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**CONDUCTIVE SUB-LAYER OF TWISTED-TAPE-INDUCED SWIRL-FLOW HEAT
TRANSFER IN VERTICAL CIRCULAR TUBES WITH VARIOUS TWISTED-TAPE
INSERTS**

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ABSTRACT

Twisted-tape-induced swirl-flow heat transfer due to exponentially increasing heat inputs with various exponential periods ($Q = Q_0 \exp(t/\tau)$, $\tau = 6.04$ to 23.07 s) and twisted-tape-induced pressure drop was systematically measured for various mass velocities ($G = 4115$ to 13656 kg/m²s), inlet liquid temperatures ($T_{in} = 285.88$ to 299.09 K), and inlet pressures ($P_{in} = 847.45$ to 943.29 kPa) using an experimental water loop flow. Measurements were made over a 59.2-mm effective length and three sections (upper, middle, and lower positions), within which four potential taps were spot-welded onto the outer surface of a 6-mm-inner-diameter, 69.6-mm-heated length, 0.4-mm-thickness platinum circular test tube. Type SUS304 twisted tapes with a width $w = 5.6$ mm, a thickness $\delta_T = 0.6$ mm, a total length $l = 372$ mm, and twist ratios $y = 2.39$ and 4.45 were employed in this study. The RANS equations (Reynolds Averaged Navier-Stokes Simulation) with a $k-\varepsilon$ turbulence model for a circular

tube 6 mm in diameter and 636 mm in length were numerically solved for heating of water with a heated section 6 mm in diameter and 70 mm in length using the CFD code, under the same conditions as the experimental ones and considering the temperature dependence of the thermo-physical properties concerned. The theoretical values of surface heat flux q on the circular tubes with twisted tapes with twist ratios y of 2.39 and 4.45 were found to be almost in agreement with the corresponding experimental values of heat flux q , with deviations of less than 30 % for the range of temperature difference between the average heater inner surface temperature and the liquid bulk mean temperature $\Delta T_L [= T_{s,av} - T_L, T_L = (T_{in} + T_{out})/2]$ considered in this study. The theoretical values of the local surface temperature T_s , local average liquid temperature $T_{f,av}$, and local liquid pressure drop ΔP_x were found to be within almost 15 % of the corresponding experimental ones. The thickness of the conductive sub-layer δ_{CSL} and the nondimensional thickness of the conductive sub-layer y^+_{CSL} on the circular tubes with various twisted-tape inserts were determined on the basis of numerical solutions for the swirl velocities u_{sw} ranging from 5.23 to 21.18 m/s. Correlations between the conductive sub-layer thickness δ_{CSL} and the nondimensional thickness of the conductive sub-layer y^+_{CSL} for twisted-tape-induced swirl-flow heat transfer in a vertical circular tube were derived.

1. INTRODUCTION

Computational study of twisted-tape-induced swirl-flow heat transfer in vertical circular tubes with various twisted-tape inserts (*TTI* tubes) is important to investigate the influence of the twist ratio of twisted tape on the swirl velocity and the thickness of the conductive sub-layer and to develop a database for the design of a diverter for a nuclear fusion facility. Little fundamental research has been conducted on twisted-tape-induced swirl-flow heat transfer with various twist ratios, and little is known about the effects of the twist ratio, $y [= H/d = (\text{pitch of } 180^\circ \text{ rotation}) / d]$, on the local surface temperature, T_s , local average liquid temperature, $T_{f,av}$, and local liquid pressure drop, ΔP_x , and the

effect of the swirl velocity, u_{sw} , on the thickness of the conductive sub-layer, δ_{CSL} , and the nondimensional thickness of the conductive sub-layer, y^+_{CSL} , for twisted-tape-induced swirl-flow heat transfer in a vertical circular tube. Knowledge of the thickness of the conductive sub-layer and the nondimensional thickness of the conductive sub-layer would be useful in computational fluid dynamics (CFD) analysis of turbulent heat transfer [1-17].

Recently, the steady-state turbulent heat transfer coefficients in a vertical platinum (Pt) circular test tube with a twisted tape with $y = 3.39$ were systematically determined for swirl velocities $u_{sw} = 5.39, 6.93, 13.38, \text{ and } 18.03$ m/s, inlet liquid temperatures $T_{in} = 300.13 \text{ to } 305.78$ K, inlet pressures $P_{in} = 866.52 \text{ to } 945.86$ kPa, and increasing heat inputs ($Q_{0exp}(t/\tau)$, with τ of 6.04 to 22.33 s). Measurements were made over a 59.2-mm effective length and three sections (upper, middle, and lower positions), within which four potential taps were spot-welded on the outer surface of a Pt test tube with an inner diameter of 6 mm, a 69.6-mm heated length, and a 0.4-mm thickness. The RANS equations for turbulent heat transfer in a circular tube with a 6-mm inner diameter, a 636-mm length, and a twisted-tape insert were numerically solved for heating of water with a heated section with a 6-mm inner diameter and a 70-mm length using the CFD code, under the same conditions as the experimental ones and considering the temperature dependence of the thermo-physical properties concerned. The numerical solutions obtained for the local surface temperature, $(T_s)_z$, local average liquid temperature, $(T_{f,av})_z$, and local liquid pressure drop, ΔP_z , were found to be within $\pm 5\%$ of the corresponding experimental determined values for $(T_s)_z$, $(T_{f,av})_z$, and ΔP_z . The thickness of the conductive sub-layer, $\delta_{CSL} [= (\Delta r)_{out} / 2]$, and the nondimensional thickness of the conductive sub-layer, $y^+_{CSL} [= (f_F/2)^{0.5} \rho_l u \delta_{CSL} / \mu_l]$, for turbulent heat transfer in a circular tube with a twisted-tape insert were clarified based on the numerical solutions [18]. The results of this study confirm that the authors' steady-state turbulent heat transfer correlation for circular tubes with various twisted-tape inserts, can

describe not only experimental data on steady-state turbulent heat transfer for circular tubes with various twisted-tape inserts but also numerical solutions, to within 0 to +20 % accuracy, for wide ranges of temperature differences between the heater inner surface temperature and the liquid bulk mean temperature ($\Delta T_L = 10$ to 140 K) and for swirl-flow velocities $u_{sw} = 5.39$ to 18.03 m/s [18]. This correlation, which was developed on the basis of experimental data [19], is expressed by Eqs. (1) and (2).

$$Nu_d = 0.02 Re_{sw}^{0.85} Pr^{0.4} \left(\frac{L_{eff}}{d} \right)^{-0.08} \left(\frac{\mu_l}{\mu_w} \right)^{0.14} \quad (1)$$

$$Re_{sw} = \frac{\rho_l u_{sw} d}{\mu_l} = Re_d \frac{\pi d^2}{\pi d^2 - 4w\delta_T} \times \frac{(4y^2 + 2\pi^2)^{0.5}}{2y} \quad (2)$$

Turbulent heat transfer for circular tubes with various twisted-tape inserts can be analyzed by introducing a term for the swirl velocity, u_{sw} , instead of the flow velocity, u , in the authors' steady-state turbulent heat transfer correlation for an empty circular tube, Eq. (3), which is based on experimental data [3, 4, 9, 20].

$$Nu_d = 0.02 Re_d^{0.85} Pr^{0.4} \left(\frac{L_{eff}}{d} \right)^{-0.08} \left(\frac{\mu_l}{\mu_w} \right)^{0.14} \quad (3)$$

The present study had the following objectives: (1) to measure the steady-state turbulent heat transfer coefficients for platinum test tubes with twisted tapes with $y = 2.39$ and 4.45, divided into three sections (upper, middle, and lower positions) for wide ranges of test tube surface temperatures (T_s) and swirl velocities (u_{sw}); (2) to obtain numerical solutions for the surface heat flux, q , and surface temperature, T_s , over the heated length, as well as the liquid temperature, T_f , and liquid pressure, P , in a 6-mm-inner-diameter tube with various twisted-tape inserts, from the RANS equations for turbulent heat transfer, under the same conditions as the experimental ones; (3) to compare the theoretical results with experimental data on surface heat flux, q , local surface temperature, $(T_s)_z$, local average liquid temperature, $(T_{f,av})_z$, and local liquid pressure drop, ΔP_z ; (4) to confirm the validity of the component

equations in the turbulent heat transfer correlation for a vertical circular tube with a twisted-tape insert, such as the equations for liquid temperature and pressure change from the inlet to the outlet; (5) to determine the thickness of the conductive sub-layer, $\delta_{CSL} [= (\Delta r)_{out}/2]$, and the nondimensional thickness of the conductive sub-layer, $y^+_{CSL} [= (f_F/2)^{0.5} \rho_l u \delta_{CSL} / \mu_l]$, where f_F is the Fanning friction factor], for turbulent heat transfer in vertical circular tubes with various twisted-tape inserts under controlled velocities; and (6) to examine the influence of the twist ratio, y , and the swirl velocity, u_{sw} , on the thickness of the conductive sub-layer, δ_{CSL} , and the nondimensional thickness of the conductive sub-layer, y^+_{CSL} , in vertical circular tubes with various twisted-tape inserts under controlled velocities.

2. EXPERIMENTAL APPARATUS AND METHOD

A schematic diagram of the experimental water loop comprised of the pressurizer is shown in Fig. 1. The loop is made of SUS304 stainless steel (Steel Use Stainless 18Cr-8Ni) and is capable of producing pressures up to 2 MPa. The loop has five test sections with inner diameters of 2, 3, 6, 9, and 12 mm. The test sections are vertically oriented, with water flowing upward. The test section with the inner diameter of 6 mm was used in this study. The water used was distilled and deionized to obtain a specific resistivity of approximately 20 $\mu\text{S/m}$. The water circulating through the loop was heated or cooled using a pre-heater or cooler as necessary to maintain the desired inlet temperature. The flow velocity was measured using a mass flow meter with a vibration tube (Nitto Seiko, CLEANFLOW 63FS25, flow range = 100 and 750 kg/min). The mass velocity was controlled by regulating the frequency of a three-phase alternating power source to a canned-type circulation pump (Nikkiso Co., Ltd., Non-Seal Pump Multi-stage Type VNH12-C4 C-3S7SP, pump flow rate = 12 m^3/h , pump head = 250 m) with an inverter installed on a 4-digit light-emitting diode (LED) monitor (Mitsubishi Electric Corp., Inverter, Model-F720-30K). The pump input frequency (f_{pi}) shows the net pump input power and pump discharge pressure, free of slip loss. The water was pressurized by saturated vapor in the

pressurizer. The pressure at the outlet of the test tube was controlled to within ± 1 kPa of the desired value using a heater controller for the pressurizer.

A cross-sectional view of the 6-mm-inner-diameter test section used in this study is shown in Fig. 2. A platinum (Pt) test tube with an inner diameter, d , of 6 mm, a heated length, L , of 69.6 mm, a commercial finish applied to the inner surface, and an SUS304 twisted-tape insert was mainly used in this study. Platinum test tubes are highly sensitive to resistance thermometry. The wall thickness of the test tube, δ , was 0.4 mm. Four 0.07-mm-diameter platinum wires were spot-welded onto the outer surface of the test tube as potential taps. The first one was located 5.2 mm from the leading edge of the test tube, and the second, third, and fourth ones were located 24.8, 44.8, and 64.4 mm, respectively, from the leading edge. The effective length, L_{eff} , of the test tube between the first potential tap and the fourth one at which heat transfer was measured was 59.2 mm. Type SUS304 twisted tapes of width $w = 5.6$ mm, thickness $\delta_T = 0.6$ mm, total length $l = 372$ mm, and twist ratios [$y = H/d = (\text{pitch of } 180^\circ \text{ rotation})/d$] = 2.39 and 4.45, as shown in Fig. 3, were mainly employed. Various twisted-tape widths ($w = 5.3, 5.4, 5.5$, and 5.6 mm) have been used to determine the optimum conditions for test tube heating. The surface of the SUS304 twisted tape was uniformly coated with alumina thermal spray to provide electrical insulation. Silver-coated 5-mm-thick copper-electrode plates used to supply a heating current were soldered onto the surfaces of both ends of the test tube. The ends of the test tube were electrically isolated from the loop using Bakelite plates of 14-mm thickness. The inner surface of the test tube was examined using a scanning electron microscope (SEM) photograph (JEOL JXA8600), and the inner surface roughness was measured using Tokyo Seimitsu Co., Ltd.'s surface texture measuring instrument (SURFCOM 120A). Figure 4 shows the SEM photograph of the platinum (Pt) test tube with an inner diameter $d = 6$ mm and a commercial finish on the inner surface. Inner surface roughness values of 0.45, 2.93, and 1.93 μm were measured for R_a , R_{max} , and R_z , respectively.

