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Transient heat transfer of helium gas for forced convective flow through a narrow tube with different lengths

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Abstract
The understanding of the heat transfer process for helium gas cooling in the fusion blanket is important for the development of fusion reactors. This research aims at clarifying the transient heat transfer characteristics of helium gas for turbulent flow in a narrow tube. A circular platinum tube with an inner diameter of 1.8 mm was heated by exponentially increasing heat inputs and cooled by helium gas. The lengths of the tube were 30 mm and 50 mm. The heat transfer coefficients gradually increased for the e-folding time shorter than about 1.5 s. This shows that the heat transfer process is in a transient state for this region. The heat transfer coefficients increased with the increase of flow velocity, and the effect of the flow velocity became weak in the transient state region. By comparing with the experimental results of tube with different lengths, it was obtained that the quasi-steady state heat transfer coefficients for the length of 30 mm were higher than those for the lengths of 50 mm and 90 mm. However, the heated length showed a weak influence for the lengths of 30 mm and 50 mm in the transient heat transfer with a relatively smaller e-folding time. The variation of transient Nusselt number with Fourier number at different Reynolds number describes well the trend for the variation of heat transfer coefficient with e-folding time at different velocities. Based on the experimental data, an empirical correlation was obtained for transient heat transfer at different lengths by using Fourier number.

Keywords : Transient heat transfer, Helium gas, e-folding time, Forced convective flow, Narrow tube, Different lengths

1. Introduction

In order to solve problems related to safety, environment, and energy shortage, the utilization of fusion energy has been expected to be an energy source of next generation (Ehrlich, 2001). Recently, research on fusion reactors is being conducted around the world. For example, the Chinese Fusion Engineering Test Reactor (CFETR) was designed to demonstrate 50-200 MW fusion power, 30-50% duty cycle, and high tritium breeding ratio (TBR) (Li et al., 2015). The design of fusion blanket is one of the most important components in CFETR, and the helium cooled solid breeder blanket is considered to be the most promising design among several blanket module concepts (Wang et al., 2018). The cooling channels are approximately 3 mm per side and must remove about 0.3 MW/m^2 of heat flux. The disruption of plasma in fusion reactor may induce powerful short-time thermal impacts (Yakushin et al., 1998) and causes the transient-state heat transfer. The surface heat flux on the first wall will increase very sharply in several milliseconds (Dobran, 2012). The blanket is subjected to severe heat load conditions, and elucidating the forced convection transient heat transfer of helium gas through the blanket is important.

There are some experimental, analytical or numerical investigations on transient heat transfer for forced convective flow. However, the works focused on the transient process heated by an exponential heat source are not sufficient. Soliman and Johnson (1968) theoretically and experimentally studied for the transient mean wall temperature of a flat plate of forced convective flow heated by an exponential time-dependent heat source. Compared with the experimental data for water, their solution of heat transfer coefficient was higher than the data. Kataoka et al. (1983) experimentally researched transient heat transfer under forced convection for water flowing over a platinum wire heated by exponentially

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increasing heat input in a round tube. They developed an empirical correlation, which presented the experimental data within $\pm 20\%$. Liu et al. (2014, 2015, 2017) experimentally studied transient heat transfer for helium gas flowing over test heater with various shapes which are horizontal plates, twisted plates and cylinders. Shibahara et al. (2017) carried out experiments and measured transient heat transfer coefficients for water flowing in a narrow tube. The experimental data indicated that the Nusselt number is affected by Fourier number. Li et al. (2017) conducted experiments to elucidate transient heat transfer of FC-72 in small diameter tubes and derived an empirical correlation of transient turbulent heat transfer. Chavagnat et al. (2021) presented an experimental and theoretical investigation of single-phase forced convection transient heat transfer with water under exponentially escalating heat inputs. They developed an analytical model that can predict well of their database.

However, there are few experimental studies of transient forced convection heat transfer of helium gas through a narrow tube. In this research, fundamental experiments on the transient heat transfer process of helium gas using a tube with an inner diameter of 1.8 mm were conducted under various experimental conditions such as flow velocity and e-folding time of the heat generation rate. The lengths of the tube were 30 mm and 50 mm. The experimental data were compared with the data of our previous work (Xu et al., 2021a) for a length of 90 mm, and the effect of L/d on heat transfer was investigated. Furthermore, the relation of transient Nusselt number with Fourier number was discussed.

