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# CONDUCTIVE SUBLAYER OF TURBULENT HEAT TRANSFER FOR HEATING OF WATER IN A CIRCULAR TUBE

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## ABSTRACT

The steady-state and transient turbulent heat transfer coefficients in circular platinum (Pt) test tubes (inner diameters: 3 and 6 mm; heated lengths: 66.5 and 100 mm and 69.6 mm, respectively) were systematically measured using an experimental water loop for a wide range of flow velocities, inlet liquid temperatures, Prandtl numbers, inlet pressures, and exponentially increasing heat inputs ( $Q_{exp}(t/\tau)$ ,  $\tau$ : exponential period). The Reynolds-averaged Navier–Stokes equations and the  $k$ – $\varepsilon$  turbulence model for unsteady turbulent heat transfer in circular test sections were numerically solved for heating of water with heated sections of diameter 3 and 6 mm and length 67 and 100 mm and 70 mm, respectively, by using computational fluid dynamics code under the same conditions as those in the experiment and with temperature-dependent thermophysical fluid properties. The thickness of the conductive sublayer,  $\delta_{CSL,st}$  and  $\delta_{CSL}$  [ $=(\Delta r)_{out}/2$ ], and the nondimensional thickness of the conductive sublayer,  $(y^+_{CSL,st})_{TEM}$  [ $=(f_F/2)^{0.5} \rho_l u \delta_{CSL,st} / \mu_l$ ] and  $(y^+_{CSL})_{TEM}$  [ $=(f_F/2)^{0.5} \rho_l u \delta_{CSL} / \mu_l$ ], for steady-state and transient turbulent heat transfer at various heated length-to-inner diameter ratios, inlet liquid temperatures, and exponential periods were measured on the basis of the numerical solutions. The correlations of the thickness of the conductive sublayer,  $\delta_{CSL,st}$  and nondimensional thickness of the conductive sublayer,  $(y^+_{CSL,st})_{TEM}$ , for steady-state turbulent heat transfer and those of the thickness of the conductive sublayer,  $\delta_{CSL}$ , and nondimensional thickness of the conductive sublayer,  $(y^+_{CSL})_{TEM}$ , for transient turbulent heat transfer in a circular tube were derived.

## KEYWORDS

Conductive Sublayer, Turbulent Heat Transfer, Heating of Water, Circular Tube

## INTRODUCTION

The thickness of the conductive sublayer for steady-state and transient turbulent heat transfer for heating of water in a circular tube is critical in gaining a comprehensive understanding of turbulent heat transfer in pipes and in the design of diverters in nuclear fusion facilities. However, few fundamental studies have investigated the thickness of conductive sublayers, and little is known on the influence of heated length, inlet liquid temperature, and heating rate on the thickness of conductive sublayers. Knowledge of the thickness of conductive sublayers and nondimensional thickness of conductive sublayers would help advance the computational fluid dynamics (CFD) code for analyzing steady-state and transient turbulent heat transfer [1-5].

The objectives of this study are to (1) measure the steady-state and transient turbulent heat transfer coefficients of three sections (upper, mid, and lower sections) of platinum (Pt) test tubes of inner diameters 3 and 6 mm for a wide range of heated lengths ( $L$ ), inlet liquid temperatures ( $T_{in}$ ), exponential periods ( $\tau$ ), test tube surface temperatures ( $T_s$ ) and flow velocities ( $u$ ); (2) obtain the numerical solutions for surface heat flux ( $q$ ) and surface temperature ( $T_s$ ) of the heated length, liquid temperature ( $T_l$ ), and flow velocity ( $u$ ) of the aforementioned test tubes by using Reynolds-averaged Navier–Stokes (RANS) equations and the  $k$ – $\varepsilon$  turbulence model for unsteady turbulent heat transfer under the same conditions as those in the experiment; (3) compare the numerically and experimentally obtained local surface heat flux ( $q$ ), local surface temperature [ $(T_s)_z$ ], and local average liquid temperature [ $(T_l)_z$ ]; (4) measure the thickness of the conductive sublayer,  $\delta_{CSL}[(Ar)_{out}/2]$ , and the nondimensional thickness of the conductive sublayer,  $y^+_{CSL}[(f_F/2)^{0.5}\rho_l u \delta_{CSL}/\mu_l]$ , for the steady-state and transient turbulent heat transfer in Pt test tubes of inner diameter 3 and 6 mm; and (5) discuss the effects of heated length-to-inner diameter ratio and inlet liquid temperature on the thickness of the conductive sublayer,  $\delta_{CSL, st}$ , and the nondimensional thickness of conductive sublayer,  $y^+_{CSL, st}$ , for steady-state turbulent heat transfer as well as those of exponential period on the thickness of the conductive sublayer,  $\delta_{CSL}$ , and the nondimensional thickness of the conductive sublayer,  $y^+_{CSL}$ , for transient turbulent heat transfer in a vertical circular tube.

## EXPERIMENTAL APPARATUS AND METHOD

### Experimental Setup

The experimental setup comprised a SUS304 stainless steel water loop and a pressurizer (Fig. 1). The loop can withstand pressures of up to 2 MPa and has five vertically oriented test sections of inner diameter 2, 3, 6, 9, and 12 mm, with water flowing upward in all sections. Test sections of inner diameter 3 and 6 mm were used in this study. The circulating water was distilled and deionized to a specific resistivity of approximately 20  $\mu\text{S/m}$ . The water circulating through the loop was heated or cooled using a preheater or cooler to maintain the desired inlet temperature. Flow velocity

was measured using a mass flow meter and a vibration tube (Nitto Seiko, CLEANFLOW 63FS25, flow range: 100–750 kg/min). Mass velocity was controlled by regulating the frequency of the three-phase alternating-power source of the nonseal multistage canned-type circulation pump (Nikkiso Co., Ltd., VNH12-C4 C-3S7SP; flow rate: 12 m<sup>3</sup>/h, pump head: 250 m) equipped with an inverter and a four-digit LED monitor (F720-30K, Mitsubishi Electric Corp.). The output of the pump input frequency indicates the outputs of the net pump input power and pump discharge pressure free of slip loss, and they are low if it is smaller and high if it is larger. The water was pressurized using saturated vapor in the pressurizer, and the pressure at the outlet of the test tube was controlled within  $\pm 1$  kPa of the desired value by using the heating controller of the pressurizer.

### Test Sections

Table 1 lists the dimensions of the finish Pt test tubes and test sections, namely THD-F167, THD-F174, and THD-F189 (see Fig. 2 for cross-section) with commercial-finish inner surfaces used in this study. Pt test tubes are adequately sensitive for resistance thermometry. Four fine 0.07-mm diameter Pt wires were spot-welded on the outer surface of THD-F167 and THD-F189 as potential taps: the first taps are welded 4.8 and 5.2 mm from the leading edge of these two test tubes, respectively, and the second to fourth taps are 19.2 and 19.6, 18.8 and 20.0, and 18.7 and 19.6 mm from the immediately preceding taps, respectively. The effective lengths,  $L_{eff}$ , of the 3- and 6-mm inner diameter test tubes between the first and fourth taps on which the average heat transfer was measured were 56.7 and 59.2 mm, respectively, for THD-F167 and THD-F189. The 5-mm-thick silver-coated copper (Cu) electrode plates used to supply heating current were soldered to the surfaces of both ends of the test tubes; both ends were electrically isolated from the loop by using 14-mm-thick bakelite plates. The inner surface of the test tubes was observed through scanning electron microscopy (SEM; JEOL JXA8600; see Fig. 3 for SEM images), and the inner surface roughness was measured using a surface texture measuring instrument (Surfcom 120A, Tokyo Seimitsu Co., Ltd.). Inner surface roughness ( $Ra$ ) and its maximum ( $Rmax$ ) and mean ( $Rz$ ) were measured as 0.40, 2.20, and 1.50  $\mu\text{m}$  for the  $d = 3$  mm test tube and 0.45, 2.93, and 1.93  $\mu\text{m}$  for the  $d = 6$  mm test tube, respectively.

### Measurement and Data Processing System

Pt test tubes were heated with an exponentially increasing heat input supplied from a direct-current source (NL035-500R, DC 35 V-3000 A, Takasago Ltd.) through two Cu electrodes (Fig. 4), and the resulting heat transfer,  $Q_{0exp}(t/\tau)$ , was measured. Exponential periods,  $\tau$ , of the heat input ranged from 15.43 ms to 26.98 s. Specifications of the direct-current source are as follows: constant-voltage (CV)-mode regulation = 4.75 mV (minimum), CV-mode ripple = 500  $\mu\text{V}$  root mean

square or higher, and typical CV-mode transient response time = <200  $\mu$ s against 5% to full-range change of load. The transient critical heat fluxes,  $q_{cr,sub}$ , were realized by the exponentially increasing heat input to the test tube. The outer surface temperature of the test tube, which was uniformly painted black using blackbody spray (JSC-3, Japan Sensor Corporation; emissivity,  $\varepsilon$ : 0.94), was monitored using an infrared thermal imaging camera (TVS-200EX, NEC Avio Infrared Technologies Co., Ltd.) of accuracy  $\pm 2\%$ .

The average temperature,  $\bar{T}$ , of each Pt test tube was measured through resistance thermometry by using a branch of a double-bridge circuit (Fig. 4). The output voltages from the bridge circuit,  $V_T$ , together with the voltage drop across the potential taps of each test tube (first and fourth taps:  $V_R = IR_T$ ; first and second taps:  $V_{R1} = IR_{T1}$ ; second and third taps:  $V_{R2} = IR_{T2}$ ; third and fourth taps:  $V_{R3} = IR_{T3}$ ) and across a standard resistance,  $V_I = IR_s$ , were amplified and transmitted through an analog–digital converter to a digital computer. Unbalance voltage,  $V_T$ , can be expressed using Ohm's law as

$$V_T = \frac{I(R_T \times R_2 - R_1 \times R_3)}{R_2 + R_3} \quad (1)$$

$V_T$  was simultaneously sampled at a constant interval ranging from 200  $\mu$ s to 200 ms. The average temperature of each Pt test tube between the first and fourth potential taps and between adjacent potential taps (i.e., first and second, second and third, and third and fourth taps) were calculated using a previously calibrated resistance–temperature relationship:  $R_T = a(1 + b\bar{T} + c\bar{T}^2)$ . The average temperature between the two electrodes,  $V_{R4}$ , were compared with those between the first and fourth potential taps,  $V_R$ ; the difference (i.e., heat loss) was not large in the high subcooling range. The heat-generation rates of the Pt test tubes between the first and fourth potential taps,  $Q = I^2 R_T$ , and between adjacent taps (first and second:  $Q_1 = I^2 R_{T1}$ ; second and third:  $Q_2 = I^2 R_{T2}$ ; third and fourth:  $Q_3 = I^2 R_{T3}$ ) were calculated from the measured voltage difference between the first and fourth taps and between the adjacent taps of the Pt test tubes,  $V_R$ ,  $V_{R1}$ ,  $V_{R2}$ , and  $V_{R3}$ , and that across the standard resistance,  $V_I$ . Surface heat fluxes between the first and fourth taps and between adjacent taps,  $q$ ,  $q_1$ ,  $q_2$ , and  $q_3$ , are the differences between the heat-generation rate per unit surface area,  $Q$ ,  $Q_1$ ,  $Q_2$ , and  $Q_3$ , and the rate of change of energy storage in the Pt test tubes and can be obtained using the fair-averaged temperature versus time curve:

$$q = \frac{V}{S} \left( Q - \rho c \frac{d\bar{T}}{dt} \right) \quad (2)$$

where  $\rho$ ,  $c$ ,  $V$ , and  $S$  are the density, specific heat, volume, and inner surface area of the test tube, respectively.

The difference in heater inner surface temperatures between the first and fourth potential taps and between adjacent taps,  $T_s$ ,  $T_{s1}$ ,  $T_{s2}$  and  $T_{s3}$ , were obtained by solving the steady one-dimensional heat conduction equation for the test tube under the conditions of measured average temperature,  $\bar{T}$ , and surface heat flux of the test tube,  $q$ . The inner and outer

surface temperatures of the test tube,  $T_s$  and  $T_{so}$ , can be obtained using the steady one-dimensional heat conduction equation.