The Pt test tube with the twisted-tape insert was heated with an exponentially increasing heat input supplied from a direct-current source (Takasago Ltd., NL035-500R, DC 35 V-3000 A) through the two copper electrodes shown in Fig. 5. The average temperature of the Pt test tube, $\bar{T}$, was measured with resistance thermometry participating as a branch of a double bridge circuit for the temperature measurement. The output voltages from the bridge circuit, V_T , together with the voltage drop across the potential taps of the test tube (the first and fourth potential taps, $V_R = IR_T$; the first and second ones, $V_{R1} = IR_{T1}$; the second and third ones, $V_{R2} = IR_{T2}$; and the third and fourth ones, $V_{R3} = IR_{T3}$) and across a standard resistance, $V_I = IR_s$, were amplified (DC amplifier, Yokogawa Electric Corporation, DC amplifier 3131 and 3132) and then sent via an analog–digital (A/D) converter to a digital computer. The unbalance voltage, V_T , is expressed by means of Ohm's law as follows.

$$V_T = \frac{I(R_T \times R_2 - R_1 \times R_3)}{R_2 + R_3} \quad (4)$$

These voltages were simultaneously sampled at constant intervals ranging from 60 to 200 ms. The average temperatures of the Pt test tube between the first and fourth potential taps and between adjacent potential taps (the first and second potential taps, the second and third ones, and the third and fourth ones) were calculated using a previously calibrated resistance–temperature relation, $R_T = a(1 + b\bar{T} + c\bar{T}^2)$. The surface heat fluxes between the first and fourth potential taps and between adjacent potential taps, $q(t)$, $q_1(t)$, $q_2(t)$, and $q_3(t)$, are the differences between the heat generation rate per unit surface area, $Q(t)$, $Q_1(t)$, $Q_2(t)$, and $Q_3(t)$, respectively, and the rate of change of energy storage in the test tube obtained from the faired average temperature versus time curve, expressed as follows:

$$q_n(t) = \frac{V}{S} \left(Q_n(t) - \rho c \frac{d\bar{T}_n(t)}{dt} \right) \quad (5)$$

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

The heater inner surface temperatures between the two electrodes and between adjacent potential taps were also obtained by solving the heat conduction equation for the test tube under the conditions of measured average temperature and surface heat flux of the test tube. All the calculations for the inner surface temperature were made using the PHOENICS code (64 bits) [21]. The basic unsteady two-dimensional heat conduction equation for the test tube is as follows:

$$\rho c \frac{\partial T}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(\lambda r \frac{\partial T}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left(\frac{\lambda}{r} \frac{\partial T}{\partial \theta} \right) + Q(t) \quad (6)$$

The calculation domain from the test tube inner radius, r_i , to the test tube outer radius, r_o , was radially divided into 500 grid points ($\Delta r = 1 \mu\text{m}$), and the time step, Δt , was set to be much smaller than 0.45 ms. The unsteady equation was numerically analyzed for the following boundary conditions.

$$q(t) = -\lambda \frac{\partial T}{\partial r} \Big|_{r=r_i} \quad (7)$$

$$\frac{\partial T}{\partial r} \Big|_{r=r_o} = 0 \quad (8)$$

The solutions for the inner and outer surface temperatures of the test tube, T_s and T_{so} , respectively, are given by the steady one-dimensional heat conduction equation. The basic equation for the test tube is as follows:

$$\frac{d^2 T}{dr^2} + \frac{1}{r} \frac{dT}{dr} + \frac{Q}{\lambda} = 0 \quad (9)$$

Integration yields the mean temperature of the test tube.

$$T(r) = -\frac{Qr^2}{4\lambda} + \frac{Qr_o^2}{2\lambda} \ln r + C \quad (10)$$

$$\bar{T} = \frac{1}{\pi(r_o^2 - r_i^2)} \int_{r_i}^{r_o} 2\pi r T(r) dr \quad (11)$$

Generating heat in the tube is equivalent to thermal conduction, and the test tube is perfectly insulated.

$$q = -\lambda \left. \frac{dT}{dr} \right|_{r=r_i} = \frac{(r_o^2 - r_i^2)Q}{2r_i} \quad (12)$$

$$\left. \frac{dT}{dr} \right|_{r=r_o} = 0 \quad (8)$$

The temperatures of the heater inner and outer surfaces, T_s and T_{so} , of the test tube with a twisted-tape insert and C in Eq. (10) can be expressed as follows:

$$T_s = T(r_i) = \bar{T} - \frac{qr_i}{4(r_o^2 - r_i^2)^2 \lambda} \times \left[4r_o^2 \left\{ r_o^2 \left(\ln r_o - \frac{1}{2} \right) - r_i^2 \left(\ln r_i - \frac{1}{2} \right) \right\} - (r_o^4 - r_i^4) \right] - \frac{qr_i}{2(r_o^2 - r_i^2) \lambda} (r_i^2 - 2r_o^2 \ln r_i) \quad (13)$$

$$T_{so} = T(r_o) = \bar{T} - \frac{qr_i}{4(r_o^2 - r_i^2)^2 \lambda} \times \left[4r_o^2 \left\{ r_o^2 \left(\ln r_o - \frac{1}{2} \right) - r_i^2 \left(\ln r_i - \frac{1}{2} \right) \right\} - (r_o^4 - r_i^4) \right] - \frac{qr_i r_o^2}{2(r_o^2 - r_i^2) \lambda} (1 - 2 \ln r_o) \quad (14)$$

$$C = \bar{T} - \frac{qr_i}{4(r_o^2 - r_i^2)^2 \lambda} \times \left[4r_o^2 \left\{ r_o^2 \left(\ln r_o - \frac{1}{2} \right) - r_i^2 \left(\ln r_i - \frac{1}{2} \right) \right\} - (r_o^4 - r_i^4) \right] \quad (15)$$

where $\bar{T}$, q , λ , r_i , and r_o are the average temperature of the test tube with the twisted-tape insert, the heat flux, thermal conductivity, test tube inner radius and test tube outer radius, respectively.

The swirl velocity, u_{sw} , is defined in terms of the inlet flow velocity, u , taking into consideration the decrease in the flow cross section and the increase in the flow length caused by the twisted tape [22-24]:

$$u_{sw} = u \frac{\pi d^2}{\pi d^2 - 4w\delta_T} \times \frac{(4y^2 + 2\pi^2)^{0.5}}{2y} \quad (16)$$

The inlet and outlet liquid temperatures 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 along the centerline of the tube at the upper and lower stream points of 283 and 63 mm from the test tube inlet and outlet points. The inlet and outlet pressures were measured using strain gauge transducers (Kyowa Electronic Instruments Co., Ltd., PHS-20A, natural frequency: approximately 30 kHz) located near the entrance of the conduit at the upper and lower stream points,

63 mm from the test tube inlet and outlet points. The thermocouples and transducers were installed in the conduits as shown in Fig. 2. The inlet and outlet pressures were calculated from the pressures measured by the inlet and outlet pressure transducers, according to the following equations:

$$P_{in} = P_{ipt} - \left\{ (P_{ipt})_{wnh} - (P_{opt})_{wnh} \right\} \times \frac{L_{ipt}}{L_{ipt} + L + L_{opt}} \quad (17)$$

$$P_{out} = P_{in} - (P_{in} - P_{opt}) \times \frac{L}{L + L_{opt}} \quad (18)$$

where $L_{ipt} = 0.063$ m and $L_{opt} = 0.063$ m. The differential pressure between the conduits at the upper and lower stream points, 63 mm from the test tube inlet and outlet points, were measured using a differential pressure transducer (Kyowa Electronic Instruments Co., Ltd., PD-500GA, natural frequency: approximately 230 kHz).

Experimental errors are estimated to be ± 1 K in inner tube surface temperature and ± 2 % in heat flux. Mass velocity, inlet and outlet subcoolings, inlet and outlet pressures, differential pressure and exponential period were measured within the accuracy ± 2 %, ± 1 K, ± 4 kPa, ± 0.15 kPa and ± 2 %, respectively.

3. NUMERICAL SOLUTION FOR TURBULENT HEAT TRANSFER

3.1. RANS Equations for k - ε Turbulence Model with High-Reynolds-Number Form

The Reynolds Averaged Navier-Stokes equations with Chen-Kim k - ε turbulence model [25] for a circular tube with a 6-mm inner diameter, a 636-mm length, and a twisted-tape insert were numerically solved for heating of water with a heated section 6 mm in diameter and 70 mm long, using the PHOENICS code (64 bits) [21], under the same conditions as the experimental ones, and taking into consideration the temperature dependence of the thermo-physical properties concerned. Twisted tapes of width $w = 5.6$ mm, thickness $\delta_T = 0.6$ mm, total length $l = 372$ mm, pitches of 180° rotation ($H =$

14.34 and 26.7 mm), and twist ratios $y = 2.39$ and 4.45 were analyzed for the same positioning as used in the experiments. The unsteady RANS equations for turbulent heat transfer, expressed in three-dimensional (3D) coordinates as shown in Fig. 6, are as follows [26].

(continuity Equation) Cylindrical coordinates (r, θ, z) :

$$\frac{\partial \rho}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r}(ru_r \rho) + \frac{1}{r} \frac{\partial}{\partial \theta}(u_\theta \rho) + \frac{\partial}{\partial z}(u_z \rho) = 0 \quad (19)$$

(Momentum Equation) Cylindrical coordinates (r, θ, z)

r-component:

$$\frac{\partial}{\partial t}(\rho u_r) + \frac{\partial}{\partial r}(\rho u_r u_r) + \frac{1}{r} \frac{\partial}{\partial \theta}(\rho u_\theta u_r) - \frac{\rho u_\theta^2}{r} + \frac{\partial}{\partial z}(\rho u_z u_r) = -\frac{\partial P}{\partial r} + \frac{1}{r} \frac{\partial}{\partial r}(r \tau_{rr}) + \frac{1}{r} \frac{\partial}{\partial \theta} \tau_{r\theta} - \frac{\tau_{\theta\theta}}{r} + \frac{\partial}{\partial z} \tau_{rz} + \rho g_r \quad (20)$$

θ -component:

$$\frac{\partial}{\partial t}(\rho u_\theta) + \frac{\partial}{\partial r}(\rho u_r u_\theta) + \frac{1}{r} \frac{\partial}{\partial \theta}(\rho u_\theta u_\theta) + \frac{\rho u_r u_\theta}{r} + \frac{\partial}{\partial z}(\rho u_z u_\theta) = -\frac{1}{r} \frac{\partial P}{\partial \theta} + \frac{1}{r^2} \frac{\partial}{\partial r}(r^2 \tau_{r\theta}) + \frac{1}{r} \frac{\partial}{\partial \theta} \tau_{\theta\theta} + \frac{\partial}{\partial z} \tau_{\theta z} + \rho g_\theta \quad (21)$$

z-component:

$$\frac{\partial}{\partial t}(\rho u_z) + \frac{\partial}{\partial r}(\rho u_r u_z) + \frac{1}{r} \frac{\partial}{\partial \theta}(\rho u_\theta u_z) + \frac{\partial}{\partial z}(\rho u_z u_z) = -\frac{\partial P}{\partial z} + \frac{1}{r} \frac{\partial}{\partial r}(r \tau_{rz}) + \frac{1}{r} \frac{\partial}{\partial \theta} \tau_{\theta z} + \frac{\partial}{\partial z} \tau_{zz} + \rho g_z \quad (22)$$