2. Nomenclature

a	thermal diffusivity of helium gas, m^2/s
A	inner surface area of the test tube, m^2
$c_{p,g}$	specific heat at constant pressure of helium gas evaluated by bulk temperature, J/(kg·K)
c_t	specific heat of the test tube, J/(kg·K)
d	inner diameter of the test tube, m
Fo	Fourier number
h	heat transfer coefficient, W/(m^2·K)
I	current through a standard resistor, A
L	length of the test tube, m
Nu_{st}	steady-state Nusselt number
Nu_{tr}	transient Nusselt number
P_{ipt}	pressure at the upstream pressure transducer, kPa
P_{in}	inlet pressure, kPa
P_{opt}	pressure at the downstream pressure transducer, kPa
P_{out}	outlet pressure, kPa
Pr	Prandtl number
q	heat flux, W/m^2
Q	heat generation rate, W
$\dot{Q}$	heat generation rate per unit volume, W/m^3
Q_0	initial heat generation rate per unit volume, W/m^3
r	radius of the test tube, m
r_i	inner radius of the test tube, m
r_o	outer radius of the test tube, m
R_1, R_2, R_3	resistance in a double-bridge circuit, Ω
R_s	standard resistance, Ω
R_T	resistance of the test tube, Ω
R_0	resistance of the test tube at 0 °C, Ω
Re	Reynolds number
t	time, s
T	temperature, K
T_a	average temperature of the test tube, °C
T_g	bulk temperature of helium gas, K
T_{in}	inlet temperature of helium gas, K
T_{out}	outlet temperature of helium gas, K

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T_s	inner surface temperature of the test tube, K
ΔT	temperature difference between inner surface temperature and bulk temperature, K
u	inlet flow velocity, m/s
V	volume of the test tube, m^3
V_I	voltage difference of a standard resistor, V
V_R	voltage difference of the test tube, V
V_T	unbalanced voltage difference, V
α	constant of Eq. (3)
β	constant of Eq. (3)
λ	thermal conductivity of helium gas, W/(m·K)
λ_t	thermal conductivity of the test tube, W/(m·K)
ν	kinematic viscosity, m^2/s
ρ_g	density of helium gas evaluated by bulk temperature, kg/m^3
ρ_t	density of the test tube, kg/m^3
τ	e-folding time, s

3. Experimental apparatus and methods
3.1 Experimental apparatus

A schematic diagram of the experimental apparatus is shown in Fig. 1, which was reported in our previous works (Xu et al., 2021a, 2021b). The gas is supplied from a gas cylinder and circulated by a compressor. The pulsation and the pressure fluctuation due to the compressor are removed by surge tanks. The flow rate in the test section is measured by a mass flow meter. The gas exiting the test section is cooled by a cooler and returns to the compressor.

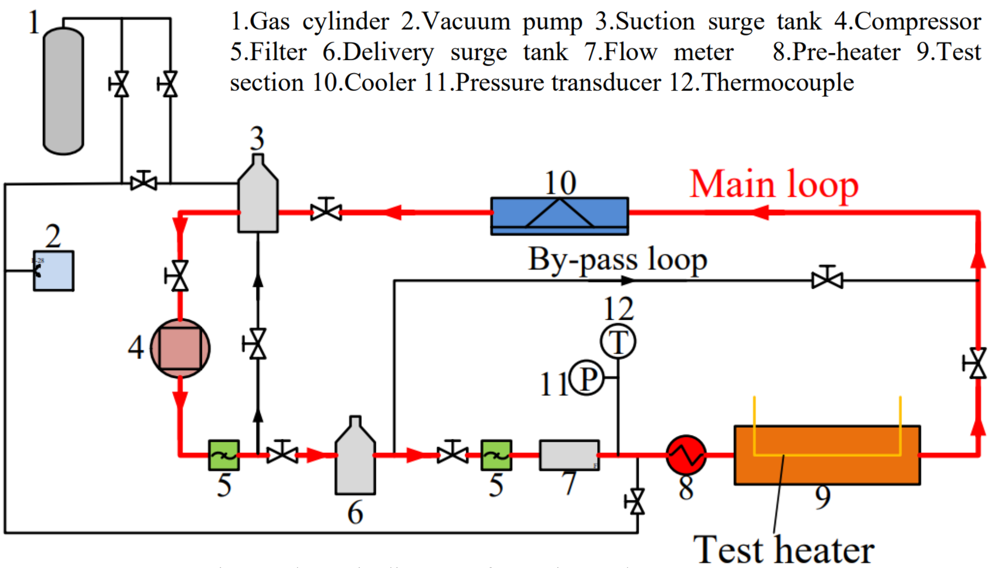

Fig. 1 Schematic diagram of experimental apparatus.

A detailed view of the test section is shown in Fig. 2. A platinum tube is used as the test heater and copper electrodes are connected to both ends of the heater and fixed. The tube is covered with Bakelite plates to insulate thermally and electrically. The silicon sheets were put in between the Bakelite plates and the electrodes to prevent gas leakage from gaps.

The method for data measurement was reported in previous works (Xu et al., 2021a, 2021b). A schematic diagram of the measurement circuit is shown in Fig. 3. The test section is incorporated into one side of the double-bridge circuit used for the measurement. Prior to starting the experiment, electrical equilibrium is established at the fluid temperature for each experiment. The unbalanced voltage difference V_T, the voltage difference V_R of the test tube, and the voltage difference V_I of the standard resistor are sufficiently amplified by a DC amplifier. The data are measured as digital values in a computer via A/D converter.

