empirical correlations for convectional sized channel could be applicable to small sized channels becomes a crucial issue. In the recent decades, many studies on single-phase forced convection heat transfer in small sized channels have been reported.

Several investigators claimed higher $Nu_d$ values than those predicted by correlations based on conventional theory. Yu et al. [2] investigated particularly the heat transfer of nitrogen gas and water in micro tubes in the turbulent regime. The $Nu_d$ values in turbulent regime were considerably larger than those predicted by means of the conventional theory. Adams et al. [3] studied single-phase heat transfer in circular tubes with diameters of 0.76 and 1.09 mm with water. They reported that $Nu_d$ were usually higher than those predicted by conventional heat transfer correlations. They observed that the deviation increased as the channel diameter decreased and $Re_d$ increased. Bucci [4] and Lee [5] also have found the Nusselt number higher than predicted by correlations based on conventional theory.

On the other hand, some researchers have found the $Nu_d$ smaller than that measured for convectional sized channels. Peng and Wang [6] investigated single-phase convective heat transfer of water and methanol for fully developed turbulent flow in micro channels. $Nu_d$ can be predicted by Dittus–Boelter' correlation [7] for conventional tubes by varying the empirical constant coefficient from 0.023 to 0.00805, which means that the measured $Nu$ were lower than the predicted value. Qu et al. [8] investigated the heat transfer characteristics of water flowing through trapezoidal silicon micro channels. They found that the measured $Nu_d$ were lower than those predicted.

There have also been many other reports in which conventional correlations are able to predict single-phase heat transfer coefficients in small sized channels. Owahib and Palm [9] performed experiments on the single-phase forced convection through small sized channels. Their experimental results in the turbulent region were in very good agreement with the conventional correlations. Agostini et al. [10] investigated the heat transfer coefficient in circular and rectangular mini-channels with hydraulic diameters of 0.77–2.01 mm. They found their experimental results in the turbulent regime coincided with the Gnielinski correlation [11].

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* Corresponding author. Fax: +81 784316303.
*E-mail addresses:* yantaoli87@gmail.com (Y. Li), fukudak@kobe-u.ac.jp (K. Fukuda), qsliu@maritime.kobe-u.ac.jp (Q. Liu), sibahara@maritime.kobe-u.ac.jp (M. Shibahara).

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**Nomenclature**

| | |
|---|---|
| $c$ | specific heat, J/kg K |
| $c_p$ | specific heat at constant pressure, J/kg K |
| $d$ | inner diameter of the test tube, m |
| $f$ | friction factor for smooth tube |
| $h$ | heat transfer coefficient, W/m$^2$ K |
| $I$ | current flowing through standard resistance, A |
| $L$ | heated length, m |
| $L_e$ | entrance length, m |
| $n$ | Exponent, or slope on the log-log graph |
| $Nu_d$ | $= hd/\lambda$, Nusselt number |
| $P$ | pressure, kPa |
| $P_{in}$ | pressure at inlet of heated section, kPa |
| $P_{ipt}$ | pressure measured by inlet pressure transducer, kPa |
| $P_{out}$ | pressure at outlet of heated section, kPa |
| $P_{opt}$ | pressure measured by outlet pressure transducer, kPa |
| $Pr$ | $= c_p\mu/\lambda$, Prandtl number |
| $Q$ | heat input per unit volume, W/m$^3$ |
| $Q_0$ | initial exponential heat input, W/m$^3$ |
| $q$ | heat flux, W/m$^2$ |
| $r$ | radial coordinate |
| $r_i$ | test tube inner radius, m |
| $r_o$ | test tube outer radius, m |
| $R_1$ to $R_3$ | resistance in a double bridge circuit, Ω |
| $\mathrm{Re}_d$ | $= ud/\nu$, Reynolds number |
| $R_s$ | standard resistance, mΩ |
| $R_T$ | test tube resistance, mΩ |
| $S$ | surface area, m$^2$ |
| $T$ | temperature, K |
| $T_a$ | average temperature of test tube, K |
| $T_{in}$ | inlet liquid temperature, K |
| $T_L$ | $= (T_{in} + T_{out})/2$, average bulk liquid temperature, K |
| $T_{out}$ | outlet liquid temperature, K |
| $(T_{out})_{cal}$ | calculated outlet liquid temperature, K |
| $T_s$ | heater inner surface temperature, K |
| $t$ | time, s |
| $\Delta T_L$ | $= (T_s - T_L)$, temperature difference between heater inner surface temperature and average bulk liquid temperature, K |
| $u$ | flow velocity, m/s |
| $V$ | volume, m$^3$ |
| $V_{FM}$ | voltage signal of flow velocity, mV |
| $V_I$ | voltage difference across standard resistance, mV |
| $V_R$ | voltage difference across test tube, mV |
| $V_T$ | output voltage of the double bridge circuit, mV |
| $V_{TLi}$ | voltage signal of inlet temperature, mV |
| $V_{TLo}$ | voltage signal of outlet temperature, mV |
| $\lambda$ | thermal conductivity, W/mK |
| $\rho$ | density, kg/m$^3$ |
| $\tau$ | exponential period, s |
| $\theta$ | angular coordinate, ° |