(Energy Equation), Cylindrical coordinates (r, θ, z) :

$$\begin{aligned} \rho \frac{\partial h}{\partial t} + \rho \frac{\partial}{\partial r}(u_r h) + \rho \frac{1}{r} \frac{\partial}{\partial \theta}(u_\theta h) + \rho \frac{\partial}{\partial z}(u_z h) = \\ \frac{1}{r} \frac{\partial}{\partial r} \left\{ \left(\lambda + c_p \frac{\mu_t}{\sigma_t} \right) r \frac{\partial T}{\partial r} \right\} + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left\{ \left(\lambda + c_p \frac{\mu_t}{\sigma_t} \right) \frac{\partial T}{\partial \theta} \right\} + \frac{\partial}{\partial z} \left\{ \left(\lambda + c_p \frac{\mu_t}{\sigma_t} \right) \frac{\partial T}{\partial z} \right\} + Q \end{aligned} \quad (23)$$

(Transport Equation for k), Cylindrical coordinates (r, θ, z) :

$$\begin{aligned} \rho \frac{\partial k}{\partial t} + \rho \frac{\partial}{\partial r}(u_r k) + \rho \frac{1}{r} \frac{\partial}{\partial \theta}(u_\theta k) + \rho \frac{\partial}{\partial z}(u_z k) = \\ \frac{1}{r} \frac{\partial}{\partial r} \left\{ \left(\mu + \frac{\mu_t}{\sigma_k} \right) r \frac{\partial k}{\partial r} \right\} + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left\{ \left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial \theta} \right\} + \frac{\partial}{\partial z} \left\{ \left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial z} \right\} + \rho(P_k + \Gamma_b - \varepsilon) \end{aligned} \quad (24)$$

(Transport Equation for ε), Cylindrical coordinates (r, θ, z) :

$$\begin{aligned} \rho \frac{\partial \varepsilon}{\partial t} + \rho \frac{\partial}{\partial r}(u_r \varepsilon) + \rho \frac{1}{r} \frac{\partial}{\partial \theta}(u_\theta \varepsilon) + \rho \frac{\partial}{\partial z}(u_z \varepsilon) = \\ \frac{1}{r} \frac{\partial}{\partial r} \left\{ \left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) r \frac{\partial \varepsilon}{\partial r} \right\} + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left\{ \left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial \theta} \right\} + \frac{\partial}{\partial z} \left\{ \left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial z} \right\} + \rho \frac{\varepsilon}{k} (C_{1e} P_k + C_{3e} \Gamma_b - C_{2e} \varepsilon) \end{aligned} \quad (25)$$

$$\text{where} \quad \tau_{rr} = 2\rho\nu \frac{\partial u_r}{\partial r} \quad (26)$$

$$\tau_{\theta\theta} = 2\rho\nu \left(\frac{1}{r} \frac{\partial u_\theta}{\partial \theta} + \frac{u_r}{r} \right) \quad (27)$$

$$\tau_{zz} = 2\rho\nu \frac{\partial u_z}{\partial z} \quad (28)$$

$$\tau_{r\theta} = \tau_{\theta r} = \rho\nu \left(r \frac{\partial}{\partial r} \left(\frac{u_\theta}{r} \right) + \frac{1}{r} \frac{\partial u_r}{\partial \theta} \right) \quad (29)$$

$$\tau_{\theta z} = \tau_{z\theta} = \rho\nu \left(\frac{\partial u_\theta}{\partial z} + \frac{1}{r} \frac{\partial u_z}{\partial \theta} \right) \quad (30)$$

$$\tau_{zr} = \tau_{rz} = \rho\nu \left(\frac{\partial u_z}{\partial r} + \frac{\partial u_r}{\partial z} \right) \quad (31)$$

$$g_r = 0, \quad g_\theta = 0, \quad g_z = -g \quad (32)$$

$$h = c_p T \quad (33)$$

$$\nu_t = C_\mu \frac{k^2}{\varepsilon} \quad (34)$$

$$P_k = \nu_t \left[2 \left\{ \left(\frac{\partial u_r}{\partial r} \right)^2 + \left(\frac{1}{r} \frac{\partial u_\theta}{\partial \theta} + \frac{u_r}{r} \right)^2 + \left(\frac{\partial u_z}{\partial z} \right)^2 \right\} + \left(\frac{1}{r} \frac{\partial u_r}{\partial \theta} + \frac{\partial u_\theta}{\partial r} - \frac{u_\theta}{r} \right)^2 + \left(\frac{\partial u_\theta}{\partial z} + \frac{1}{r} \frac{\partial u_z}{\partial \theta} \right)^2 + \left(\frac{\partial u_z}{\partial r} + \frac{\partial u_r}{\partial z} \right)^2 \right] \quad (35)$$

$$\Gamma_b = \frac{-\nu_t}{\rho \sigma_t} \left(g_r \frac{\partial \rho}{\partial r} + g_\theta \frac{1}{r} \frac{\partial \rho}{\partial \theta} + g_z \frac{\partial \rho}{\partial z} \right) \quad (36)$$

$$\lambda_e = \lambda_t + \lambda \quad (37)$$

$$\lambda_t = \frac{c_p \mu_t}{\sigma_t} \quad (38)$$

$$\nu = \frac{\mu}{\rho} + \nu_t \quad (39)$$

Note that u_r , u_θ and u_z are the r , θ , and z components, respectively, of a velocity vector. The constants, σ_k , σ_ε , σ_t , C_{1e} , C_{2e} , C_{3e} , and C_μ that appear in Eqs. (23) to (25), (34), (36) and (38) take the values given in Table 1.

3.2. Boundary Conditions

The fundamental equations were numerically analyzed for the following boundary conditions.

At the outer boundary of the heated section: constant heat flux and no slip.

$$q = -\lambda \frac{\partial T}{\partial r} = \text{constant} \quad (40)$$

At the outer boundary of the non-heated section:

$$\frac{\partial T}{\partial r} = 0 \quad (41)$$

$$\text{At the lower boundary: } T = T_{in}, \quad u_r = 0, \quad u_\theta = 0 \quad \text{and} \quad u_z = u \quad \text{for inflow} \quad (42)$$

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

3.3. Solution Method

The twisted-tape 3D geometry data file was created using the SolidWorks 3D CAD Design Engineering Software Tool [27]. The coded 3D geometry data file was loaded into the PHOENICS-VR editor, and its dimensions and orientation were then converted into cylindrical coordinates. Simple shapes with rectangular and triangular facets formed by the twisted tape were combined to obtain the designed geometry. The object was represented by facets distributed on the outer surfaces of the object, as shown in Fig. 7 (a). The PARSOL function in PHOENICS is a technique that improves the accuracy of flow simulations for situations in which a fluid–solid boundary obliquely intersects some of the cells of a Cartesian or cylindrical coordinate grid, as shown in Fig. 7 (b). PARSOL is capable of calculating fluid flow phenomena accurately, whether the flow is laminar or turbulent, provided that the cell is cut by the

interface into no more than two sub-cells, one containing fluid and the other solid. The control volume discretization equations were derived from these fundamental equations using a hybrid scheme [28]. The thermo-physical properties for each control volume were given as those at each volume temperature. The procedure used for the calculation of the flow field was the SIMPLEST algorithm, which is based on SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) [29].

The surface heat fluxes, q , for the heated length were equally distributed within the range of 2.69×10^5 to 2.34×10^7 W/m² as the initial condition, and the numerical calculation continued until a steady state was obtained. The surface temperature of the test tube with twisted-tape insert, T_s , was calculated from the analyzed temperature of the outer control volume on the test tube surface, TEM , which is supposed to be located at the center of the control volume, by solving the following equation for heat conduction in liquid:

$$T_s = \frac{q}{\lambda_l} \frac{(\Delta r)_{out}}{2} + TEM \quad (43)$$

where, $(\Delta r)_{out}$ is the outer control volume width on the r -component. The average heat transfer coefficient for the test tube surface with a twisted-tape insert was obtained by $T_{s,av}$ averaging of the calculated local surface temperatures, $(T_s)_z$, every 0.5 mm along the heated length, L . All the calculations were made using the PHOENICS code (64 bits) [21].

4. EXPERIMENTAL RESULTS AND DISCUSSION

4.1. Experimental Conditions and Parameters used for Calculation

Steady-state heat transfer processes caused by exponentially increasing heat input, $Q_0 \exp(t/\tau)$, were measured for Pt circular test tubes with SUS304 twisted-tape inserts. The exponential periods, τ , of the heat input ranged from 6.04 to 23.07 s. A decrease in the exponential period means an increase in the rate of increasing heat input. The initial experimental conditions for the pressure drop experiment and

the turbulent heat transfer experiment, such as the mass velocity, inlet liquid temperature, outlet pressure, and exponential period, were determined independently before each experimental run.

The experimental conditions were as follows:

Test tube number	THD-F169
Test tube material	Platinum
Surface condition	Commercial finish of inner surface
Surface roughness values Ra , $Rmax$, and Rz	0.45, 2.93, and 1.93 μm
Inner diameter (d)	6 mm
Heated length (L)	69.6 mm
Effective length (L_{eff})	59.2 mm
L_{12}	19.6 mm
L_{23}	20.0 mm
L_{34}	19.6 mm
L/d	11.6
L_{eff}/d	9.87
Wall thickness (δ)	0.4 mm
Twisted-tape number	TT5 and TT8
Twisted-tape material	SUS304
Width (w)	5.6 mm
Thickness (δ_T)	0.6 mm,
Total length (l)	372 mm
Pitch of 180° rotation (H)	14.34 and 26.7 mm
Twist ratio ($\gamma = H/d$)	2.39 and 4.45

Electric insulation	Alumina thermal spraying
Pump input frequency (f_{pi})	10.50–27.35 Hz
Mass velocity (G)	4115–13656 kg/m ² s
Inlet flow velocity (u)	4.142, 7.164, 10.17, and 13.673 m/s for $y = 2.39$ 4.122, 7.126, 10.121, and 13.538 m/s for $y = 4.45$
Axial velocity (u_a)	4.68 to 15.52 m/s
Swirl velocity (u_{sw})	6.42, 11.10, 15.76, and 21.18 m/s for $y = 2.39$ 5.23, 9.04, 12.84, and 17.17 m/s for $y = 4.45$
Reynolds number (Re_d)	2.221×10^4 to 1.003×10^5
Reynolds number based on swirl velocity (Re_{sw})	3.442×10^4 to 1.328×10^5
Inlet pressure (P_{in})	847.45–943.29 kPa
Outlet pressure (P_{out})	821.88–845.07 kPa
Inlet subcooling ($\Delta T_{sub,in}$)	146.37–161.24 K
Outlet subcooling ($\Delta T_{sub,out}$)	135.97–155.79 K
Differential pressure (ΔP)	8.77–73.04 kPa
Inlet liquid temperature (T_{in})	285.88–299.09 K
Exponentially increasing heat input (Q)	$Q_0 \exp(t/\tau)$, $\tau = 6.04$ – 23.07 s

The parameters used for calculation were as follows:

Inner diameter (d)	6 mm
Heated length (L)	70 mm
Entrance length (L_e)	333 mm
Exit length (L_{ex})	233 mm

Test section length (L_{ts})	636 mm
Twisted tape	Solid with smooth-wall friction
Width (w)	5.6 mm
Thickness (δ_T)	0.6 mm
Total length (l)	372 mm
Pitch of 180° rotation (H)	14.34 and 26.7 mm
Twist ratio ($y = H/d$)	2.39 and 4.45
Heat flux (q)	2.69×10^5 to 2.34×10^7 W/m ² ($q_0 \exp(t/\tau)$, $\tau = 20.04$ – 21.14 s)
Inlet flow velocity (u)	4.122–13.673 m/s
Inlet liquid temperature (T_{in})	285.88–299.09 K
Coordinate system	Cylindrical coordinates (r, θ, z)
Control volume	Cut-cell algorithm
Control volume number	(28 to 32, 60, 978)
Physical model	k – ε turbulence model with high-Reynolds-number form
Outer control volume width for r component ($(\Delta r)_{out}$)	9.72 to 34.2 μ m