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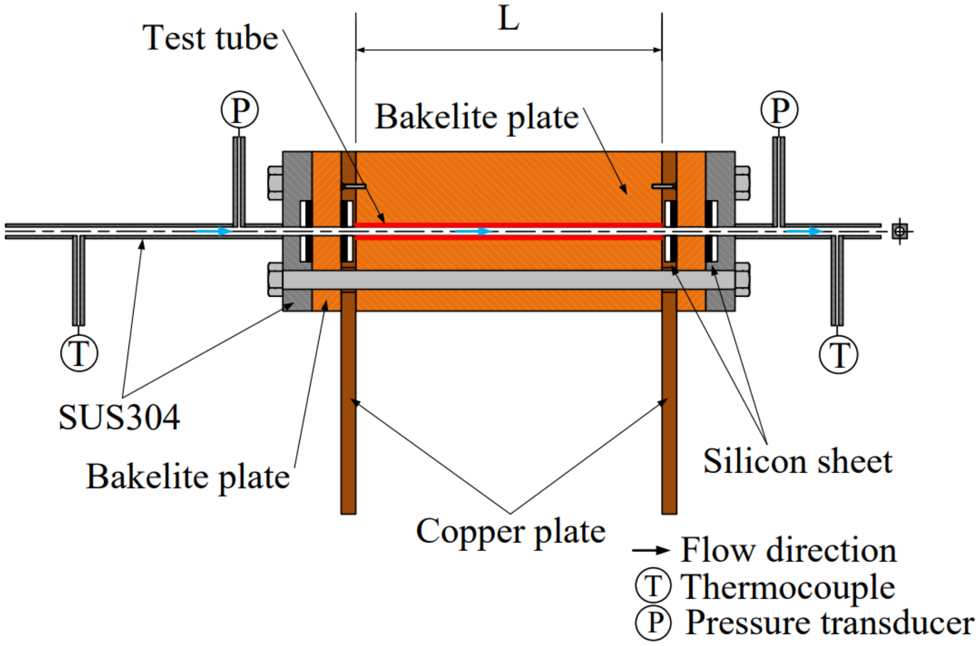

Fig. 2 Detailed view of the test section.

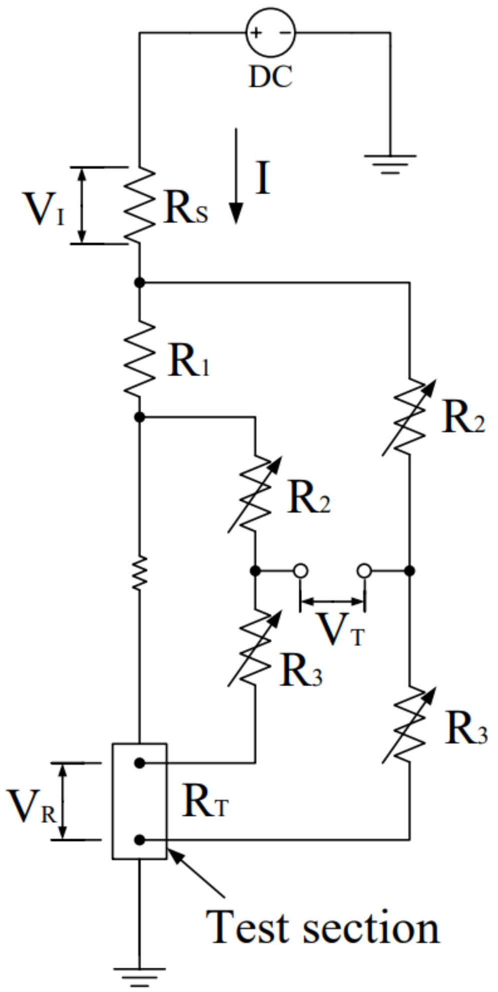

Fig. 3 Schematic diagram of the measurement circuit.

3.2 Measurement method

The heat generation rate Q due to Joule heating is described as follows:

$$Q = V_R I = V_R \frac{V_I}{R_s} \tag{1}$$

When the experiment is started and power is applied, the temperature of the test tube rises and double-bridge circuit becomes unbalanced. The resistance R_T of the test tube under measurement is given by the following equation:

$$R_T = \frac{V_T(R_2 + R_3)}{IR_2} + \frac{R_1 R_3}{R_2} \tag{2}$$

The average temperature T_a of the test tube is calculated from the relationship between resistance and temperature, which is calibrated in advance in a thermostatic chamber. The expression is as follows:

$$R_T = R_0(1 + \alpha T_a + \beta T_a^2) \tag{3}$$

The heat flux q from the surface of the test tube is calculated by the following equation from the energy balance:

$$q = \frac{V}{A}\left(\dot{Q} - \rho_t c_t \frac{dT_a}{dt}\right) \tag{4}$$

The inner surface temperature T_s of the test tube is acquired by solving the following unsteady-state heat conduction equation:

$$\rho_t c_t \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(r\lambda_t \frac{\partial T}{\partial r}\right) + \dot{Q} \tag{5}$$