The basic equation for the test tube is

$$\frac{d^2T}{dr^2} + \frac{1}{r} \frac{dT}{dr} + \frac{Q}{\lambda} = 0 \quad (3)$$

Integrating Eq. (3) yields the mean temperature of the test tube:

$$T(r) = -\frac{Qr^2}{4\lambda} + \frac{Qr_o^2}{2\lambda} \ln r + C \quad (4)$$

$$\bar{T} = \frac{1}{\pi(r_o^2 - r_i^2)} \int_{r_i}^{r_o} 2\pi r T(r) dr \quad (5)$$

Assuming that the test tube is perfectly insulated, heat generation in the tube equals heat conduction:

$$q = -\lambda \left. \frac{dT}{dr} \right|_{r=r_i} = \frac{(r_o^2 - r_i^2)Q}{2r_i} \quad (6)$$

$$\left. \frac{dT}{dr} \right|_{r=r_o} = 0 \quad (7)$$

The temperatures of the heater inner and outer surfaces,  $T_s$  and  $T_{so}$ , and  $C$  in Eq. (4) can be described as follows:

$$T_s = T(r_i) = \bar{T} - \frac{qr_i}{4(r_o^2 - r_i^2)^2 \lambda} \left[ 4r_o^2 \left\{ r_o^2 \left( \ln r_o - \frac{1}{2} \right) - r_i^2 \left( \ln r_i - \frac{1}{2} \right) \right\} - (r_o^4 - r_i^4) \right] - \frac{qr_i}{2(r_o^2 - r_i^2) \lambda} (r_i^2 - 2r_o^2 \ln r_i) \quad (8)$$

$$T_{so} = T(r_o) = \bar{T} - \frac{qr_i}{4(r_o^2 - r_i^2)^2 \lambda} \left[ 4r_o^2 \left\{ r_o^2 \left( \ln r_o - \frac{1}{2} \right) - r_i^2 \left( \ln r_i - \frac{1}{2} \right) \right\} - (r_o^4 - r_i^4) \right] - \frac{qr_i r_o^2}{2(r_o^2 - r_i^2) \lambda} (1 - 2 \ln r_o) \quad (9)$$

$$C = \bar{T} - \frac{qr_i}{4(r_o^2 - r_i^2)^2 \lambda} \left[ 4r_o^2 \left\{ r_o^2 \left( \ln r_o - \frac{1}{2} \right) - r_i^2 \left( \ln r_i - \frac{1}{2} \right) \right\} - (r_o^4 - r_i^4) \right] \quad (10)$$

where  $\bar{T}$ ,  $q$ ,  $\lambda$ ,  $r_i$ , and  $r_o$  are the average temperature of the test tube, heat flux, thermal conductivity, and inner and outer radii of the test tube, respectively.

For the 3- and 6-mm inner diameter test sections, before entering the test tube, the test water is flown through the tube with the same inner diameter as that of the Pt test tube in order to ensure that the velocity profile is fully developed. The entrance lengths,  $L_e$ , are 282 and 333 mm; thus,  $L_e/d = 94$  and 55.5.  $L_e/d$  for the  $d = 3$  and 6 mm test tubes at which the centerline velocity reaches 99% of the maximum for turbulent flow were determined as 9.8–21.9 ( $1.50 \times 10^4 \leq Re_d \leq 3.08 \times 10^5$ ) by using the Brodkey and Hershey correlation [6]:

$$\frac{L_e}{d} = 0.693 Re_d^{1/4} \quad (11)$$

The inlet and outlet liquid temperatures,  $T_{in}$  and  $T_{out}$ , were measured using 1-mm outer diameter, sheathed, K-type thermocouples (Nimblox, sheath material: SUS316, hot junction: ground, response time (63.2%): 46.5 ms) located at the centerline of the tube at the upper and lower stream points 262 and 53 mm (283 and 63 mm) from the tube inlet and outlet for the 3-mm (6-mm) inner diameter test section. Inlet and outlet pressures,  $P_{ipt}$  and  $P_{opt}$ , were measured using strain gauge transducers (PHS-20A, Kyowa Electronic Instruments Co., Ltd.; natural frequency:  $\sim 30$  kHz) located near the entrance of the conduit at upper and lower stream points 53 mm (63 mm) from the tube inlet and outlet for the  $d = 3$  mm ( $d = 6$  mm) test section. The thermocouples and transducers were installed in the conduits as shown in Fig. 2.

The inlet and outlet pressures,  $P_{in}$  and  $P_{out}$ , for the 3- and 6-mm inner diameter test sections were calculated from the pressures measured using the inlet and outlet pressure transducers,  $P_{ipt}$  and  $P_{opt}$ :

$$P_{in} = P_{ipt} - \left\{ (P_{ipt})_{wnh} - (P_{opt})_{wnh} \right\} \times \frac{L_{ipt}}{L_{ipt} + L + L_{opt}} \quad (12)$$

$$P_{out} = P_{in} - (P_{in} - P_{opt}) \times \frac{L}{L + L_{opt}} \quad (13)$$

where  $L_{ipt} = 0.053$  and  $0.063$  m and  $L_{opt} = 0.053$  and  $0.063$  m for the 3- and 6-mm inner diameter test sections, respectively. Experimental errors in the inner tube surface temperature and heat flux were estimated to be  $\pm 1$  K and  $\pm 2\%$ , respectively. Mass velocity, inlet and outlet subcooling, inlet and outlet pressures, and exponential period were measured to an the accuracy of  $\pm 2\%$ ,  $\pm 1$  K,  $\pm 4$  kPa, and  $\pm 2\%$ , respectively.

## NUMERICAL SOLUTION OF TURBULENT HEAT TRANSFER

### RANS Equations for $k$ - $\varepsilon$ Turbulence Model with High Reynolds Number Form

The RANS equations for the  $k$ - $\varepsilon$  turbulence model [7] in a circular tube of inner diameter 3 mm and length 526 mm were numerically solved for heating of water in a heated section of inner diameter 3 mm and length 67 mm by using the PHOENICS CFD code under the same conditions as in the experiment and with temperature-dependent thermophysical fluid properties [8]. The unsteady RANS equations for turbulent heat transfer can be expressed in the three-dimensional coordinate system shown in Fig. 5 as follows [9]:

Continuity equation—cylindrical coordinates ( $r, \theta, z$ ):

$$\frac{\partial \rho}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r} (r u_r \rho) + \frac{1}{r} \frac{\partial}{\partial \theta} (u_\theta \rho) + \frac{\partial}{\partial z} (u_z \rho) = 0 \quad (14)$$

Momentum equation—cylindrical coordinates ( $r, \theta, z$ ):

*r*-component

$$\frac{\partial}{\partial t}(\rho u_r) + \frac{\partial}{\partial r}(\rho u_r u_r) + \frac{1}{r} \frac{\partial}{\partial \theta}(\rho u_\theta u_r) - \frac{\rho u_\theta^2}{r} + \frac{\partial}{\partial z}(\rho u_z u_r) = -\frac{\partial P}{\partial r} + \frac{1}{r} \frac{\partial}{\partial r}(r \tau_{rr}) + \frac{1}{r} \frac{\partial}{\partial \theta} \tau_{r\theta} - \frac{\tau_{\theta\theta}}{r} + \frac{\partial}{\partial z} \tau_{rz} + \rho g_r \quad (15)$$

$\theta$ -component

$$\frac{\partial}{\partial t}(\rho u_\theta) + \frac{\partial}{\partial r}(\rho u_r u_\theta) + \frac{1}{r} \frac{\partial}{\partial \theta}(\rho u_\theta u_\theta) + \frac{\rho u_r u_\theta}{r} + \frac{\partial}{\partial z}(\rho u_z u_\theta) = -\frac{1}{r} \frac{\partial P}{\partial \theta} + \frac{1}{r^2} \frac{\partial}{\partial r}(r^2 \tau_{r\theta}) + \frac{1}{r} \frac{\partial}{\partial \theta} \tau_{\theta\theta} + \frac{\partial}{\partial z} \tau_{\theta z} + \rho g_\theta \quad (16)$$

$z$ -component

$$\frac{\partial}{\partial t}(\rho u_z) + \frac{\partial}{\partial r}(\rho u_r u_z) + \frac{1}{r} \frac{\partial}{\partial \theta}(\rho u_\theta u_z) + \frac{\partial}{\partial z}(\rho u_z u_z) = -\frac{\partial P}{\partial z} + \frac{1}{r} \frac{\partial}{\partial r}(r \tau_{rz}) + \frac{1}{r} \frac{\partial}{\partial \theta} \tau_{\theta z} + \frac{\partial}{\partial z} \tau_{zz} + \rho g_z \quad (17)$$

Energy equation—cylindrical coordinates  $(r, \theta, z)$ :

$$\begin{aligned} \rho \frac{\partial h}{\partial t} + \rho \frac{\partial}{\partial r}(u_r h) + \rho \frac{1}{r} \frac{\partial}{\partial \theta}(u_\theta h) + \rho \frac{\partial}{\partial z}(u_z h) = \\ \frac{1}{r} \frac{\partial}{\partial r} \left\{ \left( \lambda + c_p \frac{\mu_t}{\sigma_t} \right) r \frac{\partial T}{\partial r} \right\} + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left\{ \left( \lambda + c_p \frac{\mu_t}{\sigma_t} \right) \frac{\partial T}{\partial \theta} \right\} + \frac{\partial}{\partial z} \left\{ \left( \lambda + c_p \frac{\mu_t}{\sigma_t} \right) \frac{\partial T}{\partial z} \right\} + Q \end{aligned} \quad (18)$$

Transport equation for  $k$ —cylindrical coordinates  $(r, \theta, z)$ :

$$\begin{aligned} \rho \frac{\partial k}{\partial t} + \rho \frac{\partial}{\partial r}(u_r k) + \rho \frac{1}{r} \frac{\partial}{\partial \theta}(u_\theta k) + \rho \frac{\partial}{\partial z}(u_z k) = \\ \frac{1}{r} \frac{\partial}{\partial r} \left\{ \left( \mu + \frac{\mu_t}{\sigma_k} \right) r \frac{\partial k}{\partial r} \right\} + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left\{ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial \theta} \right\} + \frac{\partial}{\partial z} \left\{ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial z} \right\} + \rho(P_k + \Gamma_b - \varepsilon) \end{aligned} \quad (19)$$

Transport equation for  $\varepsilon$ —cylindrical coordinates  $(r, \theta, z)$ :

$$\begin{aligned} \rho \frac{\partial \varepsilon}{\partial t} + \rho \frac{\partial}{\partial r}(u_r \varepsilon) + \rho \frac{1}{r} \frac{\partial}{\partial \theta}(u_\theta \varepsilon) + \rho \frac{\partial}{\partial z}(u_z \varepsilon) = \\ \frac{1}{r} \frac{\partial}{\partial r} \left\{ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) r \frac{\partial \varepsilon}{\partial r} \right\} + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left\{ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial \theta} \right\} + \frac{\partial}{\partial z} \left\{ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial z} \right\} \\ + \rho \frac{\varepsilon}{k} (C_{1\varepsilon} P_k + C_{3\varepsilon} \Gamma_b - C_{2\varepsilon} \varepsilon) \end{aligned} \quad (20)$$

$$\text{where } \tau_{rr} = 2\rho\nu \frac{\partial u_r}{\partial r} \quad (21)$$

$$\tau_{\theta\theta} = 2\rho\nu \left( \frac{1}{r} \frac{\partial u_\theta}{\partial \theta} + \frac{u_r}{r} \right) \quad (22)$$

$$\tau_{zz} = 2\rho\nu \frac{\partial u_z}{\partial z} \quad (23)$$

$$\tau_{r\theta} = \tau_{\theta r} = \rho\nu \left( r \frac{\partial}{\partial r} \left( \frac{u_\theta}{r} \right) + \frac{1}{r} \frac{\partial u_r}{\partial \theta} \right) \quad (24)$$

$$\tau_{\theta z} = \tau_{z\theta} = \rho\nu \left( \frac{\partial u_\theta}{\partial z} + \frac{1}{r} \frac{\partial u_z}{\partial \theta} \right) \quad (25)$$

$$\tau_{zr} = \tau_{rz} = \rho \nu \left( \frac{\partial u_z}{\partial r} + \frac{\partial u_r}{\partial z} \right) \quad (26)$$

$$g_r = 0, \quad g_\theta = 0, \quad g_z = -g \quad (27)$$

$$h = c_p T \quad (28)$$

$$\nu_t = C_\mu \frac{k^2}{\varepsilon} \quad (29)$$

$$P_k = \nu_t \left[ 2 \left\{ \left( \frac{\partial u_r}{\partial r} \right)^2 + \left( \frac{1}{r} \frac{\partial u_\theta}{\partial \theta} + \frac{u_r}{r} \right)^2 + \left( \frac{\partial u_z}{\partial z} \right)^2 \right\} + \left( \frac{1}{r} \frac{\partial u_r}{\partial \theta} + \frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r} \right)^2 + \left( \frac{\partial u_\theta}{\partial z} + \frac{1}{r} \frac{\partial u_z}{\partial \theta} \right)^2 + \left( \frac{\partial u_z}{\partial r} + \frac{\partial u_r}{\partial z} \right)^2 \right] \quad (30)$$

$$\Gamma_b = \frac{-\nu_t}{\rho \sigma_t} \left( g_r \frac{\partial \rho}{\partial r} + g_\theta \frac{1}{r} \frac{\partial \rho}{\partial \theta} + g_z \frac{\partial \rho}{\partial z} \right) \quad (31)$$

$$\lambda_e = \lambda_t + \lambda \quad (32)$$

$$\lambda_t = \frac{c_p \mu_t}{\sigma_t} \quad (33)$$

$$\nu = \frac{\mu}{\rho} + \nu_t \quad (34)$$

Here,  $u_r$ ,  $u_\theta$  and  $u_z$  are the  $r$ ,  $\theta$ , and  $z$  components of the velocity vector, respectively. The values of the constants,  $\sigma_k$ ,  $\sigma_\varepsilon$ ,  $\sigma_t$ ,  $C_{1e}$ ,  $C_{2e}$ ,  $C_{3e}$ , and  $C_\mu$ , which are used in Eqs. (18)–(20), (29), (31), and (33), are listed in Table 2.