4.2. Inlet and Outlet Pressures, P_{in} and P_{out} , and Pressures Measured by the Inlet and Outlet Pressure Transducers, P_{ipt} and P_{opt} , for the TTI Tube

Figures 8 and 9 show the z -axis variations of the pressures P_{ipt} and P_{opt} for test tubes with twisted tapes with $y = 2.39$ and 4.45 measured by the inlet and outlet pressure transducers, respectively, and the inlet and outlet pressures P_{in} and P_{out} , respectively calculated using Eqs. (17) and (18), for inlet liquid temperatures, T_{in} , of 288.55 and 295.17 K, and swirl velocities, u_{sw} , of 15.76 and 12.84 m/s, and Fanning friction factors, f_F , of 0.02218 and 0.01507, respectively. The value of P_{ipt} at a distance of $Z = 0.27$ m from the inlet of the test section remained almost constant over

the whole experimental range, as this pressure was controlled to within ± 1 kPa of the desired value using the heater controller of the pressurizer. The value of P_{opt} at $Z = 0.466$ m was considerably lower, as shown in the figures. As Fig. 8 shows, values of P_{in} and P_{out} of 908.24 and 826.60 kPa, respectively, were obtained from the P_{ipt} and P_{opt} values of 981.71 and 753.13 kPa, respectively, at an elapsed time of 127.2 s, using Eqs. (17) and (18), respectively. The value of P_{opt} was adjusted to 753.13 kPa ($Z=0.466$ m) from the measured value of 784.47 kPa to match the pressure rise at the twisted-tape exit obtained from the numerical solution mentioned below. The local liquid pressure drop, ΔP_z , along the heated length, L , of the test section was almost 81.63 kPa. The numerical solutions for the liquid pressure, P , in the test section are shown as solid red line in the figures for comparison. The pressure was 485.63 kPa at the entrance of the twisted tape at $Z = 0.092$ m and decreased linearly with increasing distance along the z -axis, to almost 19.75 kPa at the exit of the twisted tape at $Z = 0.455$ m. The calculated pressure drop along the heated length ($P_{in} = 180.28$ kPa, $P_{out} = 86.22$ kPa, and $\Delta P_z = 94.06$ kPa) was 15.2 % higher than the experimentally measured pressure drop ($\Delta P_z = 81.63$ kPa). In Fig. 9, values of P_{in} and P_{out} of 856.27 and 802.72 kPa, respectively, were obtained from the P_{ipt} and P_{opt} values of 904.46 and 754.53 kPa, respectively. The value of P_{opt} was adjusted to 754.53 kPa ($Z=0.466$ m) from the measured value of 778.83 kPa. The calculated pressure drop along the heated length ($P_{in} = 111.46$ kPa, $P_{out} = 47.91$ kPa, and $\Delta P_z = 63.55$ kPa) was 18.7 % higher than that measured experimentally ($\Delta P_z = 53.55$ kPa). These results reflect the fact that the authors derived Eqs. (17) and (18) by linear interpretation of calculated inlet and outlet pressures, P_{in} and P_{out} , of a heated section, based on non-boiling heat transfer to the critical heat flux (CHF) point through nucleate boiling heat transfer. The numerical solutions have some error associated with them because of the difference between the experimental and computational conditions. In addition, the pressures

of the inlet and outlet pressure transducers, P_{ipt} and P_{opt} , were measured at high Jakob numbers in this study.

4.3. Inner and Outer Surface Temperatures, T_s and T_{so} , Heat Fluxes, q , and Heat Transfer Coefficients, h , for 1st, 2nd, and 3rd Positions of Three Sections, and Inlet and Outlet Liquid Temperatures, T_{in} and T_{out} for *TTI* Tube

Figures 10 and 11 show the inner surface temperatures (T_{s1} , T_{s2} , and T_{s3} , indicated by solid black circles), outer surface temperatures (T_{so1} , T_{so2} , and T_{so3} , indicated by open black circles), heat fluxes (q_1 , q_2 , and q_3 , indicated by solid black triangles), and heat transfer coefficients (h_1 , h_2 , and h_3 , indicated by open black triangles) of the test tubes with twisted tapes with $y = 2.39$ and 4.45 for the first, second, and third positions of the three sections between the first and second potential taps, the second and third ones, and the third and fourth ones, as well as the inlet and outlet liquid temperatures (T_{in} and T_{out} , indicated by solid black squares) at the swirl velocities, u_{sw} , of 15.76 and 12.84 m/s, with heat fluxes, q , of 7.17 and 7.15 MW/m², respectively. The liquid temperatures at the center of each section (heated lengths, L , of 15 , 34.8 , and 54.6 mm, indicated by open black squares) were linearly estimated from the values of the inlet liquid temperature, T_{in} , and the outlet liquid temperature, T_{out} . The inner surface temperatures (T_{s1} , T_{s2} , and T_{s3}) and outer surface temperatures (T_{so1} , T_{so2} , and T_{so3}) were obtained from Eqs. (13) and (14) using the measured average temperature of the test tube and the surface heat flux. The inner and outer surface temperatures for each position in the sections (T_{s1} , T_{s2} , and T_{s3} and T_{so1} , T_{so2} , and T_{so3}) gradually increase as the heated length increases from the leading edge of the test tube with the twisted-tape insert, whereas the heat fluxes (q_1 , q_2 , and q_3) are almost constant for each position in the sections. The inner surface temperature exhibits nearly the same increasing trend as the liquid temperature from the inlet to the outlet (T_{in} to T_{out}). The heat transfer coefficients for

each position in the three sections (h_1 , h_2 , and h_3) become almost constant, as shown in the figures.

The numerical solutions for the z-axis variations in the analyzed liquid temperature of the outer control volume of the test tube surface, TEM , are indicated for heat fluxes of 7.17 and 7.15 MW/m² and swirl velocities of 15.76 and 12.84 m/s by red dot-dashed lines. The surface temperatures of the test tubes, T_s , with twisted tapes with $y = 2.39$ and 4.45 were calculated from the analyzed liquid temperatures of the outer control volume of the test tube surface, TEM , which is supposed to be located at the center of the control volume, by solving the equation for heat conduction in liquid, as given by Eq. (43). The z-axis variations in the inner surface temperature of the test tube with the twisted-tape insert at 0.5-mm intervals along the heated length, L , are indicated by solid red lines in the figures. These temperatures gradually increase with increasing distance along the z-axis from the leading edge of the test tube. The numerical solutions for the inner surface temperatures, obtained from the RANS equations for turbulent heat transfer, Eqs. (19) to (42), are in good agreement with the experimental values given by Eq. (13), which were obtained from the steady one-dimensional heat conduction equation.

The numerical solutions for the average liquid temperatures, $T_{f,av}$, in the test tubes with twisted tapes with $y = 2.39$ and 4.45 from the inlet to the outlet are indicated by dashed red lines in Figs. 10 and 11 for comparison. The average liquid temperatures, $T_{f,av}$, were calculated from the analyzed liquid temperatures of the control volumes on the r - and θ -axis grid numbers of the r - θ plane for each z-axis grid number. These temperatures increase linearly with increase distance along the z-axis and become almost equal to the outlet liquid temperature, T_{out} , at the exit of the

heated length. These results confirm that the heat transfer coefficients, h , calculated from the heat fluxes, q , and the local liquid temperatures, T_f , at the heated lengths, L , of 15, 34.8, and 54.6 mm, which were obtained by linear interpretation between the inlet and outlet liquid temperatures, T_{in} and T_{out} , are also valid for the test tube with the twisted-tape insert, although the inlet and outlet liquid temperatures were measured using 1-mm-outer-diameter sheathed K-type thermocouples in 6-mm-diameter tubes with twisted tapes with $y = 2.39$ and 4.45.

4.4. Steady-state Turbulent Heat Transfer Characteristics of the *TTI* Tube

Figures 12 to 15 show typical examples of the steady-state turbulent heat transfer curves for a circular platinum tube with an inner diameter $d = 6$ mm and effective length $L_{eff} = 59.2$ mm for a twisted-tape insert with $y = 2.39$ at swirl velocities, u_{sw} , of 6.42 and 15.76 m/s, as well as those for a twisted-tape insert with $y = 4.45$ at swirl velocities, u_{sw} , of 5.23 and 12.84 m/s, for an exponential period, τ , of approximately 23 s. Quasi steady-state heat transfer processes on the circular platinum test tube have been confirmed to be caused by an exponentially increasing heat input, $Q_0 \exp(t/\tau)$, with an exponential period, τ , greater than 800 ms. The experimental data were compared with values derived from the authors' correlation for steady-state turbulent heat transfer for an empty tube, Eq. (3), at inlet flow velocities, u , of 3.98 and 9.89 m/s. The heat fluxes gradually increase as the temperature difference between the average inner surface temperature and liquid bulk mean temperature, ΔT_L ($= T_{s,av} - T_L$), increases, parallel to the steady-state turbulent heat transfer curve derived from Eq. (3), up to a point at which the slope begins to increase with heat flux, following the onset of nucleate boiling. The experimentally obtained values for an inner tube diameter $d = 6$ mm at the high heat flux point shown in Figs. 12 to 15 are 25.4 to 60.3 % higher than the values derived from the correlation assuming a fixed temperature difference between the average inner surface temperature and the liquid bulk mean

temperature ($\Delta T_L = \text{constant}$), and 26.02 to 60.31 K lower than the values derived from the correlation assuming a fixed heat flux ($q = \text{constant}$). The curves derived from Eqs. (1) and (2) are shown as solid red lines in Figs. 12 to 15. The turbulent heat transfer values for $d = 6$ mm, $L_{eff} = 59.2$ mm, and $L_{eff}/d = 9.87$ for twisted tapes with $y = 2.39$ and 4.45 are within 0 to +30 % of the values given by Eqs. (1) and (2) for the wide ranges in temperature difference between the heater inner surface temperature and liquid bulk mean temperature ($\Delta T_L = 10$ to 137 K) and swirl velocity ($u_{sw} = 5.23$ to 21.18 m/s), although those gradually become greater than the values derived from Eqs. (1) and (2) as ΔT_L decreases.

The numerical solutions for the relation between the heat flux, q , and the temperature difference between the average inner surface temperature and the liquid bulk mean temperature, ΔT_L , are shown for test tubes with twisted tapes with $y = 2.39$ and 4.45 at inlet flow velocities of 3.98 and 9.89 m/s as solid green circles. The 11 to 14 different values for the numerical solutions are plotted for heat fluxes ranging from 3×10^5 to 3.15×10^7 on the log–log graph. These values also increase with increasing ΔT_L along the curve derived from Eqs. (1) and (2). The values obtained using the RANS equations for turbulent heat transfer, Eqs. (19) to (42), are in good agreement with the experimentally measured values, and the values derived from Eqs. (1) and (2) differ by 0 to +20 %.

Figures 16 and 17 show the time variations in the inlet and outlet pressures calculated by Eqs. (17) and (18), P_{in} and P_{out} , heat flux, q , and heater inner surface temperature, T_s , for the platinum test tubes of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tapes of the twist ratio, y , of 2.39 and 4.45 at $P_{out}=826.60$ and 819.71 kPa, $\Delta T_{sub,in}=160.31$ and 151.6 K, $u_{sw}=15.76$ and 12.84 m/s and

$\tau=22.61$ and 22.39 s, respectively. The values of P_{in} and P_{out} keep almost constant in the whole experimental range. Figures 18 and 19 show the time variations in the inlet and outlet liquid temperatures, T_{in} and T_{out} , the inlet and outlet subcoolings measured, $\Delta T_{sub,in}$ and $\Delta T_{sub,out}$, q and T_s for the platinum test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tapes of the twist ratio, y , of 2.39 and 4.45 at $P_{out}=826.60$ and 819.71 kPa, $\Delta T_{sub,in}=160.31$ and 151.6 K, $u_{sw}=15.76$ and 12.84 m/s and $\tau=22.61$ and 22.39 s. The values of T_{in} and $\Delta T_{sub,in}$ keep almost constant in the whole experimental range, although those of T_{out} and $\Delta T_{sub,out}$ become gradually higher and lower with an increase in the heat flux respectively.