The boundary conditions are given by:

$$q = -\lambda_t \left.\frac{\partial T}{\partial r}\right|_{r=r_i} \tag{6}$$

$$\left.\frac{\partial T}{\partial r}\right|_{r=r_o} = 0 \tag{7}$$

$$T_a = \frac{1}{\pi(r_o^2 - r_i^2)}\int_{r_i}^{r_o} 2\pi r T(r)dr \tag{8}$$

The inlet and outlet pressures of the test tube can be determined using values obtained by pressure transducers located 40 mm away from the inlet and outlet, respectively. The expressions are as follows:

$$P_{in} = P_{ipt} - \left(P_{ipt} - P_{opt}\right) \times \frac{0.04}{0.08 + L} \tag{9}$$

$$P_{out} = P_{in} - \left(P_{in} - P_{opt}\right) \times \frac{L}{0.04 + L} \tag{10}$$

The thermocouple that measures the outlet temperature is located 60 mm downstream from the actual test tube outlet, and there is a heat loss from the outlet to that position. Therefore, the outlet gas temperature T_{out} is calculated by the energy balance expressions as follows:

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$$T_{out} = T_{in} + \frac{4Lq}{udc_{p,g}\rho_g} \tag{11}$$

The physical properties of the fluid in this experiment are acquired by the bulk temperature T_g as follows:

$$T_g = \frac{T_{in} + T_{out}}{2} \tag{12}$$

The estimated errors for the heat generation rate, heat flux, and inner surface temperature of the test tube were within $\pm 2\%$, $\pm 2.4\%$, and $\pm 1K$, respectively (Xu et al., 2021a).

The heat transfer coefficient h is calculated by using the temperature difference ΔT between inner surface temperature and bulk temperature as follows:

$$\Delta T = T_s - T_g \tag{13}$$

$$h = \frac{q}{\Delta T} \tag{14}$$

3.3 Experimental procedure

The experiment is performed in the following steps. First, the loop is filled with helium gas from a gas cylinder after the inside of the loop is sufficiently evacuated by a vacuum pump. The helium gas is circulated by the compressor, the gas flow velocity, pressure, and temperature are adjusted to the desired values. The flow velocity is adjusted by the bypass valve. The pressure can be set to a desired value by the amount of helium gas filled from the gas cylinder. The temperature is adjusted to a desired value by the preheater. After these values become stable, the test tube is subjected to a heat generation rate $\dot{Q}$ that increases exponentially with time as follows:

$$\dot{Q} = Q_0 \exp\left(\frac{t}{\tau}\right) \tag{15}$$

where, Q_0 is initial heat generation rate, t is time, and τ is e-folding time.

The experimental data were taken under conditions as follows: the e-folding time was from 40 ms to 15 s; the inlet gas temperatures were in the range of 303-304 K; the inlet gas pressures were in the range of 456-501 kPa; and the flow velocities were in the range of 101-256 m/s.

4. Results and discussion

4.1 Time dependence of heat generation rate, heat flux, and inner surface temperature

Figure 4 shows time dependence of heat generation rate $\dot{Q}$, heat flux q, and inner surface temperature T_s. The heat generation rate of the tube is exponentially increased and rises sharply as the e-folding time shortens. Heat flux and inner surface temperature also increase exponentially with the increase of the heat generation rate and increase more rapidly while the e-folding time is shorter.

4.2 Heat transfer in quasi-steady state and transient state

Figure 5 shows the relation of heat transfer coefficient h with e-folding time τ at various flow velocities for the tube lengths of 30 mm and 50 mm. The same phenomenon is observed for both 30 mm and 50 mm tube lengths. The heat transfer coefficients are approximately constant for the e-folding time longer than about 1.5 s under the same flow velocity. The heat transfer coefficients gradually increase for the e-folding time shorter than about 1.5 s. These results indicate that the heat transfer process is in a quasi-steady state for the e-folding time longer than about 1.5 s, and it is in a transient state for the e-folding time shorter than about 1.5 s.

In the quasi-steady state, the thermal boundary layer is fully developed and forced convection heat transfer is dominant in the heat transfer process. On the other hand, in the transient state, the wall temperature rise occurs faster than the thermal boundary layer development. And the effect of heat conduction contribution gradually increases with the decrease of e-folding time.

Figure 5 also shows that the heat transfer coefficients increase with the increase of flow velocity for each e-folding