## Boundary Conditions

The fundamental equations are numerically analyzed under the following boundary conditions. Constant heat flux and a nonslip condition are assumed on the outer boundary of the heated section:

$$q = -\lambda \frac{\partial T}{\partial r} = \text{constant} \quad (35)$$

At the outer boundary of the nonheated section:

$$\frac{\partial T}{\partial r} = 0 \quad (36)$$

At the lower boundary:

$$T = T_{in}, \quad u_r = 0, \quad u_\theta = 0 \quad \text{and} \quad u_z = u \quad \text{for in-flow} \quad (37)$$

where  $T_{in}$  and  $u$  are inlet liquid temperature and flow velocity at the entrance of the test section, respectively.

## Solution Method

Control volume discretization equations were derived from the fundamental equations by using the hybrid scheme [10]. The thermophysical properties for the control volume obtained at each control volume temperature are numerically analyzed. The flow field is calculated the Semi-Implicit Method for Pressure-Linked Equations algorithm [11]. Uniform heat flux,  $q$ , was assumed at the heated pipe wall in the range  $3.41 \times 10^5$  to  $2.68 \times 10^7$  W/m<sup>2</sup> as the boundary condition, and the calculations were performed until the steady-state was obtained. The test tube surface temperature,  $T_s$ , was obtained from the temperature of the outer control volume on the test tube surface,  $TEM$ , located at the center of the control volume, by solving the heat conduction equation in liquids [1-5]:

$$T_s = \frac{q (\Delta r)_{out}}{\lambda_l} + TEM \quad (38)$$

where  $(\Delta r)_{out}$  is the outer control volume width along the  $r$ -component. The average heat transfer coefficient on the test tube surface was obtained by averaging the calculated local surface temperatures,  $(T_s)_z$ , at every 0.5 mm along the heated length,  $L$ . Flow and temperature field predictions were obtained using the PHOENICS CFD code [8].

## RESULTS AND DISCUSSION

### Experimental Conditions and Calculation Parameters

Steady-state and transient heat transfer processes on Pt test tubes of inner diameter 3 and 6 mm subject to exponentially increasing heat inputs,  $Q_{0exp}(t/\tau)$ , were measured. The exponential periods,  $\tau$ , of the heat input ranged from 15.43 ms to 26.98 s. The initial experimental conditions, such as inlet flow velocity, inlet liquid temperature, inlet pressure, and exponential period, for the single-phase flow heat transfer experiment were determined independent of each other before each experimental run.

The experimental conditions were as follows:

|                                                     |                                                                                                      |
|-----------------------------------------------------|------------------------------------------------------------------------------------------------------|
| Heater material                                     | Platinum (Pt)                                                                                        |
| Surface condition                                   | Commercial finish of inner surface                                                                   |
| Surface roughness for $R_a$ , $R_{max}$ , and $R_z$ | 0.40, 2.20, and 1.50 $\mu\text{m}$ for $d = 3$ mm; 0.45, 2.93, and 1.93 $\mu\text{m}$ for $d = 6$ mm |
| Inner diameter ( $d$ )                              | 3 and 6 mm                                                                                           |
| Heated length ( $L$ )                               | 66.5 and 100 mm for $d = 3$ mm; 69.6 mm for $d = 6$ mm                                               |
| Effective length ( $L_{eff}$ )                      | 56.7 and 100 mm for $d = 3$ mm; 59.2 mm for $d = 6$ mm                                               |
| $L_{12}$ , $L_{23}$ , and $L_{34}$                  | 19.2, 18.8, and 18.7 mm for $d = 3$ mm; 19.6, 20.0, and 19.6 mm for $d = 6$ mm                       |
| $L/d$                                               | 22.2 and 33.3 for $d = 3$ mm; 11.6 for $d = 6$ mm                                                    |

|                                             |                                                   |
|---------------------------------------------|---------------------------------------------------|
| $L_{eff}/d$                                 | 18.9 and 33.3 for $d = 3$ mm; 9.87 for $d = 6$ mm |
| Wall thickness ( $\delta$ )                 | 0.5 and 0.4 mm                                    |
| Inlet flow velocity ( $u$ )                 | 4.01–21.79 m/s                                    |
| Inlet pressure ( $P_{in}$ )                 | 803.32–1143.96 kPa                                |
| Outlet pressure ( $P_{out}$ )               | 786.99–1123.41 kPa                                |
| Inlet subcooling ( $\Delta T_{sub,in}$ )    | 92.01–146.96 K                                    |
| Outlet subcooling ( $\Delta T_{sub,out}$ )  | 80.76–145.41 K                                    |
| Inlet liquid temperature ( $T_{in}$ )       | 298.56–362.32 K                                   |
| Prandtl number ( $Pr$ )                     | 1.85–5.89                                         |
| Exponentially increasing heat input ( $Q$ ) | $Q_0 \exp(t/\tau)$ , $\tau = 15.43$ ms to 26.98 s |

The parameters used in the calculation were as follows:

|                                       |                                                                                                                |
|---------------------------------------|----------------------------------------------------------------------------------------------------------------|
| Inner diameter ( $d$ )                | 3 and 6 mm                                                                                                     |
| Heated length ( $L$ )                 | 67 and 100 mm for $d = 3$ mm; 70 mm for $d = 6$ mm                                                             |
| Entrance length ( $L_e$ )             | 282 mm for $d = 3$ mm; 333 mm for $d = 6$ mm                                                                   |
| Exit length ( $L_{ex}$ )              | 177 mm for $d = 3$ mm; 233 mm for $d = 6$ mm                                                                   |
| Test section length ( $L_{ts}$ )      | 526 and 559 mm for $d = 3$ mm; 636 mm for $d = 6$ mm                                                           |
| Heat flux ( $q$ )                     | $3.41 \times 10^5$ to $2.68 \times 10^7$ W/m <sup>2</sup> ( $q_0 \exp(t/\tau)$ , $\tau = 15.43$ ms to 26.98 s) |
| Inlet flow velocity ( $u$ )           | 4.01–21.79 m/s                                                                                                 |
| Inlet liquid temperature ( $T_{in}$ ) | 298.56–362.32 K                                                                                                |
| Coordinate system                     | cylindrical ( $r, \theta, z$ )                                                                                 |
| Control volume number                 | 17–34, 60, 894–978                                                                                             |
| Physical model                        | $k$ – $\varepsilon$ turbulence model with high Reynolds number form                                            |
| Wall functions                        | logarithmic law ( $8.404 \leq y^+ \leq 30.86$ )                                                                |

### **Inner and Outer Surface Temperatures, $T_s$ and $T_{so}$ , Heat Fluxes, $q$ , and Heat Transfer Coefficients, $h$ , at the Three Sections, and Inlet and Outlet Liquid Temperatures, $T_{in}$ and $T_{out}$ , for the 3-mm Inner Diameter Test Tube**

Figure 6 shows the inner surface temperatures ( $T_{s1}$ ,  $T_{s2}$ , and  $T_{s3}$ ; solid black circles), outer surface temperatures ( $T_{so1}$ ,  $T_{so2}$ , and  $T_{so3}$ ; open black circles), heat fluxes ( $q_1$ ,  $q_2$ , and  $q_3$ ; solid black triangles), and heat transfer coefficients ( $h_1$ ,  $h_2$ , and  $h_3$ ; open black triangles) at the first, second, and third sections (i.e., lengths between the first and second potential taps,

second and third taps, and third and fourth taps, respectively) and the inlet and outlet liquid temperatures ( $T_{in}$  and  $T_{out}$ ; solid black squares) at a flow velocity,  $u$ , of 13.91 m/s under a heat flux,  $q$ , of 7.13 MW/m<sup>2</sup>. Liquid temperatures at the center of each section (heated lengths,  $L$ , of 14.4, 33.4, and 52.15 mm; open black squares) were linearly estimated from  $T_{in}$  and  $T_{out}$ . The inner ( $T_{s1}$ ,  $T_{s2}$ , and  $T_{s3}$ ) and outer ( $T_{so1}$ ,  $T_{so2}$ , and  $T_{so3}$ ) surface temperatures were obtained using Eqs. (8) and (9) and the measured average temperature and surface heat flux; in all three sections, these temperatures increased gradually with increase in heated length, whereas the heat fluxes ( $q_1$ ,  $q_2$ , and  $q_3$ ) were nearly constant. The rate of increase of the inner surface temperature exhibited nearly the same trend as that of liquid temperature from the inlet to the outlet ( $T_{in}$  to  $T_{out}$ ). The heat transfer coefficients for each section ( $h_1$ ,  $h_2$ , and  $h_3$ ) gradually stabilized to a near-constant value.

In the same figure, the red dotted–dashed line represents the numerical solution of z-axis variations in the analyzed liquid temperature of the outer control volume on the test tube surface,  $TEM$ , at a heat flux of 7.13 MW/m<sup>2</sup> and a flow velocity of 13.91 m/s. The surface temperatures on the test tube,  $T_s$ , were calculated using  $TEM$ , which is located at the center of the control volume, by solving the heat conduction equation in liquids [Eq. (38)]. The red solid line represents the z-axis variations in the inner surface temperature of the test tube at every 0.5 mm along the heated length,  $L$ . Both  $TEM$  and the z-axis variations gradually increased with increasing z-axis distance from the leading edge of the test tube. The numerical solutions of inner surface temperatures solved by the RANS equations for unsteady turbulent heat transfer agree well with the experimental data ( $T_{s1}$ ,  $T_{s2}$ , and  $T_{s3}$ ) obtained using the basic two-dimensional unsteady heat conduction equation and Eq. (8), which in turn is obtained from the steady one-dimensional heat conduction equation.

The outer surface temperature of the test tube was monitored through infrared thermal imaging. In Fig. 6, the blue solid line represents the z-axis variations in the outer surface temperature of the test tube,  $T_{so}$ , for an exponential period of 7.96 s at a flow velocity of 13.91 m/s and a heat flux of 7.13 MW/m<sup>2</sup>. The outer surface temperature of the test tube steeply increased from 299.6 K at the electrode of the inlet to 405.2 K at 5.6 mm from the leading edge of the test tube; it gradually initially increased, then decreased, and then increased again, peaking at approximately 435.6 K 12.2 mm away from the test tube outlet and finally linearly decreasing to 301.6 K at the outlet electrode. Both ends of the Pt test tube were soldered to Cu electrode plates of width 80 mm, length 120 mm, and thickness 5 mm. The heat capacities of the Cu electrode plates subcooled with water flow is much higher than the joule heat of the test tube of wall thickness,  $\delta$ , 0.5 mm. The liquid temperature of the outlet ( $T_{out} = 313.88$  K) was 12.13 K higher than that of the inlet ( $T_{in} = 301.75$  K) under uniform heating of the test tube with  $q = 7.13$  MW/m<sup>2</sup>. The outer surface temperatures in the first, second, and third sections (solid blue line) gradually increased with increase in heated length, and the rate of increase of the outer surface temperature became the same as that of the outer surface temperature calculated using Eq. (9). Accordingly, the heat transfer coefficient of the inner surface of the test tube is nearly constant along the heated length in the nonboiling region. The outer surface temperatures

(dashed blue line) measured with  $\varepsilon = 0.94$  in the first, second, and third sections were 6.4% higher than those calculated using Eq. (9) [12], despite the  $\pm 2\%$  accuracy of the infrared thermal imaging camera. The temperature curve in the figure was asymptotically revised on the basis of the values calculated using Eq. (9).