*Subscripts*

| | |
|---|---|
| $a$ | average |
| $cal$ | calculated |
| $in$ | inlet |
| $out$ | outlet |
| $l$ | liquid |
| $w$ | wall |

Recent literature overviews [12–14] also indicate that there is still disagreement among the researchers. Carefully designed and systematic experimental investigation is still necessary before final conclusions can be drawn.

It can be concluded that lots of researches are performed for water according to the previous brief literature review. However, water is not an ideal refrigerant in cooling systems for electronical devices due to its corrosiveness and electronical conductivity. It would cause electronical devices burnout or failure once leakage occurred. Therefore, FC-72 was proposed for their use in electronics cooling system because of its high thermal conductivity, low viscosity and surface tension, and super dielectric characteristic.

This study was motivated by demands for investigating the heat transfer behavior of FC-72 flowing in small diameter tubes and further developing turbulent heat transfer correlations.

## 2. Experimental apparatus and methods

### 2.1. Experimental facility and loop

The experimental apparatus used in this investigation consisted of a flow loop, a test section assembly, a heat input control system and a data reduction system as shown in Fig. 1. The liquid was circulated by a canned type non-seal pump through a previously evacuated loop shown with arrows. Upstream of the non-seal pump is a 440 μm filter which is used to remove any particles. The flow rate was measured with a precision vortex flow meter (Kofloc, FM01) which is located upstream of the test section. Fine adjustment of flow rate was accomplished by regulating the frequency of a three-phase alternating power source to the circulation pump with an inverter (Hitachi, WJ200). The test section's inlet temperature was maintained by a preheater or a cooler (TRL N135 type cryostat). The test section's inlet pressure was established by saturated vapor in pressurizer and could be maintained within 1 kPa of a desired value by utilizing the heater controller of the pressurizer. The test section was held in a vacuum tank so that the heat loss from test section to the surroundings could be minimized.

### 2.2. Test section

Fig. 2 illustrates the construction and dimensions of the test section. The circular SUS304 tube with inner diameters of 1 and 1.8 mm, wall thickness of 0.5 mm, several heated lengths, $L$, and a commercial finish surface condition were used in this experiment. Both ends of the test tube were soldered to the copper plates which served as electrodes to provide direct current to heat the test tube. The solder joint with a very low resistance could withstand high pressure and high temperature. A Bakelite block was used to provide support for the test tube. Bakelite plates insulated the test tube from the rest of the loop both electrically and thermally. Sealing between the copper plates, insulating Bakelite plates and SUS304 instrumentation blocks was accomplished by using gaskets. All the plates and block were tightened together with four steel bolts. PTFE sleeves were inserted between the bolts and copper plates for electrical and thermal isolation from each other. The SUS304 instrumentation blocks facilitated the installation of thermocouples and pressure transducers. Two 1-mm-diameter, sheathed, K-type thermocouples were inserted at the center line of the conduit for measuring the inlet and outlet liquid temperatures. All thermocouples were calibrated for better accuracy before installation. The inlet and outlet pressures were measured by two