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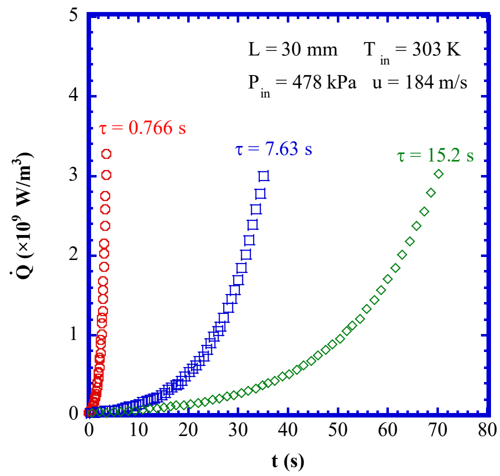
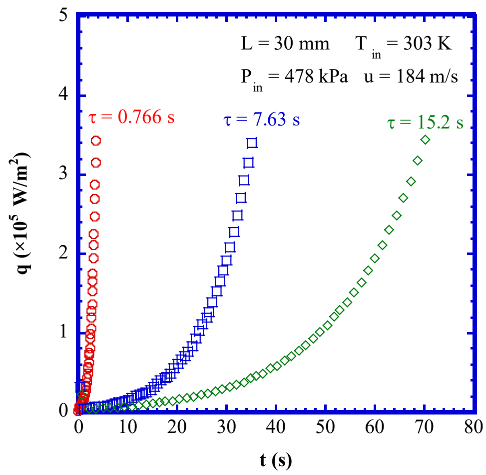
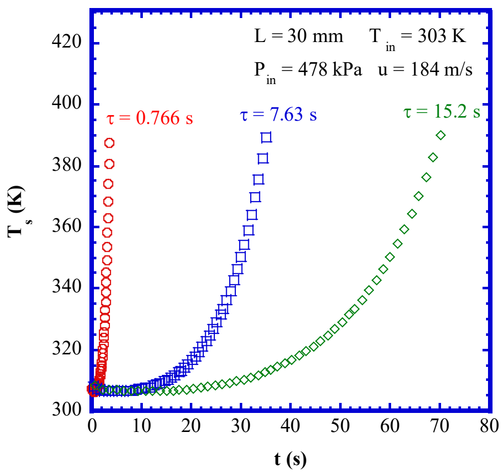
time. However, the effect of the flow velocity becomes weak in the transient state region. When the flow velocities raise from 101 to 208 m/s for the length of 30 mm, the heat transfer coefficients are raised by 64% at the e-folding time of 15.3 s and the increment is 39% at the e-folding time of 0.04 s. This trend can be considered that the heat transfer process in this region is governed by conductive heat transfer rather than the convective heat transfer.



(a) Heat generation rate $\dot{Q}$

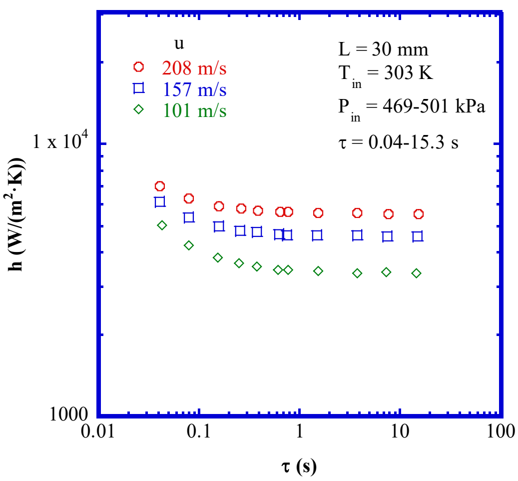


(b) Heat flux q

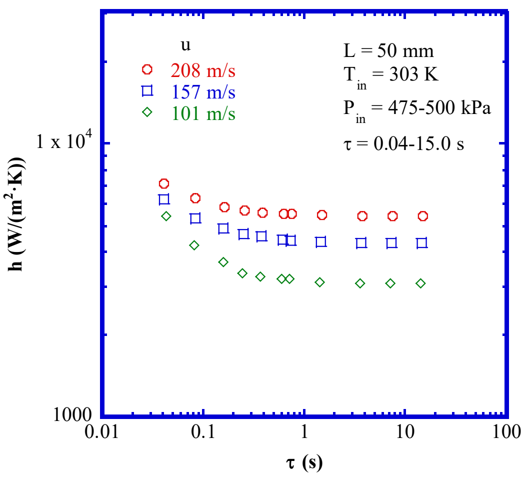


(c) Inner surface temperature T_s

Fig. 4 Time dependence of heat generation rate, heat flux, and inner surface temperature.



(a) $L = 30\ mm$



(b) $L = 50\ mm$

Fig. 5 The relation of heat transfer coefficient with e-folding time at various flow velocity for the tube lengths of 30 mm and 50 mm.