For comparison, the red dashed line in Fig. 6 represents the numerical solution of the average liquid temperature,  $T_{f,av}$ , from the inlet to the outlet, calculated using the analyzed liquid temperatures of the control volumes on the  $r$ - and  $\theta$ -axis grid numbers of the  $r$ - $\theta$  plane for each  $z$ -axis grid number. This curve became increasingly linearly with increase in the  $z$ -axis distance,  $Z$ , and was almost the same as the outlet liquid temperature,  $T_{out}$ , at the exit of the heated length. Thus, the heat transfer coefficients,  $h$ , calculated from the heat fluxes,  $q$ , and the local liquid temperatures,  $T_f$ , at heated lengths,  $L$ , of 14.4, 33.4, and 52.15 mm, which were obtained through linear interpolation of the experimental inlet and outlet liquid temperatures,  $T_{in}$  and  $T_{out}$ , will also be true, although these inlet and outlet liquid temperatures were measured using one 1-mm outer diameter, sheathed, K-type thermocouple in the  $d = 3$  mm test tube. Namely, the outlet liquid temperature, measured using a 1-mm outer diameter, sheathed, K-type thermocouple, indicates the local liquid bulk mean temperature at the lower stream point, 53 mm (63 mm) from the tube outlet, for the 3-mm (6-mm) inner diameter test section. The local liquid bulk mean temperature increases linearly from the inlet liquid temperature at the tube inlet and attains the outlet liquid temperature at the tube outlet with an increase in heated length.

## Steady-state and Transient Turbulent Heat Transfer Characteristics

### Influence of the heated length-to-inner diameter ratio

Figures 7, 8, and 9 show typical examples of the steady-state turbulent heat transfer curves for the circular Pt tube of inner diameter ( $d$ ) 3 mm, heated lengths ( $L$ ) 32.7, 66.5, and 100 mm, and heated length-to-inner diameter ratios ( $L/d$ ) 10.9, 22.2, and 33.3 for an exponential period ( $\tau$ ) of approximately 25 s at flow velocities ( $u$ ) of 13.32, 13.78, and 14.12 m/s, respectively. The experimental data were compared with the values derived from the authors' correlation of the steady-state turbulent heat transfer for an empty tube, Eq. (39), at flow velocities,  $u$ , of 13.32, 13.78, and 14.12 m/s, respectively.

$$Nu_d = 0.02 Re_d^{0.85} Pr^{0.4} \left( \frac{L_{eff}}{d} \right)^{-0.08} \left( \frac{\mu_l}{\mu_w} \right)^{0.14} \quad (39)$$

The heat fluxes gradually increased with increase in the difference between the average inner surface temperature and the liquid bulk mean temperature,  $\Delta T_L (= T_{s,av} - T_L)$ , on the steady-state turbulent heat transfer curve derived using Eq. (39) [13]. In Figs. 7–9, the numerical solutions for the relationship between heat flux,  $q$ , and the difference between the average inner surface temperature and liquid bulk mean temperature,  $\Delta T_L$ , are represented as green solid circles for  $q$  ranging from  $6.47 \times 10^5$  to  $1.67 \times 10^7$  W/m<sup>2</sup>, from  $3.91 \times 10^5$  to  $1.10 \times 10^7$  W/m<sup>2</sup>, and from  $6.42 \times 10^5$  to  $1.49 \times 10^7$  W/m<sup>2</sup> at flow

velocities of 13.32, 13.78, and 14.12 m/s, respectively. The 14 numerical solutions are plotted for  $\Delta T_L$  ranging from 8.74 to 216.89 K, from 5.45 to 149.14 K, and from 8.98 to 205.17 K on log–log scales. These solutions also increased with an increase in  $\Delta T_L$  along the curve obtained using Eq. (39). The numerical solutions solved using the RANS equations for unsteady turbulent heat transfer are in good agreement with the experimental data, and the values derived using Eq. (39) are within  $\pm 10\%$ . The outer control volume widths for the  $r$ -component,  $(\Delta r)_{outs}$  are 13.04  $\mu\text{m}$ .

These experimental results were compared with the results derived from other researchers' correlations of steady-state turbulent heat transfer in pipes. The experimental results for  $d = 3$  mm and  $L = 32.7$ , 66.5, and 100 mm at high heat flux in Figs. 7, 8, and 9 are 17.6%–53.8%, 15.4%–56.0%, and 8.0%–46.8% higher than the values derived using the correlations at a fixed temperature difference between the average inner surface temperature and liquid bulk mean temperature ( $\Delta T_L = \text{constant}$ ), and 20–64 K, 17.1–72.0 K, and 8.9–56.5 K lower than the values derived using the correlations at a fixed heat flux ( $q = \text{constant}$ ), respectively [13]. In this study, the sixth term,  $(\mu_l/\mu_w)^{0.14}$ , in the authors' correlation of the steady-state turbulent heat transfer in pipe, Eq. (39), was found to strongly influence the correlation with the experimental data for a wide range of  $\Delta T_L$ . The steady-state turbulent heat transfer curves for the experimental data gradually increased with increase in  $\Delta T_L$  on the log–log graphs and was not linear.

#### Influence of inlet liquid temperature

Figures 10, 11, and 12 show typical examples of steady-state turbulent heat transfer curves for circular Pt tubes of  $d = 6$  mm,  $L = 69.6$  mm,  $L/d = 11.6$ ,  $L_{eff} = 59.2$  mm, and  $L_{eff}/d = 9.87$  with inlet liquid temperatures,  $T_{in}$ , of 302.68, 322.38, and 362.02 K at flow velocities,  $u$ , of 13.51, 13.64, and 13.92 m/s, respectively. The experimental data were compared with the values derived from the authors' correlation of steady-state turbulent heat transfer for an empty tube [Eq. (39)] at flow velocities,  $u$ , of 13.51, 13.64, and 13.92 m/s, respectively. The heat fluxes gradually increased with increase in the difference between the average inner surface temperature and liquid bulk mean temperature,  $\Delta T_L (= T_{s,av} - T_L)$ , on the steady-state turbulent heat transfer curve derived from Eq. (39) [13].

In the figures, the numerical solutions for the relationship between heat flux,  $q$ , and the difference between the average inner surface temperature and liquid bulk mean temperature,  $\Delta T_L$ , are shown as green solid circles for  $q$  ranging from  $3.41 \times 10^5$  to  $8.49 \times 10^6$  W/m<sup>2</sup>, from  $6.70 \times 10^5$  to  $2.68 \times 10^7$  W/m<sup>2</sup>, and from  $6.66 \times 10^5$  to  $2.49 \times 10^7$  W/m<sup>2</sup> at flow velocities of 13.51, 13.64, and 13.92 m/s, respectively. The 12, 14, and 14 numerical solutions are plotted for  $\Delta T_L$  ranging from 4.75 to 119.11 K, from 8.28 to 350.63 K, and from 7.27 to 336.48 K on log–log scales, respectively. These solutions also increased with increase in  $\Delta T_L$  along the curve derived from Eq. (39). The numerical solutions obtained using the RANS equations for

unsteady turbulent heat transfer are in good agreement with the experimental data, and the values derived from Eq. (39) were within  $\pm 10\%$ . The outer control volume widths for the  $r$ -component,  $(\Delta r)_{out}$ , are 13.04, 10.48, and 8.5  $\mu\text{m}$ , respectively.

### Influence of exponential period

Figures 13, 14, and 15 show typical examples of transient turbulent heat transfer curves for circular Pt tubes of  $d = 6$  mm,  $L = 69.6$  mm,  $L/d = 11.6$ ,  $L_{eff} = 59.2$  mm, and  $L_{eff}/d = 9.87$  for exponential periods,  $\tau$ , of 6.51 s, 124.8 ms, and 28.2 ms at flow velocities,  $u$ , of 4.13, 4.01, and 4.19 m/s, respectively. Experimental data were compared with data derived from the authors' correlations of the steady-state and transient turbulent heat transfer, Eqs. (39) and (40), at flow velocities,  $u$ , of 4.13, 4.01, and 4.19 m/s, respectively [13, 14].

$$Nu_d = 0.02 Re_d^{0.85} Pr^{0.4} \left( \frac{L_{eff}}{d} \right)^{-0.08} \left( \frac{\mu_l}{\mu_w} \right)^{0.14} (1 + 9.46 P^{*-0.8}) \quad (40)$$

As shown in Figs. 13 and 14, the heat fluxes for  $\tau = 6.51$  s and 124.8 ms increased with increase in the  $\Delta T_L$  on the curves derived from Eq. (39) up to the point where the slope began to increase with heat flux following the onset of nucleate boiling similar, to the case in steady-state heat transfer curve. However, for  $\tau$  less than approximately 124.8 ms, the heat fluxes gradually increased with increase in  $\Delta T_L$  on the higher curve parallel to the steady-state turbulent heat transfer curve derived from Eq. (39) up to the boiling initiation point. For  $\tau = 28.2$  ms, the experimental data at the boiling initiation point in Fig. 15 were 44.1% higher than that derived from Eq. (39) with  $u = 4.19$  m/s at a fixed temperature difference between the heater inner surface temperature and liquid bulk mean temperature ( $\Delta T_L = \text{constant}$ ), and 62.7 K lower than that derived from Eq. (39) at a fixed heat flux ( $q = \text{constant}$ ) [13, 14]. For comparison, the curves for the transient turbulent heat transfer correlation, Eq. (40), are presented in Figs. 13–15. The transient turbulent heat transfer data for a wide range of exponential periods are in good agreement with the values obtained using Eq. (40).

In Figs. 13–15, the numerical solutions for the relationship between heat flux,  $q$ , and the difference between the average inner surface temperature and liquid bulk mean temperature,  $\Delta T_L$ , are shown as green solid circles for  $q$  ranging from  $3.86 \times 10^5$  to  $1.75 \times 10^7$  W/m<sup>2</sup>, from  $4.19 \times 10^5$  to  $1.31 \times 10^7$  W/m<sup>2</sup>, and from  $3.31 \times 10^5$  to  $5.86 \times 10^6$  W/m<sup>2</sup> for exponential periods,  $\tau$ , of 6.51 s, 124.8 ms, and 28.2 ms, respectively. The 13, 11, and 11 numerical solutions are plotted for  $\Delta T_L$  ranging from 15.41 to 732.34 K, from 17.25 to 521.89 K, and from 9.00 to 152.19 K on log–log scales, respectively. These solutions also increased with an increase in  $\Delta T_L$  along the curve derived from Eq. (40). The numerical solutions solved using the RANS equations for unsteady turbulent heat transfer are in good agreement with the experimental data, and the values derived from Eq. (40) are within  $\pm 10\%$  difference. The outer control volume widths for the  $r$ -component,  $(\Delta r)_{out}$ , are 38.34, 38.34, and 21.18  $\mu\text{m}$  for exponential periods,  $\tau$ , of approximately 6.51 s, 124.8 ms, and 28.2 ms, respectively.

### Thickness of Conductive Sublayer, $\delta_{CSL,st}$ and $\delta_{CSL}$ , and Nondimensional Thickness of Conductive Sublayer, $y^+_{CSL,st}$ and $y^+_{CSL}$ for Steady-state and Transient Turbulent Heat Transfer

The numerical solutions solved using the RANS equations for unsteady turbulent heat transfer are in good agreement with the experimental data for a wide range of heated length-to-inner diameter ratios, inlet liquid temperatures, and exponential periods, and the values derived from Eqs. (39) and (40) are within  $\pm 10\%$  (Figs 7–15). In Fig. 16, the test tube surface of the  $d = 3$  mm test tube is located at  $r = -1.5$  mm, and a conductive sublayer [15, 16] exists on the test tube surface. The liquid temperatures in the conductive sublayer on the test tube surface decrease linearly with decrease in the radius because of heat conduction from the test tube surface:  $T_f = T_s - \Delta r q / \lambda_l$ . Let  $T_f$  equal the analyzed liquid temperature of the outer control volume on the test tube surface,  $TEM$ , in the turbulent flow region, which is located at the center of the control volume, as expressed in Eq. (38). Half the outer control volume width for the  $r$ -component,  $(\Delta r)_{out}/2$ , would become the thickness of the conductive sublayer,  $\delta_{CSL}$ , for turbulent heat transfer in a vertical circular tube under the two-phase model, which comprises a conductive sublayer and an inner region of turbulent flow.

Relationships between  $\delta_{CSL,st}$  and the nondimensional thickness of conductive sublayer,  $y^+_{CSL,st}$ , numerically solved for steady-state turbulent heat transfer, and  $u$  for  $d = 3$  mm,  $L = 33, 67$ , and  $100$  mm, and  $L/d = 11, 22.33$ , and  $33.33$  with inlet liquid temperature,  $T_{in}$ , of approximately  $308$  K, and for  $d = 6$  mm,  $L = 70$  mm, and  $L/d = 11.67$  with  $T_{in}$  of  $296.47, 322.38, 353.37$ , and  $362.02$  K are  $\Delta T_L (= T_{s,av} - T_L)$  of  $100$  K in Figs. 17 and 18, respectively. To calculate the nondimensional thicknesses of the conductive sublayer,  $(y^+_{CSL,st})_{TEM}$ , all properties in the equation are evaluated at  $TEM$ . The nondimensional thicknesses of the conductive sublayer for steady-state and transient turbulent heat transfer,  $(y^+_{CSL,st})_{TEM}$  and  $(y^+_{CSL})_{TEM}$ , are defined as follows:

$$(y^+_{CSL,st})_{TEM} = \left( \frac{f_F}{2} \right)^{0.5} \frac{\rho_l u \delta_{CSL,st}}{\mu_l} \text{ and } (y^+_{CSL})_{TEM} = \left( \frac{f_F}{2} \right)^{0.5} \frac{\rho_l u \delta_{CSL}}{\mu_l} \quad (41)$$

$$f_F = 0.126 \text{Re}_d^{-0.25} \quad (42)$$

where  $f_F$  is Fanning friction factor [17, 18].  $\delta_{CSL,st}$  for  $d = 3$  mm with  $L/d = 11, 22.33$ , and  $33.33$  and for  $d = 6$  mm with  $T_{in} = 296.47, 322.38, 353.37$ , and  $362.02$  K decreased linearly with increase in flow velocity,  $u$ , on a log–log scale; however, those of  $(y^+_{CSL,st})_{TEM}$  increased a little with increase in  $u$  but was nevertheless nearly constant over the entire numerical range.