4.5. Thickness of the Conductive Sub-layer, δ_{CSL} , and the Nondimensional Thickness of the Conductive Sub-layer, y^+_{CSL} , for Steady-state Turbulent Heat Transfer in a *TTI* Tube

The values obtained using the RANS equations for steady-state turbulent heat transfer, Eqs. (19) to (42), are in good agreement with the experimentally measured values, and the values derived from Eqs. (1) and (2) differ by 0 to +20 %, as shown in Figs. 12 to 15. As Fig. 20 shows, the liquid temperatures on the test tube surface in the conductive sub-layer [1, 2] decrease linearly with the radius of the heat conduction from the surface temperature on the test tube with the twisted-tape insert, $T_f = T_s - \Delta r q / \lambda_l$. Let T_f equal the analyzed temperature of the outer control volume on the test tube surface, TEM , in the inner region of the turbulent flow, which is supposed to be located at the center of the control volume, as given by Eq. (43). Half of the outer control volume width for the r component, $(\Delta r)_{out}/2$, becomes the thickness of the conductive sub-layer, δ_{CSL} , for turbulent heat transfer in a vertical circular tube, according to a two-phase model that distinguishes between a conductive sub-layer and an inner region of turbulent flow. The relationships between δ_{CSL} and the nondimensional thickness of the conductive sub-layer,

y^+_{CSL} , for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 , and swirl velocity u_{sw} are shown for a temperature difference of 100 K between the average inner surface temperature and the liquid bulk mean temperature, $\Delta T_L (= T_{s,av} - T_L)$, by blue, green, and red symbols, respectively, in Fig. 21. Two nondimensional thicknesses of the conductive sub-layer, $(y^+_{CSL})_{TEM}$ and $(y^+_{CSL})_{TL}$, are given, as it is very difficult to solve for the analyzed temperature of the outer control volume on the test tube surface with a twisted-tape insert, TEM . All of the parameters in the equation were evaluated at the analyzed temperature of the outer control volume on the test tube surface, TEM , and the liquid bulk mean temperature, $TL = (T_{in} + T_{out})/2$. The nondimensional thickness of the conductive sub-layer, y^+_{CSL} , is defined as follows:

$$y^+_{CSL} = \left(\frac{f_F}{2} \right)^{0.5} \frac{\rho_l u \delta_{CSL}}{\mu_l} \quad (44)$$

where f_F is the Fanning friction factor. The values of δ_{CSL} decrease linearly as the swirl velocity, u_{sw} , increases, on the log-log graph. Those of $(y^+_{CSL})_{TEM}$ and $(y^+_{CSL})_{TL}$ increase slightly as the swirl velocity, u_{sw} , increases but are almost constant over the whole numerical range. The numerical solutions for $(\Delta r)_{out}$, δ_{CSL} , $(y^+_{CSL})_{TEM}$, and $(y^+_{CSL})_{TL}$ for circular tubes with twisted tapes with $y = 2.39$ and 4.45 are given in Table 2 for u_{sw} values ranging from 5.23 to 21.18 m/s. These solutions can be expressed by the following correlations:

$$\delta_{CSL} = 59.12 \times 10^{-6} u_{sw}^{-0.85} \quad (45)$$

$$(y^+_{CSL})_{TEM} = 12.93 u_{sw}^{0.1} \quad \text{for } \Delta T_L = 100 \text{ K} \quad (46)$$

$$(y^+_{CSL})_{TL} = 7.32 u_{sw}^{0.1} \quad \text{for } \Delta T_L = 100 \text{ K} \quad (47)$$

The numerical solutions for δ_{CSL} , $(y^+_{CSL})_{TEM}$, and $(y^+_{CSL})_{TL}$ for an empty circular tube are indicated with black symbols in the figure for comparison. These can be expressed for u values ranging from 4.11 to 41.07 m/s by the following correlations [3, 4, 9]:

$$\delta_{CSL} = 65 \times 10^{-6} u^{-0.85} \quad (48)$$

$$(y_{CSL}^+)_{TEM} = 9.08 u^{0.1} \quad \text{for } \Delta T_L = 100 \text{ K} \quad (49)$$

$$(y_{CSL}^+)_{TL} = 5.50 u^{0.1} \quad \text{for } \Delta T_L = 100 \text{ K} \quad (50)$$

Equations (45) and (48), $1/\delta_{CSL} = h \propto u_{sw}^{0.85}$ and $u^{0.85}$, derived from the numerical solutions for liquid temperatures on the test tube surface, represent the authors' steady-state turbulent heat transfer correlations, Eqs. (1) and (3), derived from experimental data for circular tubes with various twisted-tape inserts and empty tubes, respectively.

The numerical solutions for $(y_{CSL}^+)_{TEM}$ and $(y_{TBL}^+)_{TL}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 are also plotted against the Reynolds number based on the swirl velocity and the Reynolds number, Re_{sw} and Re_d , respectively, for temperature differences of 40, 80, 100, and 120 K between average inner surface temperature and liquid bulk mean temperature, ΔT_L , as indicated by the blue, green, and red symbols in Figs. 22 and 23. These numerical solutions for $(y_{CSL}^+)_{TEM}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 can be expressed for Re_{sw} values ranging from 3.106×10^4 to 1.572×10^5 by the following correlations:

$$(y_{CSL}^+)_{TEM} = 2.93 e^{\frac{\Delta T_L}{180}} Re_{sw}^{0.1} \quad \text{for } \Delta T_L = 40, 80, 100, \text{ and } 120 \text{ K} \quad (51)$$

$$(y_{CSL}^+)_{TL} = 2.93 Re_{sw}^{0.1} \quad \text{for } \Delta T_L = 40, 80, 100, \text{ and } 120 \text{ K} \quad (52)$$

The numerical solutions for $(y_{CSL}^+)_{TEM}$ and $(y_{TBL}^+)_{TL}$ for empty circular tubes are also shown for comparison and are indicated by black symbols. These values can be expressed for Re_d values ranging from 2.71×10^4 to 3.74×10^5 by the following correlations [3, 4, 9]:

$$(y_{CSL}^+)_{TEM} = 2.23 e^{\frac{\Delta T_L}{210}} Re_d^{0.1} \quad \text{for } \Delta T_L = 40, 80, 100, \text{ and } 120 \text{ K} \quad (53)$$

$$(y_{CSL}^+)_{TL} = 2.23 Re_d^{0.1} \quad \text{for } \Delta T_L = 40, 80, 100, \text{ and } 120 \text{ K} \quad (54)$$

The relationships between the thickness of the conductive sub-layer, δ_{CSL} , for numerically solved steady-state turbulent heat transfer and the twist ratio, y , are given by Eqs. (16) and (45) and can be expressed as follows:

$$\delta_{CSL} = 59.12 \times 10^{-6} u_{sw}^{-0.85} = 59.12 \times 10^{-6} \left[1.1349u \frac{(4y^2 + 19.739)^{0.5}}{2y} \right]^{-0.85} \quad (55)$$

In Fig. 24, the curves for δ_{CSL} and u_{sw} calculated from Eqs. (55) and (16) are drawn for various inlet flow velocities, u . At a fixed inlet flow velocity, the thickness of the conductive sub-layer increases and the swirl velocity decreases as the twist ratio increases, and both are almost constant for twist ratios greater than 10. The numerical solutions for δ_{CSL} and u_{sw} for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 at u values of $3.98, 6.93, 9.89$, and 13.32 m/s are also shown for comparison and are indicated by black and red symbols in the figure. These curves are in fairly good agreement with the numerical solutions for δ_{CSL} and u_{sw} for circular 6-mm-inner-diameter tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 .

5. CONCLUSIONS

Steady-state turbulent heat transfer in vertical circular platinum (Pt) test tubes with twisted tapes with $y = 2.39$ and 4.45 was systematically measured for swirl velocities $u_{sw} = 5.23$ to 21.18 m/s, inlet liquid temperatures $T_{in} = 285.88$ to 299.09 K, inlet pressures $P_{in} = 847.45$ to 943.29 kPa, and increasing heat inputs ($Q_{0exp}(t/\tau)$, where $\tau = 6.04$ to 23.07 s). Measurements were made over a 59.2-mm effective length and three sections (upper, middle, and lower positions), within which four potential taps were spot-welded on the outer surface of a Pt test tube with a 6-mm inner diameter, a 69.6-mm heated length, and a 0.4-mm thickness. The RANS equations for turbulent heat transfer in a circular tube 6 mm in diameter and 636 mm long with a twisted-tape insert were numerically solved for heating of water with a heated section 6 mm in diameter and

70 mm long using the PHOENICS code, under the same conditions as the experimental ones and considering the temperature dependence of the thermo-physical properties concerned. The experimental and computational study results lead to the following conclusions.

- 1) The numerically determined liquid pressure, P , on the test section is 485.63 kPa at the entrance of the twisted tape at $Z = 0.092$ m, for the circular tube with a twisted tape with $y = 2.39$. The liquid pressure decreases linearly with increasing distance along the z -axis and is almost 19.75 kPa at the exit of the twisted tape at $Z = 0.455$ m, although the calculated pressure drop over the heated length ($\Delta P_z = 94.06$ kPa) was 15.2 % higher than that indicated by the experimental measurements ($\Delta P_z = 81.63$ kPa). The calculated pressure drop over the heated length ($\Delta P_z = 63.55$ kPa) was 18.7 % higher than that measured experimentally ($\Delta P_z = 53.55$ kPa) for the circular tube with a twisted tape with $y = 4.45$.
- 2) The inner and outer surface temperatures for each position in the three sections (T_{s1} , T_{s2} , and T_{s3} and T_{so1} , T_{so2} , and T_{so3}) gradually increase as the heated length from the leading edge of the test tube with the twisted-tape insert increases, whereas the heat fluxes (q_1 , q_2 , and q_3) remain almost constant for each position in the sections. The inner surface temperature exhibits nearly the same increasing trend as the liquid temperature from the inlet to the outlet (T_{in} to T_{out}). The heat transfer coefficients for each position in the three sections (h_1 , h_2 , and h_3) become almost constant.
- 3) The numerical solutions for the inner surface temperatures obtained using the RANS equations for turbulent heat transfer, Eqs. (19) to (42), are in good agreement with the experimental values (T_{s1} , T_{s2} , and T_{s3}) given by Eq. (13), which were obtained from the steady one-dimensional heat conduction equation. The calculated values for the average liquid

temperatures, $T_{f,av}$, in a test tube with a twisted-tape insert from inlet to outlet increase linearly with increasing distance along the z -axis distance, Z , and become almost equal to the outlet liquid temperature, T_{out} , as measured by a 1-mm-outer-diameter sheathed K-type thermocouple at the exit of the heated length.

- 4) The numerical solutions for the relation between the heat flux, q , and the temperature difference between the average inner surface temperature and the liquid bulk mean temperature, ΔT_L , obtained using the RANS equations for turbulent heat transfer, Eqs. (19) to (42), are in good agreement with the experimental data, and the values derived from Eqs. (1) and (2) differ by 0 to +20 % for heat flux values, q , ranging from 3×10^5 to 3.15×10^7 W/m² at swirl velocities, u_{sw} , ranging from 5.23 to 21.18 m/s.
- 5) The thickness of the conductive sub-layer, δ_{CSL} , which is half of the outer control volume width for the r component, $(\Delta r)_{out}/2$, decreases linearly as the swirl velocity, u_{sw} , increases, on the log-log graph. The values of the nondimensional thickness of the conductive sub-layer, $(y^+_{CSL})_{TEM}$ and $(y^+_{CSL})_{TL}$, increase slightly as the swirl velocity, u_{sw} , increases but are almost constant over the whole numerical range. The numerical solutions for δ_{CSL} , $(y^+_{CSL})_{TEM}$, and $(y^+_{CSL})_{TL}$ can be expressed for u_{sw} values ranging from 5.23 to 21.18 m/s using Eqs. (45) to (47), (51), and (52).
- 6) Relationships between the thickness of the conductive sub-layer, δ_{CSL} , for the numerically solved steady-state turbulent heat transfer and the twist ratio, y , are also given. The curves calculated using Eq. (55) are almost in agreement with the numerical solutions for δ_{CSL} for 6-mm-inner-diameter circular tubes and twisted tapes with $y = 2.39, 3.39$, and 4.45 .