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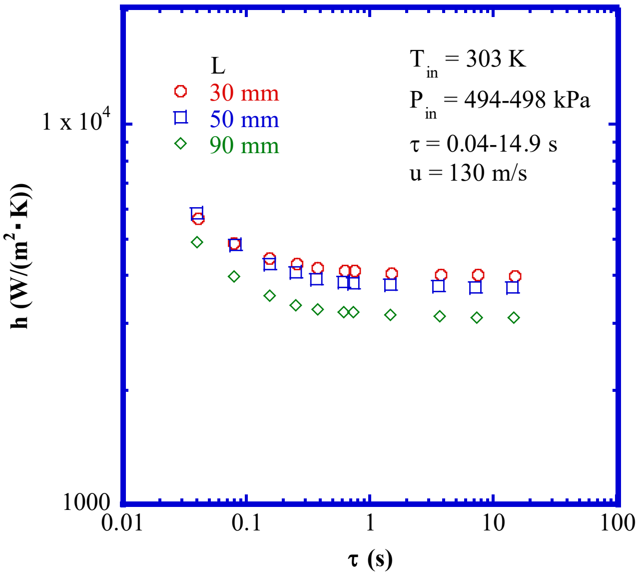
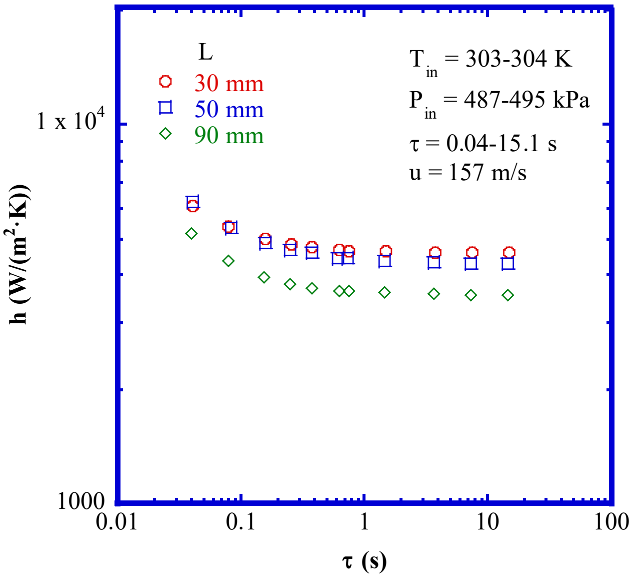
4.3 Effect of tube length on heat transfer

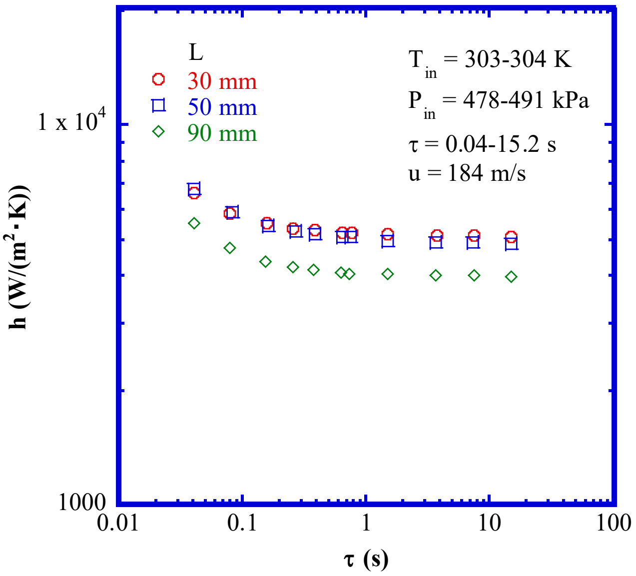
Figure 6 shows the relation of heat transfer coefficient h with e-folding time τ at various tube lengths for typical flow velocities. By comparing the experimental data at different lengths, it is obtained that quasi-steady state heat transfer coefficients for the length of 30 mm are higher than those for the lengths of 50 mm and 90 mm at the respective flow velocities. When the length of tube decreases from 90 to 30 mm, the heat transfer coefficient is raised by 29% at the e-folding time of 15.1 s and velocity of 157 m/s. It is considered that the entrance effect due to the reduction in tube length affect the heat transfer (Sparrow et al., 1957). However, the tube length showed a weak influence for the lengths of 30 mm and 50 mm in the transient heat transfer region with a relatively smaller e-folding time. The entrance effect is considered to have little effect on conductive heat transfer.



(a) $u = 130\ m/s$



(b) $u = 157\ m/s$



(c) $u = 184\ m/s$

Fig. 6 The relation of heat transfer coefficient with e-folding time at various tube lengths.

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4.4 An empirical correlation for steady state heat transfer in a narrow tube

In our previous study, Xu et al. (2021b) reported the following correlation for steady state heat transfer of helium gas flowing in a narrow tube with a diameter of 1.8 mm and a length of 90 mm.

$$Nu_{st} = 0.0333Re^{0.8}Pr^{0.4}\left(T_s/T_g\right)^{-0.5}$$
$$(d\text{: 1.8 mm; } L\text{: 90 mm; } Re\text{: } 5000-16000\text{; deviation: } \pm 10\%) \tag{16}$$

where, $Nu_{st}\ (= hd/\lambda)$ is steady-state Nusselt number, $Pr\ (= \nu/a)$ is Prandtl number, and $Re\ (= ud/\nu)$ is Reynolds number.

It is based on the classical Dittus-Boelter expression (1930), and the term of $\left(T_s/T_g\right)^{-0.5}$ takes into account changes in thermal physical properties such as viscosity due to changes in temperature. Based on the experimental data of this study, an empirical correlation for steady state heat transfer of helium gas through a narrow tube considering L/d with reference to Eq. (16) is obtained as follows:

$$Nu_{st} = 0.0682Re^{0.8}Pr^{0.4}(L/d)^{-0.18}\left(T_s/T_g\right)^{-0.5}$$
$$(d\text{: 1.8 mm; } L\text{: 30,50,90 mm; } Re\text{: } 6000-15000\text{; deviation: } \pm 10\%) \tag{17}$$

The experimental data under steady state are plotted in Fig. 7, and the solid line represents values by Eq. (17). The deviation between the experimental data and the correlation is within $\pm 10\%$.