Relationships between  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$ , and  $d$  for the turbulent heat transfer numerically solved for circular tube of  $d = 3, 6$ , and  $9$  mm;  $L = 33, 70$ , and  $50$  mm; and  $L/d = 11, 11.67$ , and  $5.56$  with  $T_{in}$  of  $308.20$ – $311.77$  K are shown  $\Delta T_L (=$

$T_{s,gv} - T_L$ ) of 100 K in Fig. 19. Both  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  slightly increased linearly with increase in  $d$  on a log–log scale. Relationships between  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  numerically solved and  $L/d$  for  $L/d = 11, 22.33$ , and  $33.33$  with  $d = 3$  mm and  $L = 33, 67$ , and  $100$  mm at  $T_{in}$  of approximately 308 K are presented in Fig. 20. Both  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  also slightly increased linearly with increase in  $L/d$  on a log–log scale. Relationships between  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  numerically solved and the Prandtl number,  $Pr$ , for  $T_{in}$  of 298.56, 322.38, 353.37, and 362.02 K with  $d = 6$  mm,  $L = 70$  mm, and  $L/d = 11.67$  are shown in Fig. 21.  $\delta_{CSL,st}$  increased with increase in  $Pr$  on a log–log scale; by contrast,  $(y^+_{CSL,st})_{TEM}$  decreased with increase in  $Pr$ . The slopes of the aforementioned log–log curves are almost constant at approximately 0.45 for  $\delta_{CSL,st}$  and  $-0.25$  for  $(y^+_{CSL,st})_{TEM}$ , with  $Pr$  ranging from 1.85 to 5.89.

The thickness of the conductive sublayer,  $\delta_{CSL}$ , and the ratios of the thickness of conductive sublayer for transient turbulent heat transfer to the thickness of conductive sublayer for steady-state turbulent heat transfer,  $\delta_{CSL}/\delta_{CSL,st}$  for  $d = 6$  mm,  $L = 70$  mm, and  $L/d = 11.67$ , and  $d = 3$  mm,  $L = 67$  mm, and  $L/d = 22.33$  at  $T_{in}$  of approximately 300 K are plotted versus the exponential period,  $\tau$  (15.4 ms to 6.33 s) at flow velocities,  $u$ , of 4, 6.9, 9.9, and 13.3 m/s in Fig. 22.  $\delta_{CSL,st}$  for the steady-state turbulent heat transfer are given the values of thickness of conductive sublayer at  $\tau_{st} =$  around 6 s, which is the maximum. The figure also illustrates the trends in the variation of the thickness of the conductive sublayer with decreasing exponential period.  $\delta_{CSL}$  decreased with decrease in the exponential period at low  $\tau$  for a fixed  $u$ , and was almost constant for  $\tau$  exceeding 50 ms; the thickness decreased with decrease in  $\tau$ . Over the entire experimental range,  $\delta_{CSL}$  decreased with increase in flow velocity at a fixed  $\tau$ . The ratios,  $\delta_{CSL}/\delta_{CSL,st}$ , can be expressed using the following equation:

$$\frac{\delta_{CSL}}{\delta_{CSL,st}} = \left[ 1^{-8} + (13.12\tau^{0.8})^{-8} \right]^{-0.125} \quad (43)$$

The nondimensional thickness of conductive sublayer,  $(y^+_{CSL})_{TEM}$ , and the ratios of the nondimensional thickness of conductive sublayer for transient turbulent heat transfer to the nondimensional thickness of conductive sublayer for steady-state turbulent heat transfer,  $(y^+_{CSL})_{TEM}/(y^+_{CSL,st})_{TEM}$ , for  $d = 6$  mm,  $L = 70$  mm, and  $L/d = 11.67$ , and  $d = 3$  mm,  $L = 67$  mm, and  $L/d = 22.33$  at  $T_{in}$  of approximately 300 K are plotted versus  $\tau$  (15.4 ms to 6.33 s) at flow velocities,  $u$ , of 4, 6.9, 9.9, and 13.3 m/s for the  $\Delta T_L$  of 40, 80, 100, and 120 K in Figs. 23 and 24. These figures also illustrate the trends in the variation of the nondimensional thickness of conductive sublayer with decreasing exponential period. The nondimensional thicknesses of the conductive sublayer decreased with decrease in the exponential period at a low  $\tau$  for a fixed  $u$  and  $\Delta T_L$ . The nondimensional thickness of conductive sublayer,  $(y^+_{CSL})_{TEM}$ , was nearly constant for  $\tau$  exceeding 50 ms and decreased with decrease in  $\tau$ . Over the entire experimental range,  $(y^+_{CSL})_{TEM}$  increased with increase in  $u$  and  $\Delta T_L$  at a fixed  $\tau$ .  $(y^+_{CSL})_{TEM}/(y^+_{CSL,st})_{TEM}$  can be expressed as follows:

$$\frac{(y_{CSL})_{TEM}}{(y_{CSL,st})_{TEM}} = \left[ 1^{-8} + (3.85\tau^{0.45})^{-8} \right]^{-0.125} \quad (44)$$

These numerical solutions of  $\delta_{CSL,st}$  and  $(y_{CSL,st})_{TEM}^+$  for steady-state turbulent heat transfer can be expressed for  $L/d$  of 11–33.33,  $T_{in}$  of 302.68–362.02 K, and  $Pr$  of 1.85–5.89 for  $\Delta T_L$  of 40–120 K by using the following correlations:

Steady-state turbulent heat transfer:

$$\delta_{CSL,st} = 50.864 \times 10^{-6} d^{0.15} u^{-0.85} \left( \frac{L}{d} \right)^{0.08} Pr^{0.45} \quad (45)$$

$$(y_{CSL,st})_{TEM}^+ = 12.538 e^{\frac{\Delta T_L}{205}} d^{0.1} u^{0.1} \left( \frac{L}{d} \right)^{0.08} Pr^{-0.25} \quad (46)$$

In Eq. (45),  $1/\delta_{CSL} \propto d^{-0.15} u^{0.85} (L/d)^{-0.08} Pr^{-0.45}$ , represents the authors' steady-state turbulent heat transfer correlation, Eq. (39), derived from the experimental data on the test tube,  $h \propto d^{-0.15} u^{0.85} (L/d)^{-0.08} Pr^{-0.45}$ . The numerical solutions of  $\delta_{CSL}$  and  $(y_{CSL})_{TEM}^+$  for transient turbulent heat transfer can also be expressed for  $\tau$  ranging from 15.4 ms to 6.33 s for  $\Delta T_L$  ranging from 40 to 120 K by using the following correlations:

Transient turbulent heat transfer:

$$\delta_{CSL} = 50.864 \times 10^{-6} d^{0.15} u^{-0.85} \left( \frac{L}{d} \right)^{0.08} Pr^{0.45} \left[ 1^{-8} + (13.12\tau^{0.8})^{-8} \right]^{-0.125} \quad (47)$$

$$(y_{CSL})_{TEM}^+ = 12.538 e^{\frac{\Delta T_L}{205}} d^{0.1} u^{0.1} \left( \frac{L}{d} \right)^{0.08} Pr^{-0.25} \left[ 1^{-8} + (3.85\tau^{0.45})^{-8} \right]^{-0.125} \quad (48)$$

In Eq. (47),  $1/\delta_{CSL} \propto d^{-0.15} u^{0.85} (L/d)^{-0.08} Pr^{-0.45} \tau^{-0.8}$ , represents the authors' transient turbulent heat transfer correlation, Eq. (40), derived from the experimental data on the test tube,  $h \propto d^{-0.15} u^{0.85} (L/d)^{-0.08} Pr^{-0.45} \tau^{-0.8}$ . For comparison, the curves derived from Eqs. (43) to (48) are shown in Figs. 17 to 24.  $\delta_{CSL,st}$  and  $(y_{CSL,st})_{TEM}^+$  for steady-state turbulent heat transfer and the  $\delta_{CSL}$  and  $(y_{CSL})_{TEM}^+$  for transient turbulent heat transfer are almost in good agreement with the values given by these correlations. These equations are composed of the parameter  $d$ ,  $u$ ,  $L/d$ ,  $Pr$ ,  $\Delta T_L$ , and  $\tau$ .

## CONCLUSIONS

Steady-state and transient turbulent heat transfer coefficients in a circular Pt test tube for  $u = 4.01$ – $21.79$  m/s,  $T_{in} = 298.56$ – $362.32$  K,  $Pr = 1.85$ – $5.89$ ,  $P_{in} = 803.32$ – $1143.96$  kPa and  $Q_{0exp}(t/\tau)$  with  $\tau = 15.43$  ms to  $26.98$  s were systematically measured. Measurements were made on  $d = 3$  and  $6$  mm with  $L = 66.5$  to  $100$  mm and  $L/d = 11.6$  to  $33.3$ . The RANS equations for unsteady turbulent heat transfer in circular tubes of  $d = 3$  and  $6$  mm and length  $526$ – $636$  mm were numerically solved for heating of water with  $L = 67$  to  $100$  mm by using the PHOENICS CFD code under the same conditions as the experiment and with temperature-dependent thermophysical fluid properties.

- 1) The inner and outer surface temperatures,  $T_{s1}$ ,  $T_{s2}$ , and  $T_{s3}$ , and  $T_{so1}$ ,  $T_{so2}$ , and  $T_{so3}$ , respectively, increased with increase in the heated length from the leading edge of the test tube, whereas heat fluxes,  $q_1$ ,  $q_2$ , and  $q_3$ , were almost constant in each section. The rate of increase of the inner surface temperature followed nearly the same trend as that of liquid temperature from the inlet to the outlet,  $T_{in}$  to  $T_{out}$ . The heat transfer coefficients for each section,  $h_1$ ,  $h_2$ , and  $h_3$ , was nearly constant.
- 2) The numerical solutions of the inner surface temperatures solved using the RANS equations for unsteady turbulent heat transfer are in good agreement with the experimental data ( $T_{s1}$ ,  $T_{s2}$ , and  $T_{s3}$ ) obtained using the basic two-dimensional unsteady heat conduction equation and from Eq. (8), which is obtained using the steady one-dimensional heat conduction equation.
- 3) The numerical solutions for the relationship between  $q$  and  $\Delta T_L$  solved using the RANS equations for unsteady turbulent heat transfer are in good agreement with the experimental data, and the values derived from Eqs. (39) and (40) are within  $\pm 10\%$  difference for  $d = 3$  and  $6$  mm,  $L = 32.7$  to  $100$  mm,  $L/d = 5.51$  to  $33.3$ ,  $T_{in} = 296.47$  to  $362.32$  K,  $Pr = 1.85$  to  $5.89$ ,  $\Delta T_L = 5$  to  $200$  K,  $u = 4.01$  to  $21.79$  m/s, and  $\tau = 15.43$  ms to  $26.98$  s.
- 4) The thickness of the conductive sublayer,  $\delta_{CSL,st}$  and  $\delta_{CSL}$ , for steady-state and transient turbulent heat transfer linearly decreased with increase in  $u$  on a log-log scale, whereas the nondimensional thickness of conductive sublayer,  $(y^+_{CSL,st})_{TEM}$  and  $(y^+_{CSL})_{TEM}$ , for steady-state and transient turbulent heat transfer increased slightly with increase in  $u$  but was nevertheless almost constant over the entire numerical range. These numerical solutions of  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  for steady-state turbulent heat transfer can be expressed for  $L/d = 11$  to  $33.33$ ,  $T_{in} = 302.68$  to  $362.02$  K, and  $Pr = 1.85$  to  $5.89$  for  $\Delta T_L = 40$  to  $120$  K by using Eqs. (45) and (46), and those of  $\delta_{CSL}$  and  $(y^+_{CSL})_{TEM}$  for transient turbulent heat transfer can be expressed for  $\tau = 15.4$  ms to  $6.33$  s for  $\Delta T_L = 40$  to  $120$  K by using Eqs. (47) and (48).