6. NOMENCLATURE

$C_{1e}, C_{2e}, C_{3e}, C_{\mu}$ coefficients in approximated turbulent transport equations

d test tube inner diameter, m

f_F Fanning friction factor

f_{pi} pump input frequency, Hz

h turbulent heat transfer coefficient, W/m^2K , and enthalpy, kJ/kg

k turbulent kinetic energy, m^2/s^2

L heated length, m

L_e entrance length, m

L_{eff} effective length, m

L_{ex} exit length, m

L_{ts} test section length, m

l_e entrance length of the test tube with twisted-tape insert, m

$Nu_d = hd/\lambda_l$, Nusselt number

P_{in} pressure at inlet of heated section, kPa

P_{ipt} pressure measured by inlet pressure transducer, kPa

P_k volumetric production rate of k by shear force

P_{out} pressure at outlet of heated section, kPa

P_{opt} pressure measured by outlet pressure transducer, kPa

$Pr = c_p\mu/\lambda$, Prandtl number

$Pr_t = c_p\mu_t/\lambda_t$, turbulent Prandtl number

Q heat input per unit volume, W/m^3

Q_0 initial exponential heat input, W/m^3

q heat flux, W/m^2

q_0	initial exponential heat flux, W/m ²
Re_d	$= Gd/\mu_l$, Reynolds number
Re_{sw}	$= \rho_l u_{sw} d/\mu_l$, Reynolds number based on swirl velocity
r	radial distance in cylindrical coordinates and test tube radius, 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
$\bar{T}$	average temperature, K
TEM	analyzed temperature of the outer control volume, K
T_f	liquid temperature, 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
TTI	twisted-tape insert
T_s	heater inner surface temperature, K
$T_{s,av}$	average inner surface temperature, K
t	time, s
ΔT_L	$= (T_{s,av} - T_L)$, temperature difference between average inner surface temperature and liquid bulk mean temperature, K
u	inlet flow velocity, m/s
u_{sw}	swirl velocity, m/s

w	width of twisted tape, m
y	$= H / d = (\text{pitch of } 180^\circ \text{ rotation}) / d$, twist ratio of twisted tape
y^+_{CSL}	$= (f_F / 2)^{0.5} \rho_l u \delta_{CSL} / \mu_b$, nondimensional thickness of the conductive sub-layer
Z	length, m
z	rectangular coordinate, m
Γ_b	volumetric production rate of k by gravitational force
δ	wall thickness, mm
δ_T	thickness of twisted tape, mm
δ_{CSL}	$= (\Delta r)_{out} / 2$, thickness of the conductive sub-layer, m
ε	emissivity and rate of dissipation of turbulent energy, m^2/s^3
θ	angle in cylindrical coordinates, radian
λ	molecular thermal conductivity, W/mK
λ_e	$= \lambda_t + \lambda$, effective turbulent thermal conductivity, W/mK
λ_l	thermal conductivity, W/mK
λ_t	turbulent thermal conductivity, W/mK
μ	molecular viscosity, Ns/m^2
μ_e	$= \mu_t + \mu$, effective turbulent viscosity, Ns/m^2
μ_t	turbulent viscosity, Ns/m^2
μ_w	viscosity at the temperature of the tube wall, Ns/m^2
ν	$= \mu / \rho + \nu_b$, effective kinetic viscosity, m^2/s
σ_k	effective turbulent Prandtl number for transport of k

σ_t effective turbulent Prandtl number

σ_ε effective turbulent Prandtl number for transport of ε

τ exponential period, s

Subscript

CSL conductive sub-layer

in inlet

l liquid

out outlet

TEM analyzed temperature of the outer control volume, K

w wall

wnh with no heating

7. ACKNOWLEDGMENTS

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Table and Figure Captions

Table 1 Values of the constants in the Chen–Kim k – ε turbulence model [26].

Table 2 Numerical solutions of $(\Delta r)_{out}$, δ_{CSL} , $(y^+_{CSL})_{TEM}$, and $(y^+_{CSL})_{TL}$ for circular tubes with twisted tapes with $y = 2.39$ and 4.45 .

Fig. 1 Schematic diagram of experimental water loop.

Fig. 2 Vertical cross-sectional view of a 6-mm-inner-diameter test section with a twisted-tape insert.

Fig. 3 Photograph of the SUS304 twisted tape coated with alumina thermal spray.

Fig. 4 SEM photograph of the platinum test tube with an inner diameter of 6 mm and commercial finish of the inner surface.

Fig. 5 Measurement and data processing system.

Fig. 6 Physical model for numerical analysis.

Fig. 7 Twisted-tape 3D geometry data (a) in the PHOENICS-VR editor and (b) for the cylindrical coordinate grid.

Fig. 8 Z-axis distributions of pressures, P_{ipt} , P_{opt} , P_{in} , and P_{out} , for circular tube with twisted tape with $y = 2.39$, $T_{in} = 288.35$ K, $u_{sw} = 15.76$ m/s, and $f_F = 0.022$, compared with numerical solutions for liquid pressure, P .

Fig. 9 Z-axis distributions of pressures, P_{ipt} , P_{opt} , P_{in} , and P_{out} , for circular tube with twisted tape with $y = 4.45$, $T_{in} = 295.17$ K, $u_{sw} = 12.84$ m/s, and $f_F = 0.015$, compared with numerical solutions for liquid pressure, P .

Fig. 10 T_s , T_{so} , q , and h for three sections and T_{in} and T_{out} for circular tube with twisted tape with $y = 2.39$ and $u_{sw} = 15.76$ m/s, compared with numerical solutions for TEM , T_{in} , and T_{out} .

Fig. 11 T_s , T_{so} , q , and h for three sections and T_{in} and T_{out} for circular tube with twisted tape with

$y = 4.45$ and $u_{sw} = 12.84$ m/s, compared with numerical solutions for TEM , T_{in} , and T_{out} .

Fig. 12 Relationship between q and ΔT_L for circular tube with $d = 6$ mm and $L_{eff} = 59.2$ mm for twisted tape with $y = 2.39$, $u_{sw} = 6.42$ m/s, and $P_{in} = 847.45$ kPa.

Fig. 13 Relationship between q and ΔT_L for circular tube with $d = 6$ mm and $L_{eff} = 59.2$ mm for twisted tape with $y = 2.39$, $u_{sw} = 15.76$ m/s, and $P_{in} = 908.24$ kPa.

Fig. 14 Relationship between q and ΔT_L for circular tube with $d = 6$ mm and $L_{eff} = 59.2$ mm for twisted tape with $y = 4.45$, $u_{sw} = 5.23$ m/s, and $P_{in} = 830.65$ kPa.

Fig. 15 Relationship between q and ΔT_L for circular tube with $d = 6$ mm and $L_{eff} = 59.2$ mm for twisted tape with $y = 4.45$, $u_{sw} = 12.84$ m/s, and $P_{in} = 856.27$ kPa.

Fig. 16 Time variations in P_{in} , P_{out} , q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 2.39 at $P_{out}=826.60$ kPa, $\Delta T_{sub,in}=160.31$ K, $u_{sw}=15.76$ m/s and $\tau=22.61$ s.

Fig. 17 Time variations in P_{in} , P_{out} , q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 4.45 at $P_{out}=819.71$ kPa, $\Delta T_{sub,in}=151.6$ K, $u_{sw}=12.84$ m/s and $\tau=22.39$ s.

Fig. 18 Time variations in T_{in} , T_{out} , $\Delta T_{sub,in}$, $\Delta T_{sub,out}$, q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 2.39 at $P_{out}=826.60$ kPa, $\Delta T_{sub,in}=160.31$ K, $u_{sw}=15.76$ m/s and $\tau=22.61$ s.

Fig. 19 Time variations in T_{in} , T_{out} , $\Delta T_{sub,in}$, $\Delta T_{sub,out}$, q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 4.45 at $P_{out}=819.71$ kPa, $\Delta T_{sub,in}=151.6$ K, $u_{sw}=12.84$ m/s and $\tau=22.39$ s.

Fig. 20 Liquid temperatures in the conductive sub-layer for circular tube with $d = 6$ mm and twisted-tape insert.

Fig. 21 Relationships between δ_{CSL} , $(y^+_{CSL})_{TEM}$, and $(y^+_{CSL})_{TL}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 and u_{sw} at $\Delta T_L = 100$ K.

Fig. 22 Values of $(y^+_{CSL})_{TEM}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 and Re_{sw} at $\Delta T_L = 40, 80, 100$, and 120 K.

Fig. 23 Values of $(y^+_{CSL})_{TL}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 and Re_{sw} at $\Delta T_L = 40, 80, 100$, and 120 K.

Fig. 24 Relationship between δ_{CSL} and u_{sw} for numerically solved turbulent heat transfer and y at $u = 3.98, 6.93, 9.89$, and 13.32 m/s.

Table 1 Values of the constants in the Chen–Kim k – ε turbulence model [26].

σ_k	σ_ε	σ_t	C_{1e}	C_{2e}	C_{3e}	C_μ
0.75	1.3	1.0	1.15	1.9	0.25	0.5478

Table 2 Numerical solutions of $(\Delta r)_{out}$, δ_{CSL} , $(y^+_{CSL})_{TEM}$, and $(y^+_{CSL})_{TL}$ for circular tubes with twisted tapes with $y = 2.39$ and 4.45

	$y = 2.39$				$y = 4.45$			
u_{sw} (m/s)	6.42	11.10	15.76	21.18	5.23	9.04	12.84	17.17
$(\Delta r)_{out}$ (μm)	27.96	16.48	13.04	10.48	34.2	17.32	12.86	9.72
δ_{CSL} (μm)	13.98	8.24	6.52	5.24	17.1	8.66	6.43	4.86
$(y^+_{CSL})_{TEM}$	16.61	17.06	18.56	20.01	14.34	14.62	14.26	15.04
$(y^+_{CSL})_{TL}$	9.520	9.278	10.29	11.31	9.133	8.510	7.970	8.352

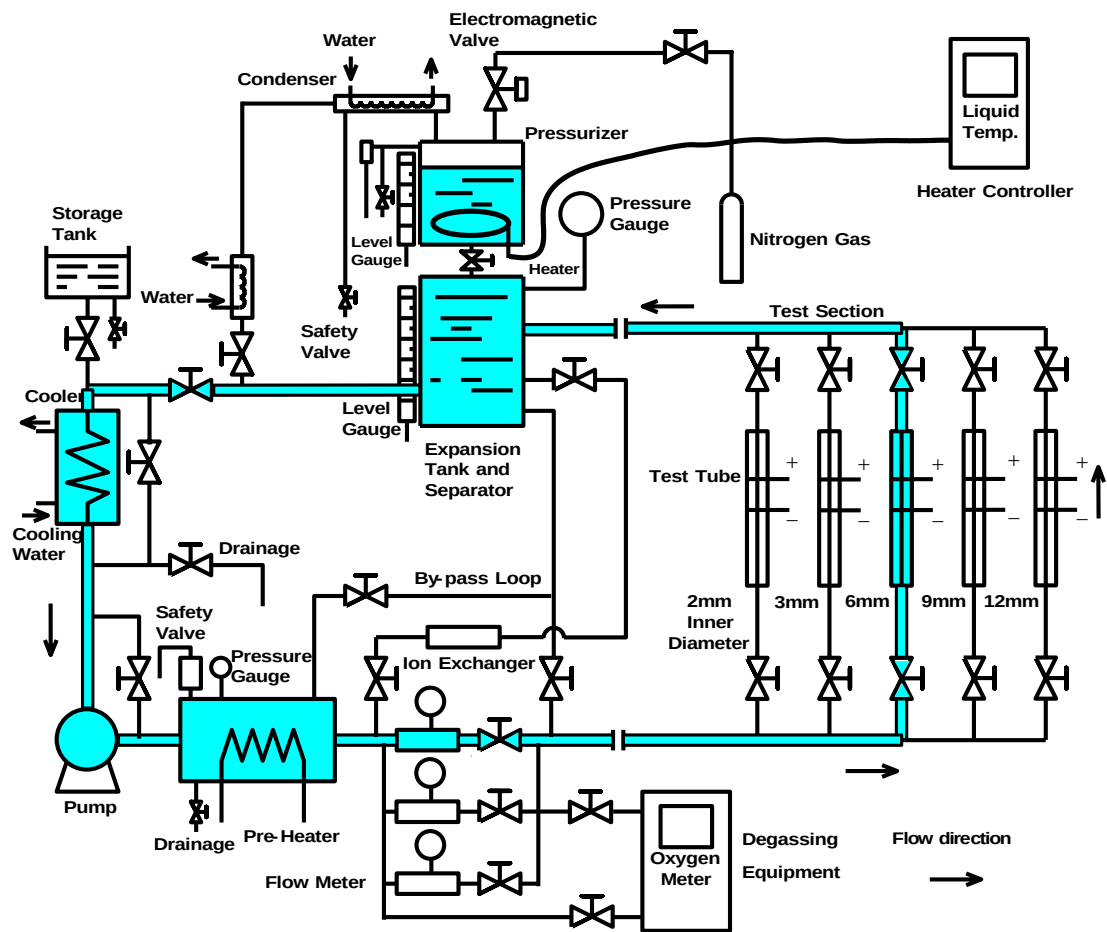


Fig. 1 Schematic diagram of experimental water loop.