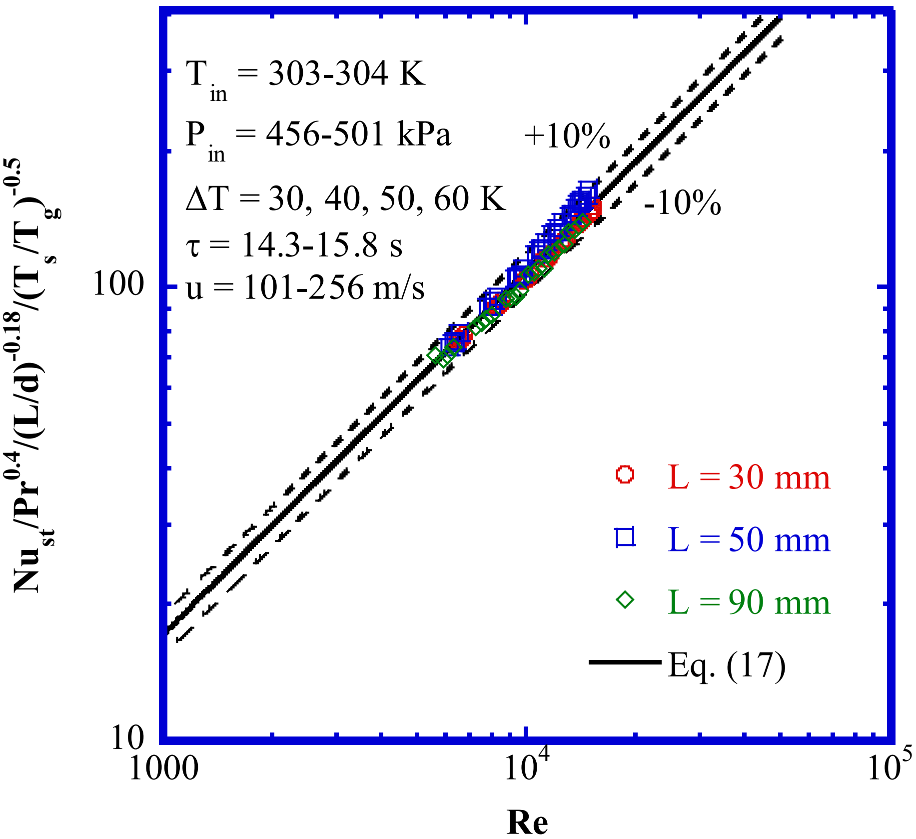

Fig. 7 Empirical correlation for steady state of helium gas through a narrow tube.

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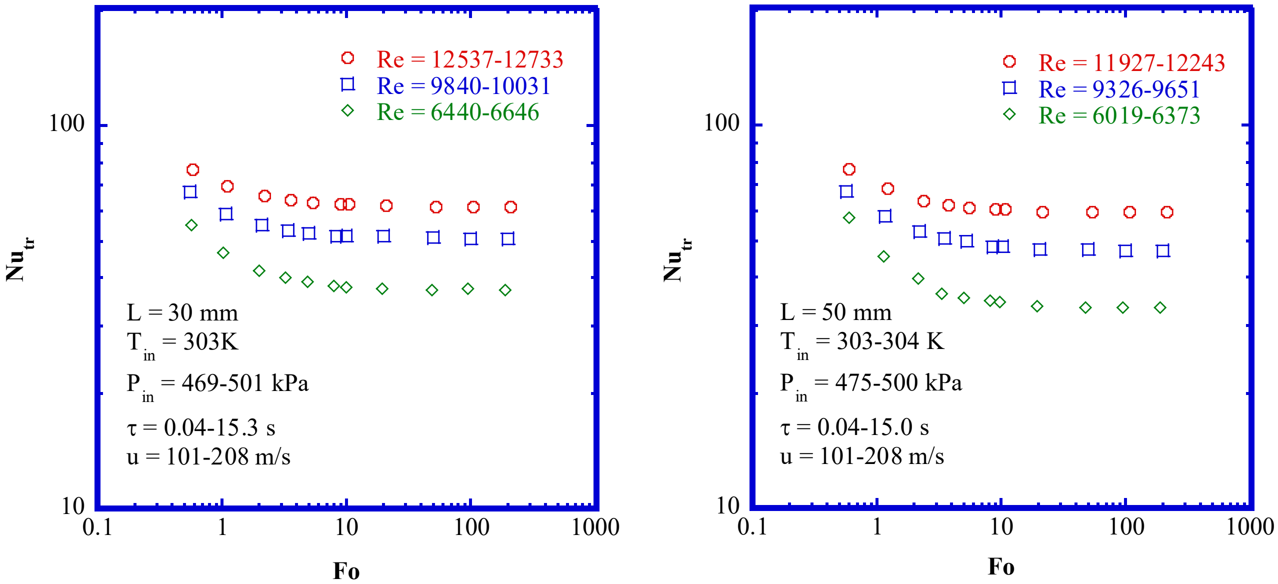
4.5 An empirical correlation for transient state heat transfer in a narrow tube

In order to investigate the transient heat transfer characteristics for forced convection, a non-dimensional parameter of Fourier number Fo is used as follows:

$$Fo = \frac{a\tau}{d^2} \tag{18}$$

Figure 8 shows the relation of transient Nusselt number Nu_{tr} with Fourier number Fo at different ranges of Reynolds number Re for the tube lengths of 30 mm and 50 mm. At each Reynolds number, Nusselt number decreases with the increase in Fourier number and approaches an asymptotic value. As shown in Fig. 8, the heat transfer process can be divided into transient state and quasi-steady state at the Fourier number of near 20.