## NOMENCLATURE

|                                                  |                                                            |
|--------------------------------------------------|------------------------------------------------------------|
| $C_{1e}$ , $C_{2e}$ , $C_{3e}$ , $C_d$ , $C_\mu$ | coefficients in approximated turbulent transport equations |
| $d$                                              | test tube inner diameter, m                                |
| $f_F$                                            | Fanning friction factor                                    |
| $G$                                              | $= \rho u$ , mass velocity, kg/m <sup>2</sup> s            |
| $L$                                              | heated length, m                                           |
| $L_e$                                            | entrance length, m                                         |
| $L_{eff}$                                        | effective length, m                                        |

|                    |                                                                           |
|--------------------|---------------------------------------------------------------------------|
| $L_{ex}$           | exit length, m                                                            |
| $l_m$              | length of energy-containing eddies, m                                     |
| $Nu_d$             | $= hd/\lambda_l$ , nusselt number                                         |
| $P_{in}$           | pressure at inlet of heated section, kPa                                  |
| $P_{ipt}$          | pressure measured using inlet pressure transducer, kPa                    |
| $P_{out}$          | pressure at outlet of heated section, kPa                                 |
| $P_{opt}$          | pressure measured using outlet pressure transducer, kPa                   |
| $Pr$               | $= c_p\mu/\lambda$ , Prandtl number                                       |
| $Pr_t$             | $= c_p\mu_t/\lambda_t$ , turbulent Prandtl number                         |
| $Q$                | heat input per unit volume, W/m <sup>3</sup>                              |
| $Q_0$              | initial exponential heat input, W/m <sup>3</sup>                          |
| $q$                | heat flux, W/m <sup>2</sup>                                               |
| $q_{cr,sub,st}$    | steady-state critical heat flux for subcooled condition, W/m <sup>2</sup> |
| $Ra$               | average roughness, $\mu\text{m}$                                          |
| $Re_d$             | $= Gd/\mu_l$ , Reynolds number                                            |
| $R_{max}$          | maximum roughness depth, $\mu\text{m}$                                    |
| $Rz$               | mean roughness depth, $\mu\text{m}$                                       |
| $r_i$              | test tube inner radius, m                                                 |
| $r_o$              | test tube outer radius, m                                                 |
| $(\Delta r)_{out}$ | outer control volume width for $r$ -component, m                          |
| $TEM$              | calculated temperature of outer control volume, K                         |
| $\bar{T}$          | average temperature of test tube, K                                       |
| $T_{f,av}$         | average liquid temperature, K                                             |
| $T_{in}$           | inlet liquid temperature, K                                               |
| $T_L$              | $= (T_{in} + T_{out})/2$ , liquid bulk mean temperature, K                |
| $T_{out}$          | outlet liquid temperature, K                                              |
| $(T_{out})_{cal}$  | calculated outlet liquid temperature, K                                   |
| $T_s$              | heater inner surface temperature, K                                       |
| $T_{so}$           | heater outer surface temperature, K                                       |
| $T_{s,av}$         | average inner surface temperature, K                                      |

|                      |                                                                                                                             |
|----------------------|-----------------------------------------------------------------------------------------------------------------------------|
| $\Delta T_L$         | $= (T_{s,av} - T_L)$ , temperature difference between average inner surface temperature and liquid bulk mean temperature, K |
| $\Delta T_{sat}$     | $= T_s - T_{sat}$ , inner surface superheat, K                                                                              |
| $\Delta T_{sat,so}$  | $= T_s - T_{sat}$ , outer surface superheat, K                                                                              |
| $u$                  | flow velocity, m/s                                                                                                          |
| $u_r$                | $r$ component of velocity vector                                                                                            |
| $u_\theta$           | $\theta$ component of velocity vector                                                                                       |
| $u_z$                | $z$ component of velocity vector                                                                                            |
| $y^+$                | $= y(\tau_w \rho_l)^{0.5} / \nu_l$ , dimensionless normal-distance coordinate                                               |
| $y_{CSL}^+$          | $= (f_F/2)^{0.5} \rho_l u \delta_{CSL} / \mu_l$ , nondimensional thickness of conductive sublayer                           |
| $\Gamma_b$           | volumetric production rate of $k$ by gravitational force                                                                    |
| $\delta$             | thickness of conductive sublayer on nucleate boiling heat transfer                                                          |
| $\delta_{CSL}$       | $= (\Delta r)_{out}/2$ , thickness of conductive sublayer and conductive sublayer on forced convection                      |
| $\varepsilon$        | emissivity and rate of dissipation of turbulent energy, $m^2/s^3$                                                           |
| $\lambda_e$          | $= \lambda_t + \lambda$ , effective turbulent thermal conductivity, W/mK                                                    |
| $\mu_e$              | $= \mu_t + \mu$ , effective turbulent viscosity, $Ns/m^2$                                                                   |
| $\mu_l$              | viscosity, $Ns/m^2$                                                                                                         |
| $\mu_w$              | viscosity at tube wall temperature, $Ns/m^2$                                                                                |
| $\nu_l$              | $= \mu_l / \rho_l$ , kinematic viscosity of fluid, $Ns\ m/kg$                                                               |
| $\rho_l$             | density of fluid, $kg/m^3$                                                                                                  |
| $\sigma_k$           | effective turbulent Prandtl number for transport of $k$                                                                     |
| $\sigma_t$           | effective turbulent Prandtl number                                                                                          |
| $\sigma_\varepsilon$ | effective turbulent Prandtl number for transport of $\varepsilon$                                                           |
| $\tau_w$             | shear stress at the wall, $N/m^2$                                                                                           |

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**Table 1** Dimensions of the experimental test tubes and test sections

| Test tube | Material | $d$ (mm) | $L$ (mm) | $L_{eff}$ (mm) | $L_{12}$ (mm) | $L_{23}$ (mm) | $L_{34}$ (mm) | $L/d$ | $L_{eff}/d$ | $\delta$ (mm) | $L_e$ (mm) | $L_e/d$ | $L_{ex}$ (mm) | $L_{ex}/d$ |
|-----------|----------|----------|----------|----------------|---------------|---------------|---------------|-------|-------------|---------------|------------|---------|---------------|------------|
| THD-F167  | Platinum | 3        | 66.5     | 56.7           | 19.2          | 18.8          | 18.7          | 22.2  | 18.9        | 0.5           | 240        | 80.0    | 120           | 40.00      |
| THD-F174  | Platinum | 3        | 100.0    | 100.0          | -             | -             | -             | 33.3  | 33.3        | 0.5           | 240        | 80.0    | 120           | 40.00      |
| THD-F189  | Platinum | 6        | 69.6     | 59.2           | 19.6          | 20            | 19.6          | 11.6  | 9.87        | 0.4           | 333        | 55.5    | 233           | 38.83      |

**Table 2** Values of constants used in the Chen–Kim  $k$ – $\varepsilon$  turbulence model [9].

| $\sigma_k$ | $\sigma_\varepsilon$ | $\sigma_t$ | $C_{1e}$ | $C_{2e}$ | $C_{3e}$ | $C_\mu$ |
|------------|----------------------|------------|----------|----------|----------|---------|
| 0.75       | 1.3                  | 1.0        | 1.15     | 1.9      | 0.25     | 0.5478  |

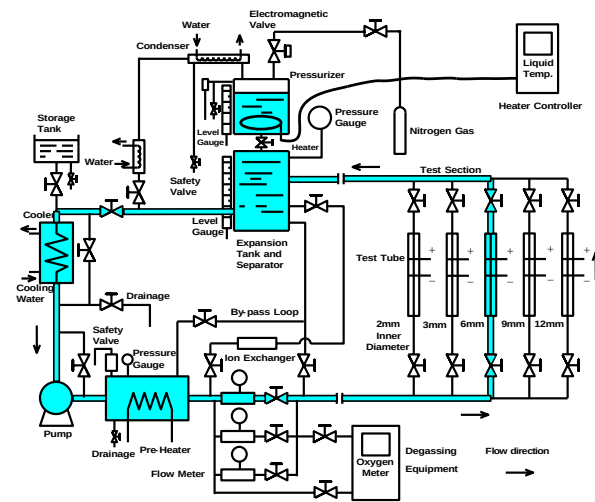


Figure 1. Schematic of the experimental water loop.

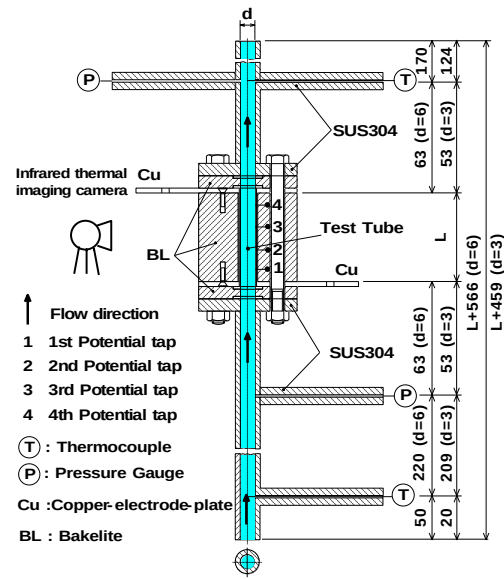


Figure 2. Vertical cross-sectional view of the 3- and 6- mm inner diameter test sections.

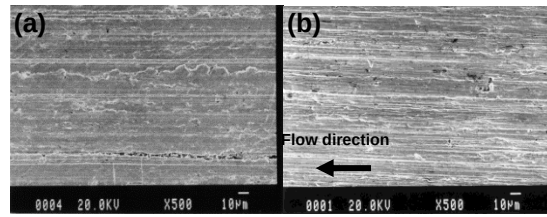


Figure 3. SEM images of the Pt test tubes of inner diameters (a) 3 and (b) 6 mm, both with commercial inner surface finish.



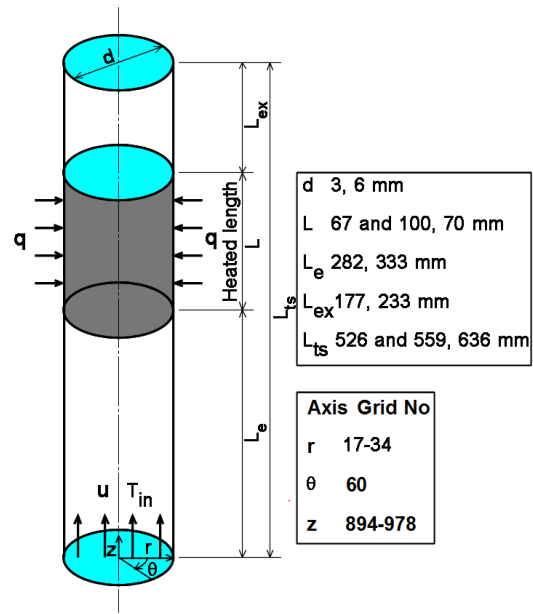


Figure 5. Physical model used in the numerical analysis.

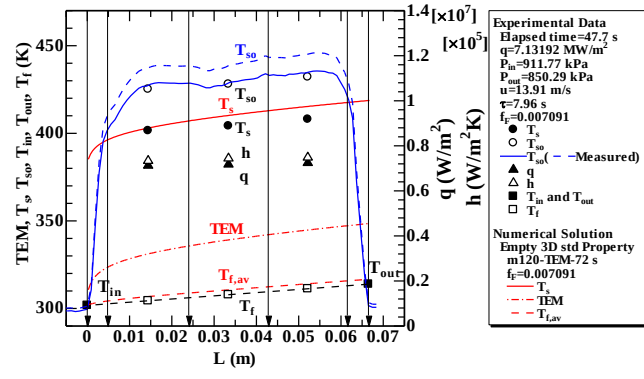


Figure 6.  $T_s$ ,  $T_{so}$ ,  $q$ , and  $h$  at the first, second, and third positions of the three sections and  $T_{in}$  and  $T_{out}$  for  $\tau = 7.96 \text{ s}$  at  $u = 13.91 \text{ m/s}$  compared with the numerical solutions of  $T_s$  and  $T_{so}$  obtained through infrared thermal imaging.

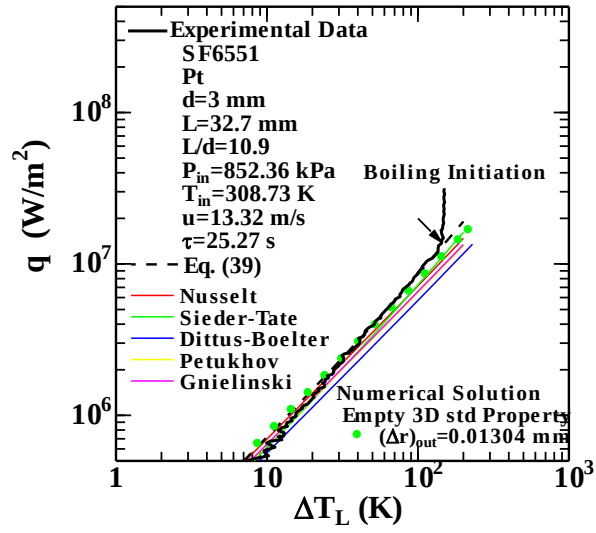


Figure 7. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for circular tube of  $d = 3$  mm and  $L = 32.7$  mm with  $T_{in} = 308.73$  K and  $u = 13.32$  m/s at  $P_{in} = 852.36$  kPa.