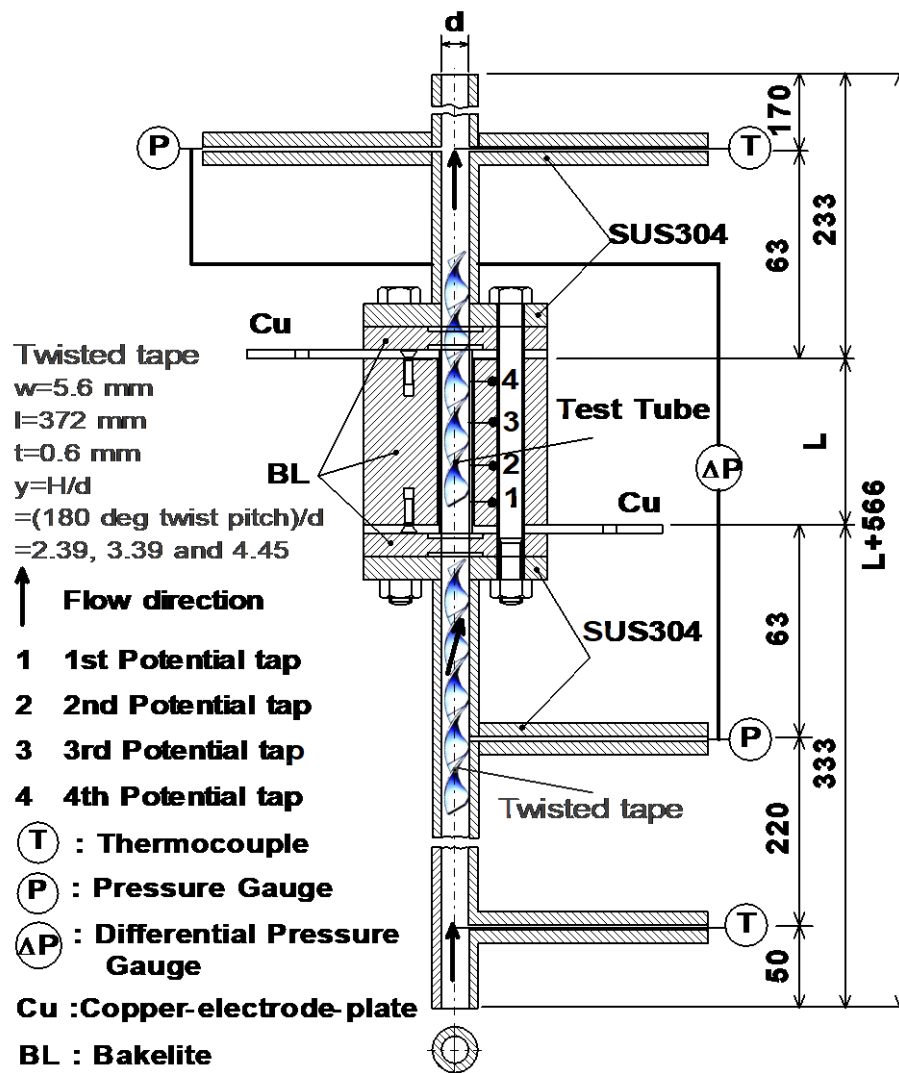


Fig. 2 Vertical cross-sectional view of a 6-mm-inner-diameter test section with a twisted-tape insert.

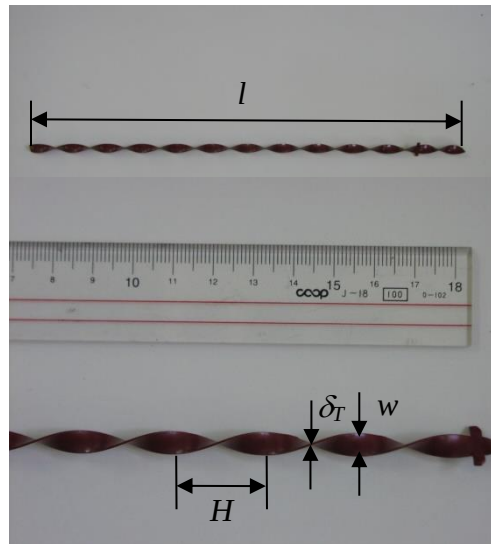


Fig. 3 Photograph of the SUS304 twisted tape coated with alumina thermal spray.

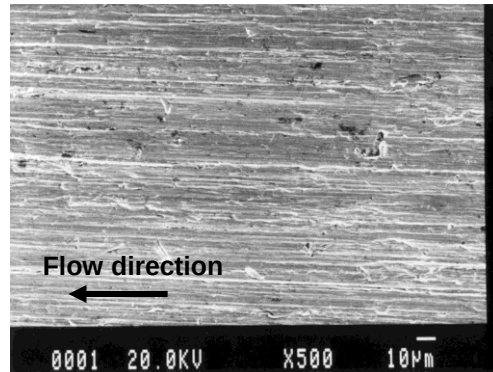


Fig. 4 SEM photograph of the platinum test tube with an inner diameter of 6 mm and commercial finish of the inner surface.

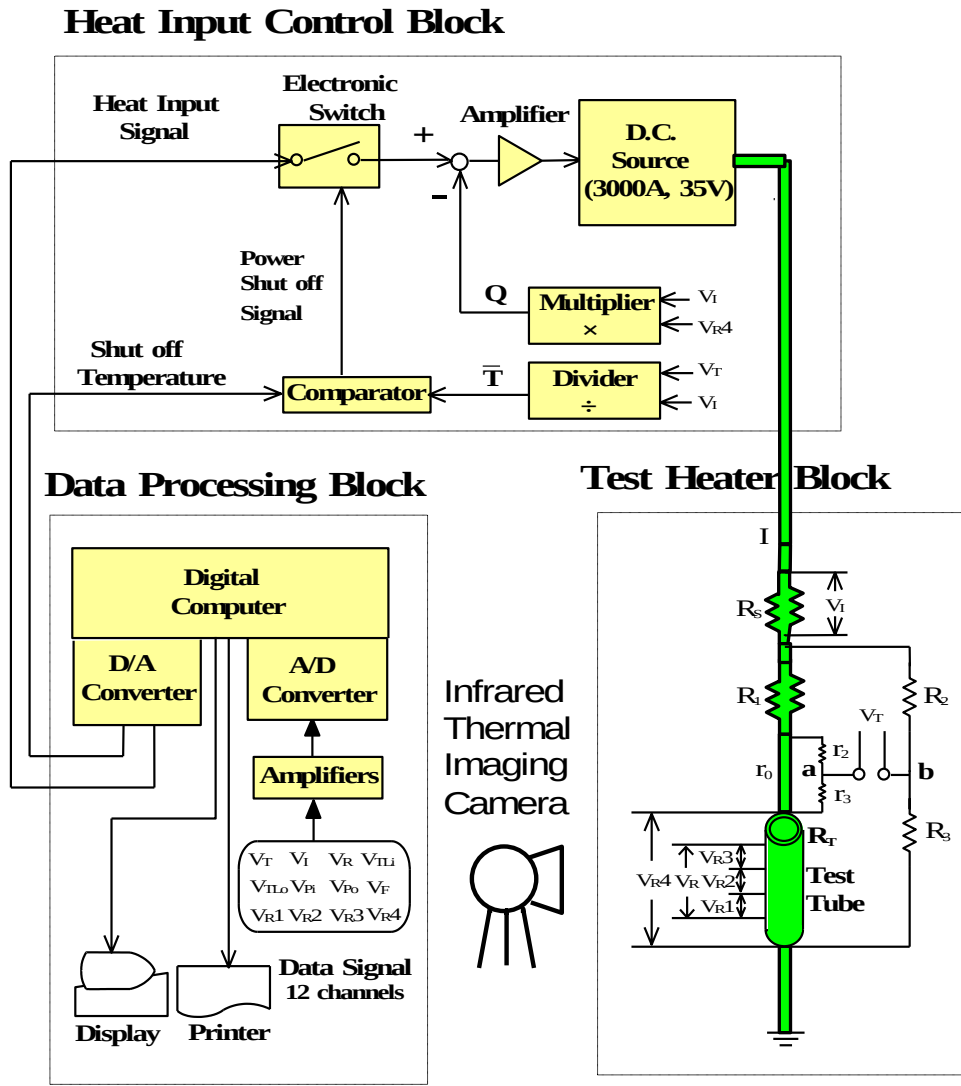


Fig. 5 Measurement and data processing system.