(a) $L = 30\ mm$ (b) $L = 50\ mm$

Fig. 8 The relation of Nusselt number with Fourier number at different ranges of Reynolds number for the tube lengths of 30 mm and 50 mm.

Based on the experimental data, an empirical correlation for transient heat transfer of helium gas through a narrow tube with different lengths is obtained as follows:

$$Nu_{tr}/Nu_{st} = 1 + 0.187Fo^{-1.5}$$
$$(d\text{: 1.8 mm}; L\text{: 30,50,90 mm}; Re\text{: } 6000 - 15000; \text{deviation: } \pm 25\%) \tag{19}$$

Figure 9 shows the relation between ratio of transient Nusselt number Nu_{tr} to steady state Nusselt number Nu_{st} and Fourier number Fo at various tube lengths. The solid line represents Eq. (19). The correlation can express the experimental data within $\pm 25\%$.

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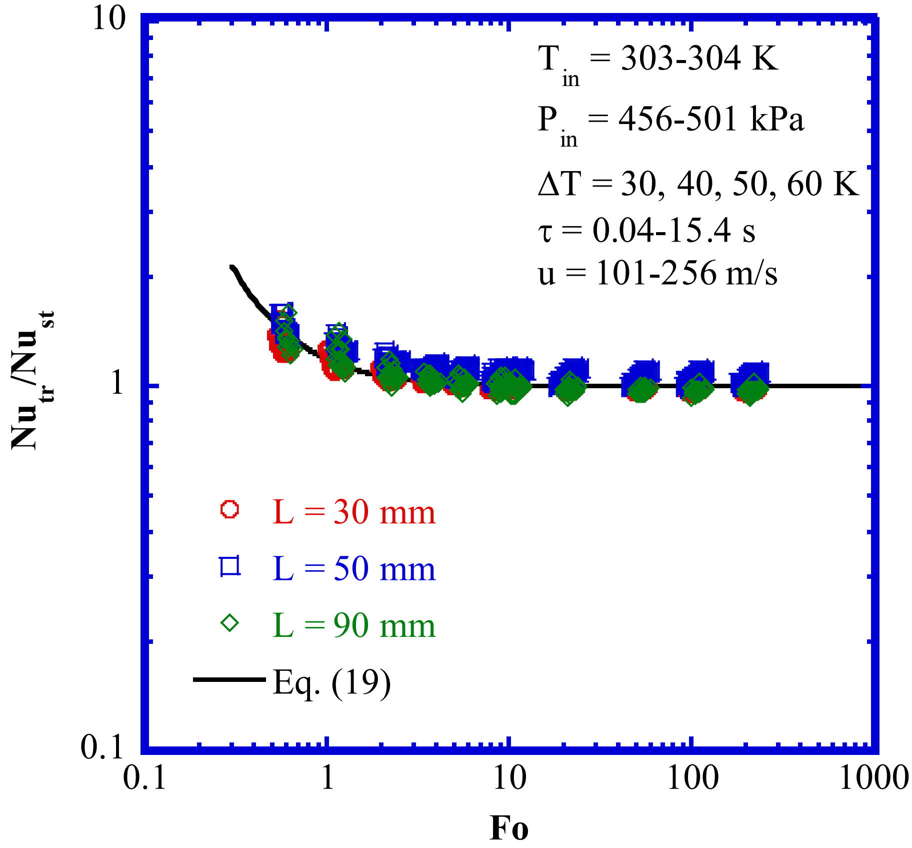

Fig. 9 The relation between ratio of transient Nusselt number to steady state Nusselt number and Fourier number at various tube lengths.

5. Conclusions

Experiments were carried out for the transient heat transfer of helium gas through a narrow tube with various lengths. The heat transfer coefficients gradually increased for the e-folding time shorter than about 1.5 s. This shows that the heat transfer process is in a transient state for this region. The heat transfer coefficients increased with the increase of flow velocity, and the effect of the flow velocity became weak in the transient state region. It was obtained that the quasi-steady state heat transfer coefficients for the length of 30 mm were higher than those for the lengths of 50 mm and 90 mm. However, the tube length showed a weak influence for the lengths of 30 mm and 50 mm in the transient heat transfer with a relatively smaller e-folding time. The variation of transient Nusselt number with Fourier number at different Reynolds number describes well the trend for the variation of heat transfer coefficient with e-folding time at different velocities. Based on the experimental data, an empirical correlation was obtained for transient heat transfer at different lengths by using a non-dimensional parameter of Fourier number.

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