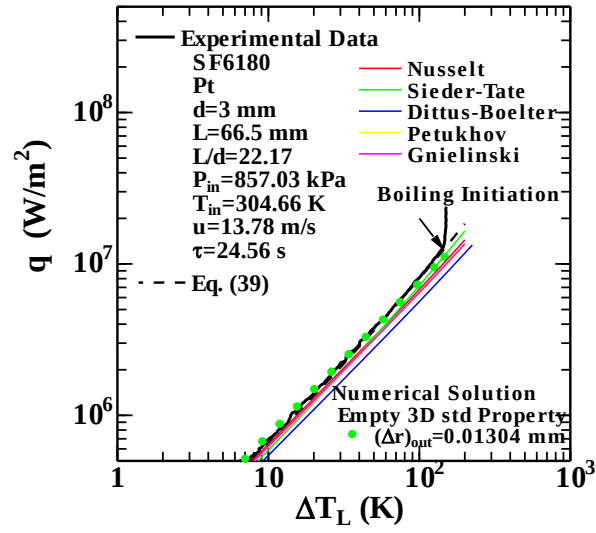


Figure 8. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for circular tube of  $d = 3$  mm and  $L = 66.5$  mm with  $T_{in} = 304.66$  K and  $u = 13.78$  m/s at  $P_{in} = 857.03$  kPa.

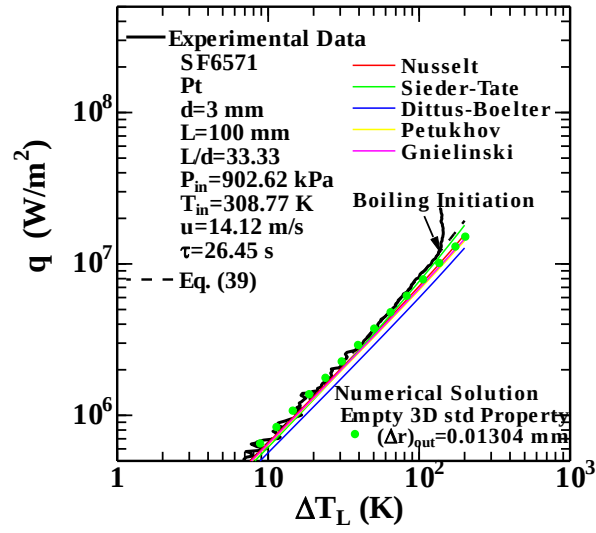


Figure 9. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for circular tube of  $d = 3$  mm and  $L = 100$  mm with  $T_{in} = 308.77$  K and  $u = 14.12$  m/s at  $P_{in} = 902.62$  kPa.

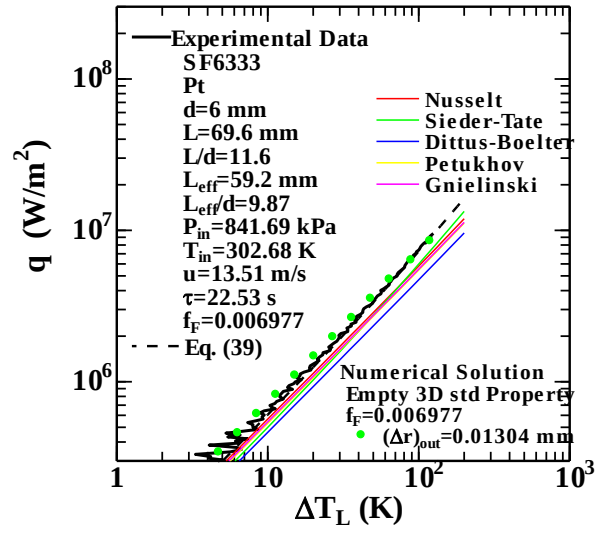


Figure 10. Relationship between  $q$  and  $\Delta T_L [= (T_{s,av} - T_L)]$  for circular tube of  $d = 6$  mm and  $L_{eff} = 59.2$  mm with  $T_{in} = 302.68$  K and  $u = 13.51$  m/s at  $P_{in} = 841.69$  kPa.

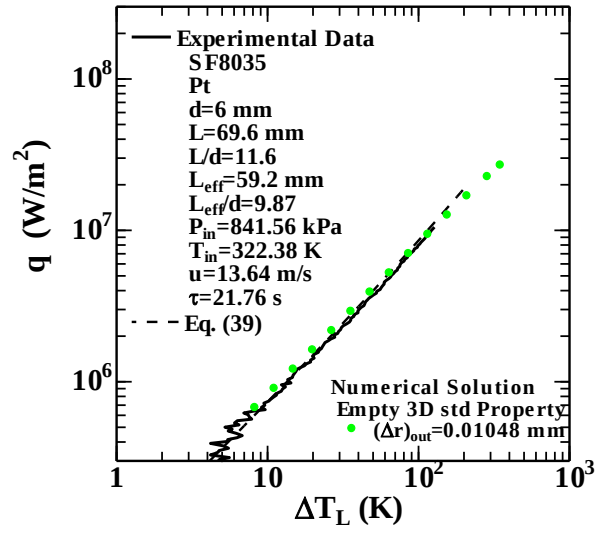


Figure 11. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for circular tube of  $d = 6$  mm and  $L_{eff} = 59.2$  mm with  $T_{in} = 322.38$  K and  $u = 13.64$  m/s at  $P_{in} = 841.56$  kPa.

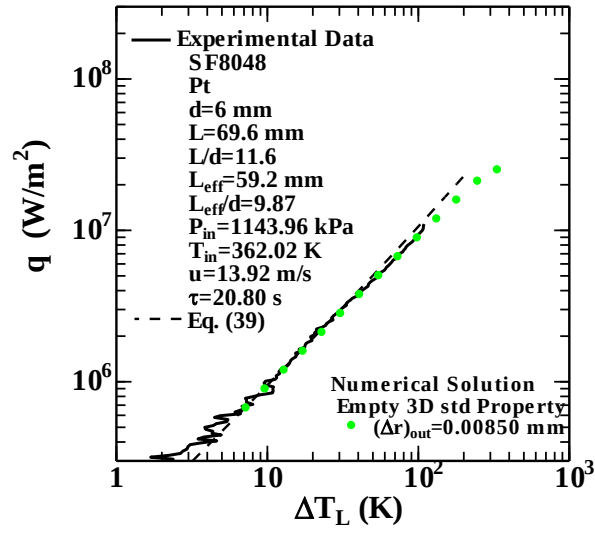


Figure 12. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for circular tube of  $d = 6$  mm and  $L_{eff} = 59.2$  mm with  $T_{in} = 362.02$  K and  $u = 13.92$  m/s at  $P_{in} = 1143.96$  kPa.

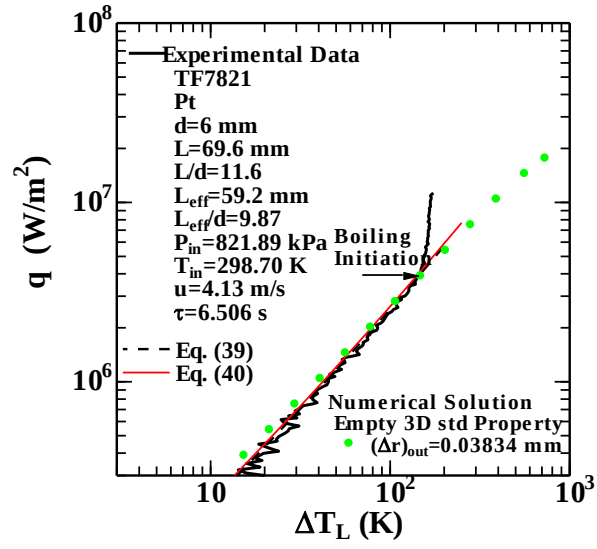


Figure 13. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for Pt test tube of  $d = 6$  mm and  $L_{\text{eff}} = 59.2$  mm with  $\tau = 6.51$  s at  $u = 4.13$  m/s and  $P_{\text{in}} = 822$  kPa.

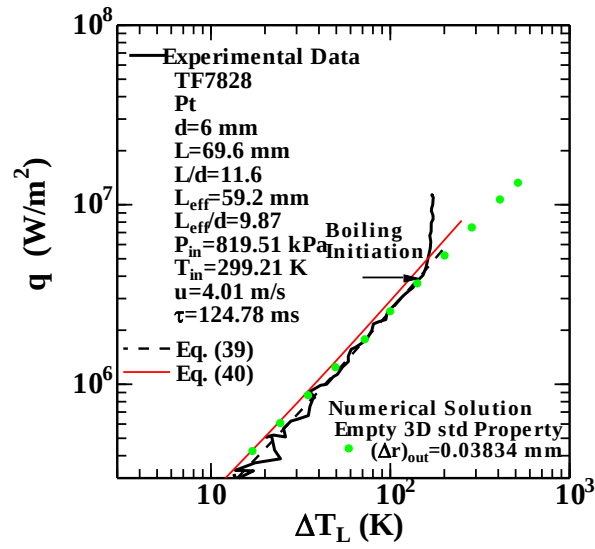


Figure 14. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for Pt test tube of  $d = 6$  mm and  $L_{\text{eff}} = 59.2$  mm with  $\tau = 124.8$  ms at  $u = 4.01$  m/s and  $P_{\text{in}} = 820$  kPa.

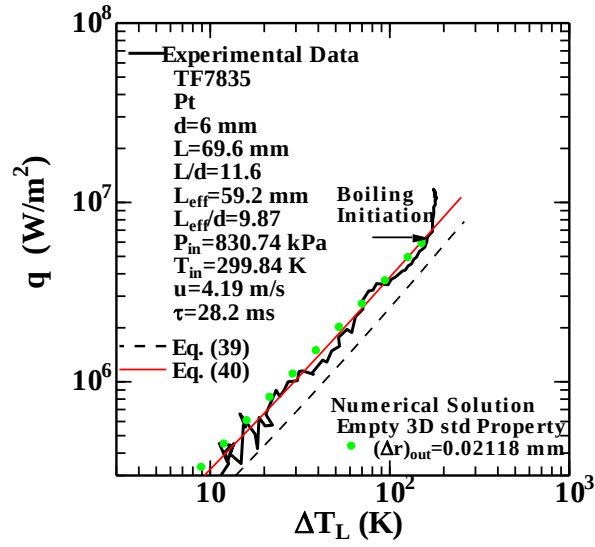


Figure 15. Relationship between  $q$  and  $\Delta T_L [= (T_s - T_L)]$  for Pt test tube of  $d = 6$  mm and  $L_{\text{eff}} = 59.2$  mm with  $\tau = 28.2$  ms at  $u = 4.19$  m/s and  $P_{\text{in}} = 831$  kPa.

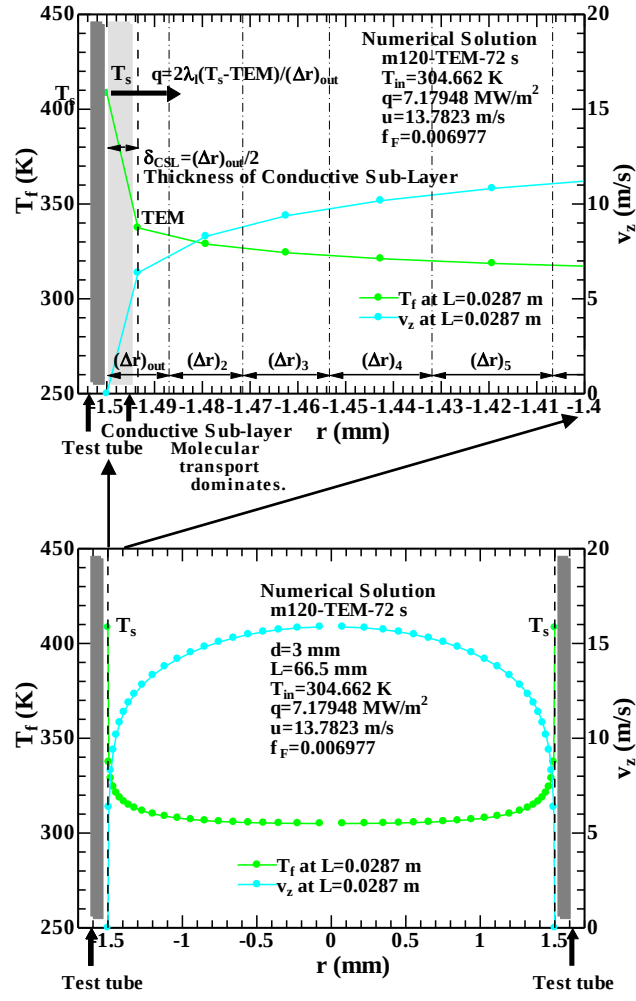


Figure 16. Liquid temperatures in the conductive sublayer,  $\delta_{CSL}$ , obtained using numerically predicted data points (i.e., by solving RANS equations) for the  $d = 3$  mm test tube.

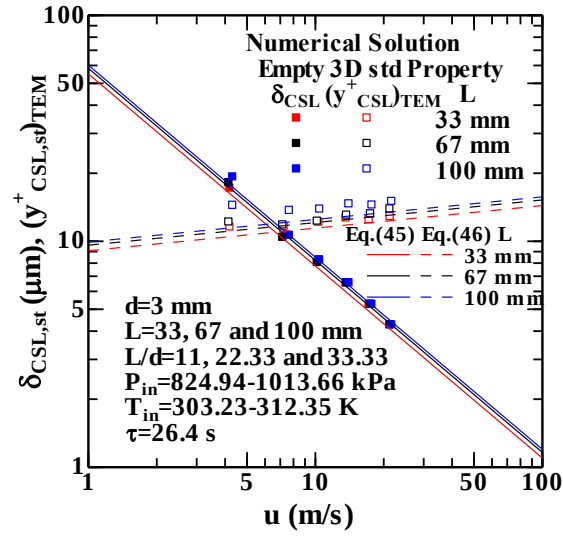


Figure 17. Relationship between  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  for turbulent heat transfer numerically solved for circular tubes of  $d = 3$  mm;  $L = 33, 67$ , and  $100$  mm; and  $u$  at  $\Delta T_L = 100$  K.

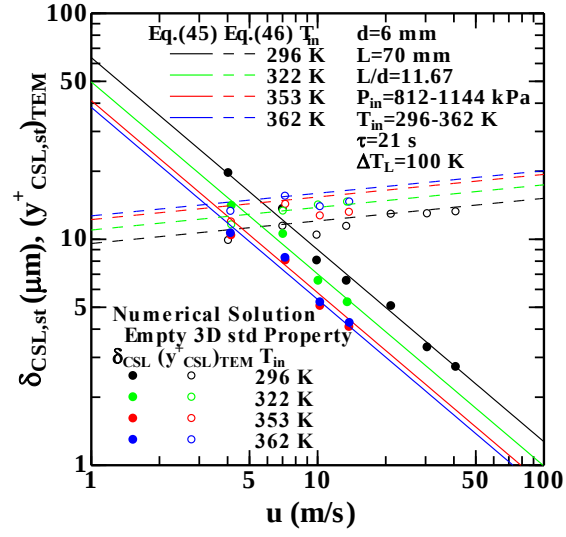


Figure 18. Relationships between  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  for turbulent heat transfer numerically solved for circular tubes of  $d = 6$  mm;  $L = 70$  mm;  $T_{in} = 296, 322, 353$ , and  $362$  K; and  $u$  at  $\Delta T_L = 100$  K.

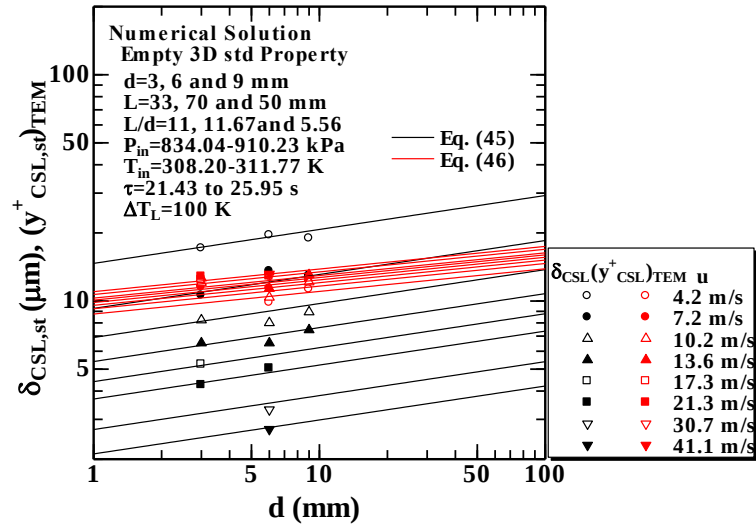


Figure 19. Relationships between  $\delta_{CSL,st}^+$  and  $(y_{CSL,st}^+)^{TEM}$  for turbulent heat transfer numerically solved for circular tube of  $d = 3, 6,$  and  $9$  mm;  $L=33, 70,$  and  $50$  mm;  $T_{in} = 308.20-311.77$  K; and  $d$  at  $\Delta T_L = 100$  K.

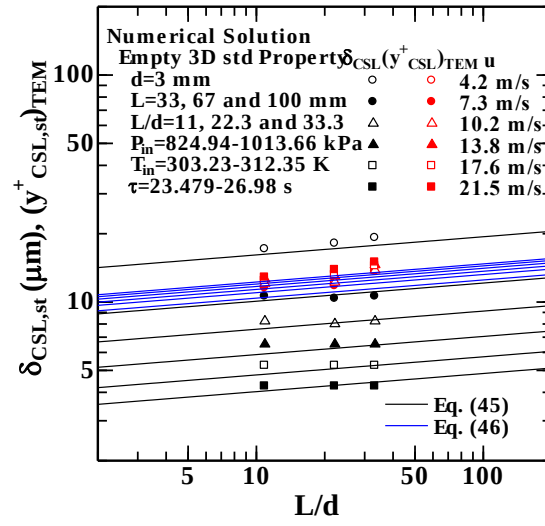


Figure 20. Relationships between  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  for turbulent heat transfer numerically solved for circular tubes of  $d = 3$  mm and  $L = 33, 67$ , and  $100$  mm, and  $L/d$  at  $\Delta T_L = 100$  K.

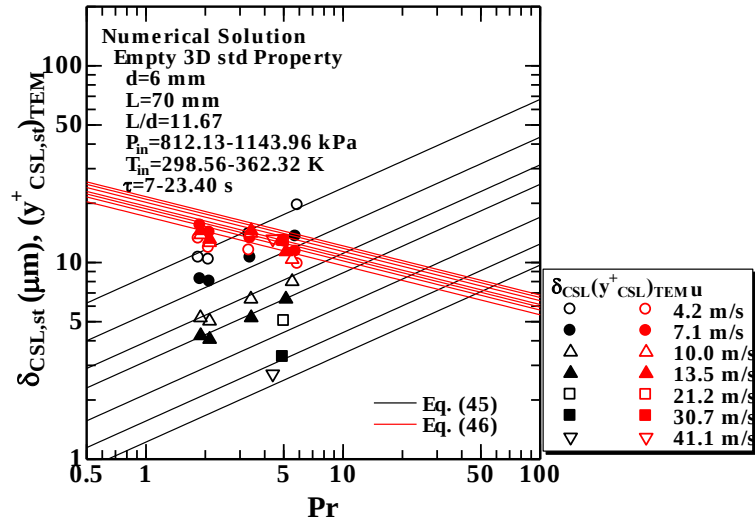


Figure 21. Relationships between  $\delta_{CSL,st}$  and  $(y^+_{CSL,st})_{TEM}$  for turbulent heat transfer numerically solved for circular tube of  $d = 6$  mm,  $L = 70$  mm, and  $L/d = 11.67$ , and Prandtl number (inlet liquid temperature) at  $\Delta T_L = 100$  K.

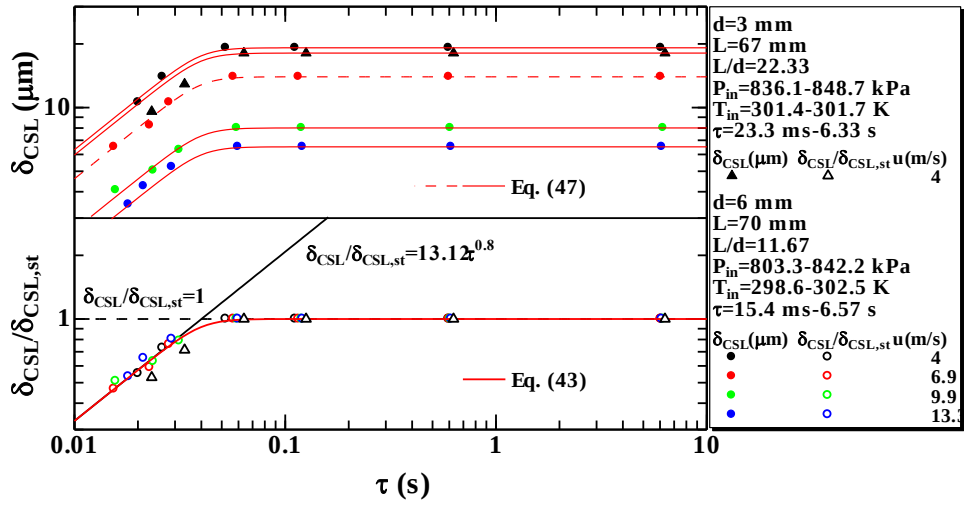


Figure 22. Relationships between  $\delta_{CSL}$  and  $\delta_{CSL}/\delta_{CSL,st}$  for turbulent heat transfer numerically solved for circular tubes of  $d = 6$  mm and  $L = 70$  mm and  $d = 3$  mm and  $L = 67$  mm, and  $\tau$ .

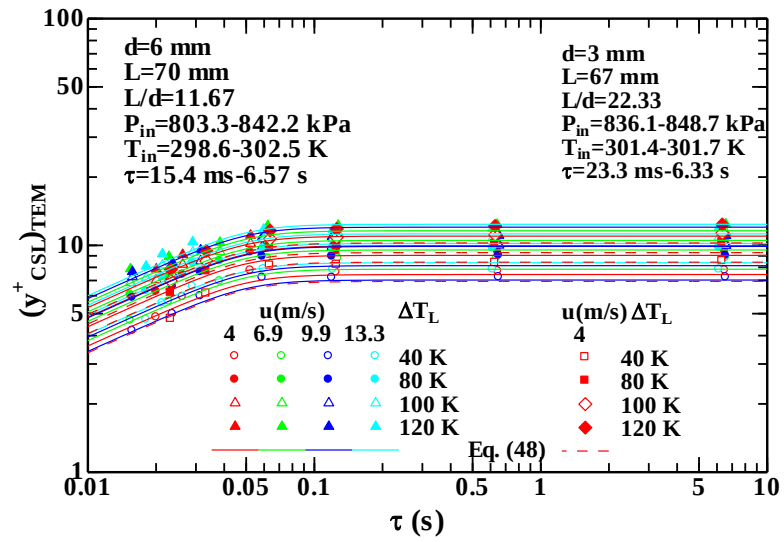


Figure 23. Relationships between  $(y^+_{CSL})_{TEM}$  for turbulent heat transfer numerically solved at  $\Delta T_L$  of 40, 80, 100, and 120 K for circular tubes of  $d = 6$  mm and  $L = 70$  mm and  $d = 3$  mm and  $L = 67$  mm, and  $\tau$ .

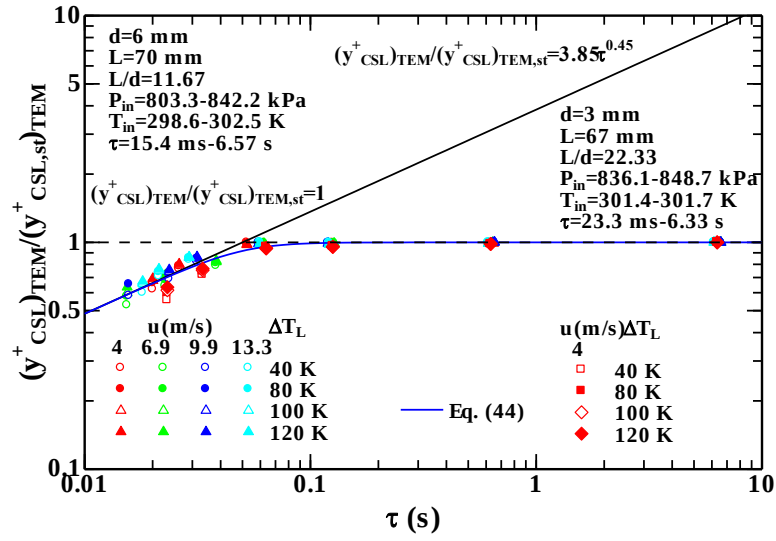


Figure 24. Relationships between  $(y^+_{CSL})_{TEM}/(y^+_{CSL})_{TEM,st}$  for turbulent heat transfer numerically solved at  $\Delta T_L$  of 40, 80, 100, and 120 K for circular tubes of  $d = 6$  mm and  $L = 70$  mm and  $d = 3$  mm and  $L = 67$  mm, and  $\tau$ .