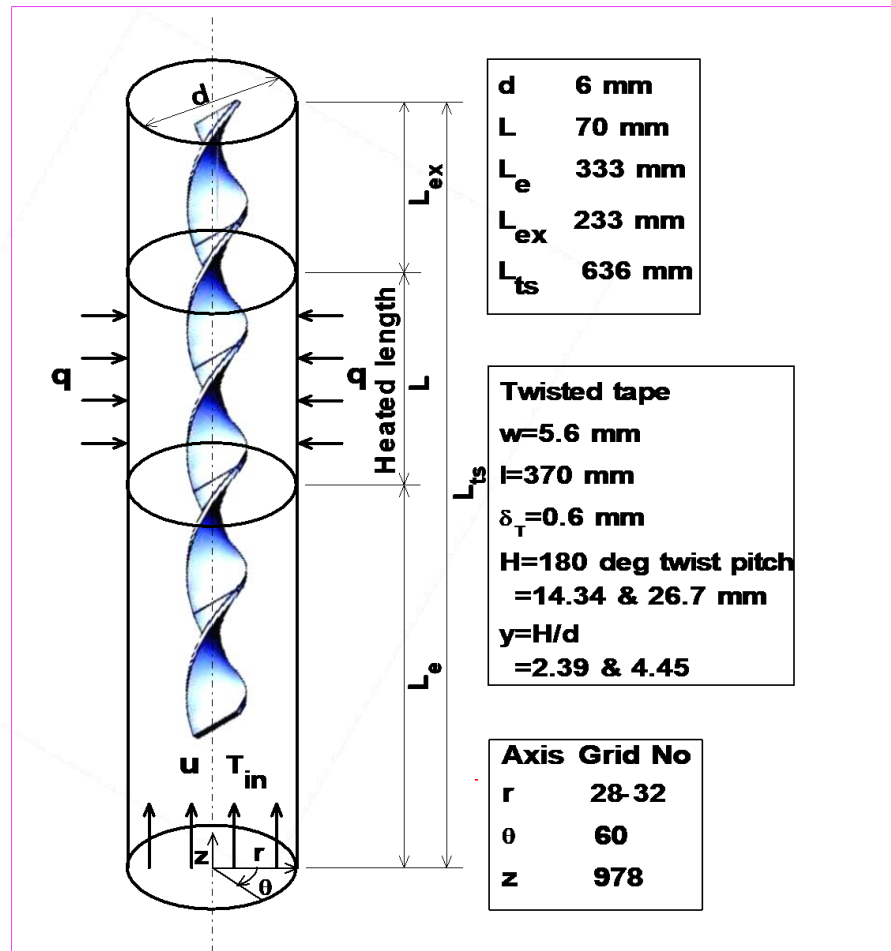
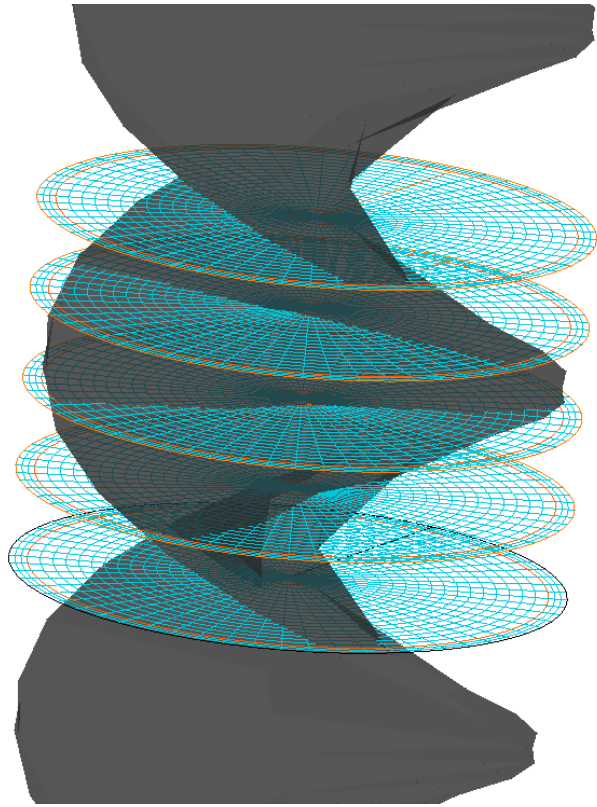
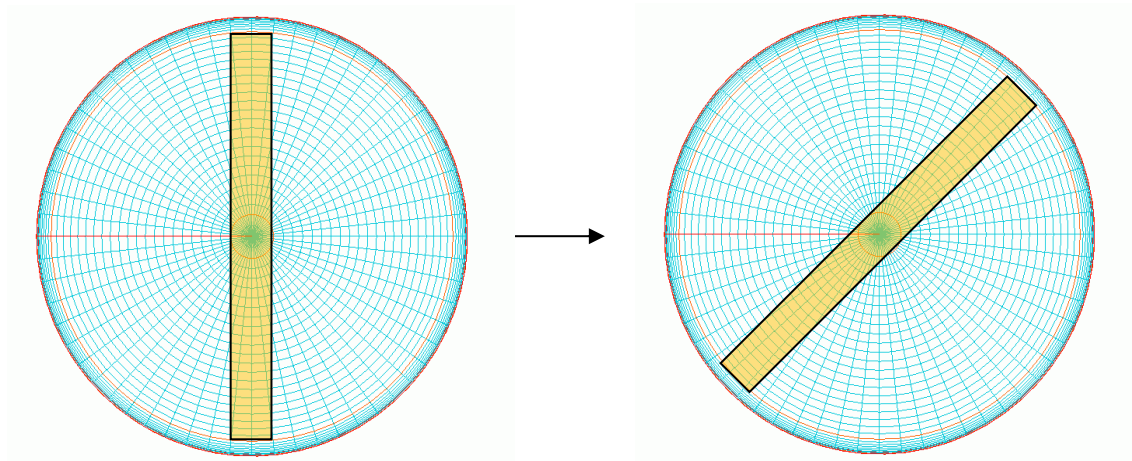


Fig. 6 Physical model for numerical analysis.



(a)



(b)

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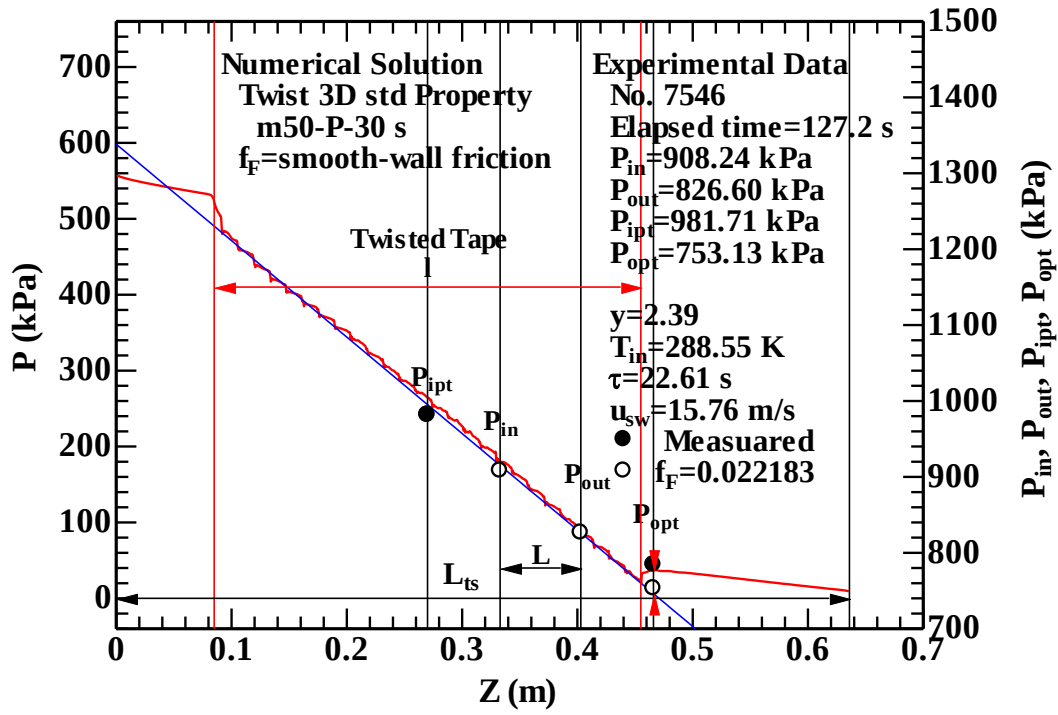
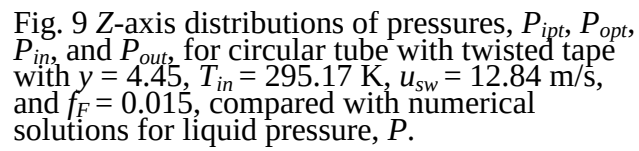


Fig. 8 Z-axis distributions of pressures, P_{ipt} , P_{opt} , P_{in} , and P_{out} , for circular tube with twisted tape with $y = 2.39$, $T_{in} = 288.35$ K, $u_{sw} = 15.76$ m/s, and $f_F = 0.022$, compared with numerical solutions for liquid pressure, P .



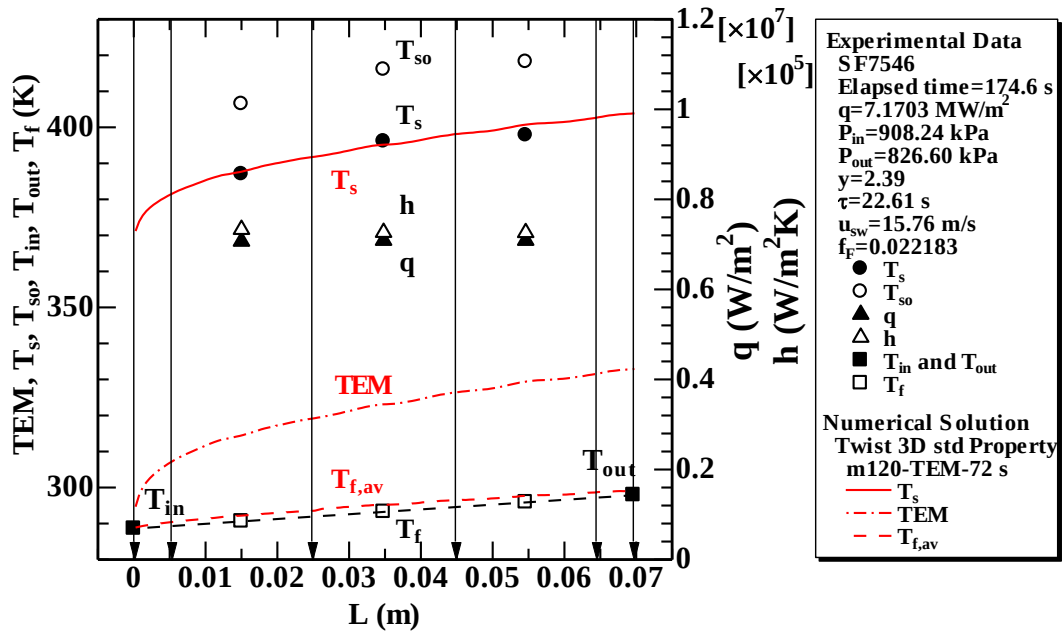


Fig. 10 T_s , T_{so} , q , and h for three sections and T_{in} and T_{out} for circular tube with twisted tape with $y = 2.39$ and $u_{sw} = 15.76 \text{ m/s}$, compared with numerical solutions for TEM , T_{in} , and T_{out} .

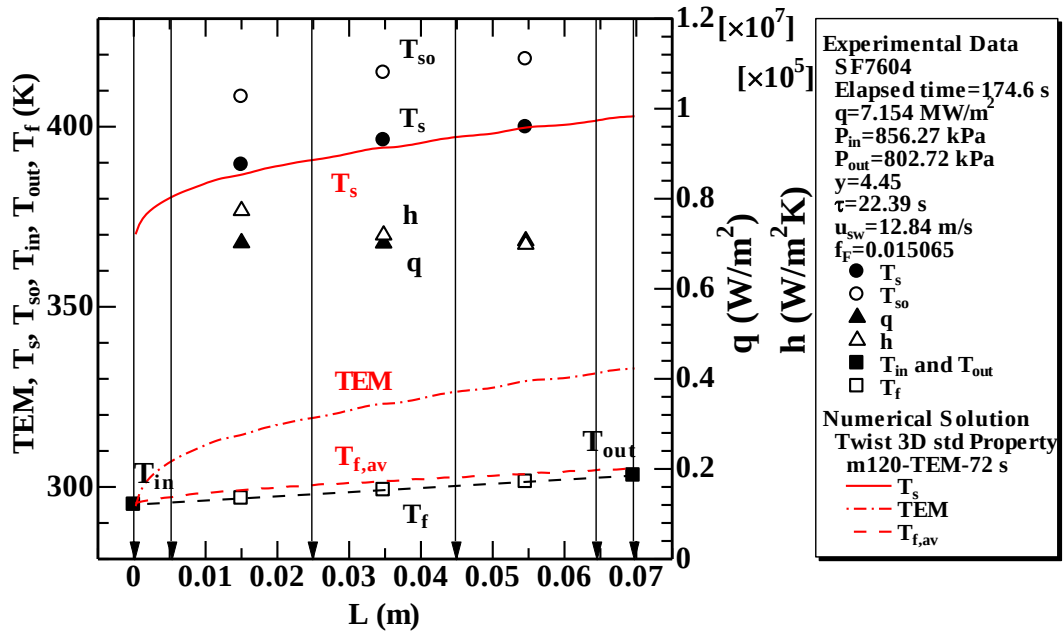


Fig. 11 T_s , T_{so} , q , and h for three sections and T_{in} and T_{out} for circular tube with twisted tape with $y = 4.45$ and $u_{sw} = 12.84 \text{ m/s}$, compared with numerical solutions for TEM , T_{in} , and T_{out} .

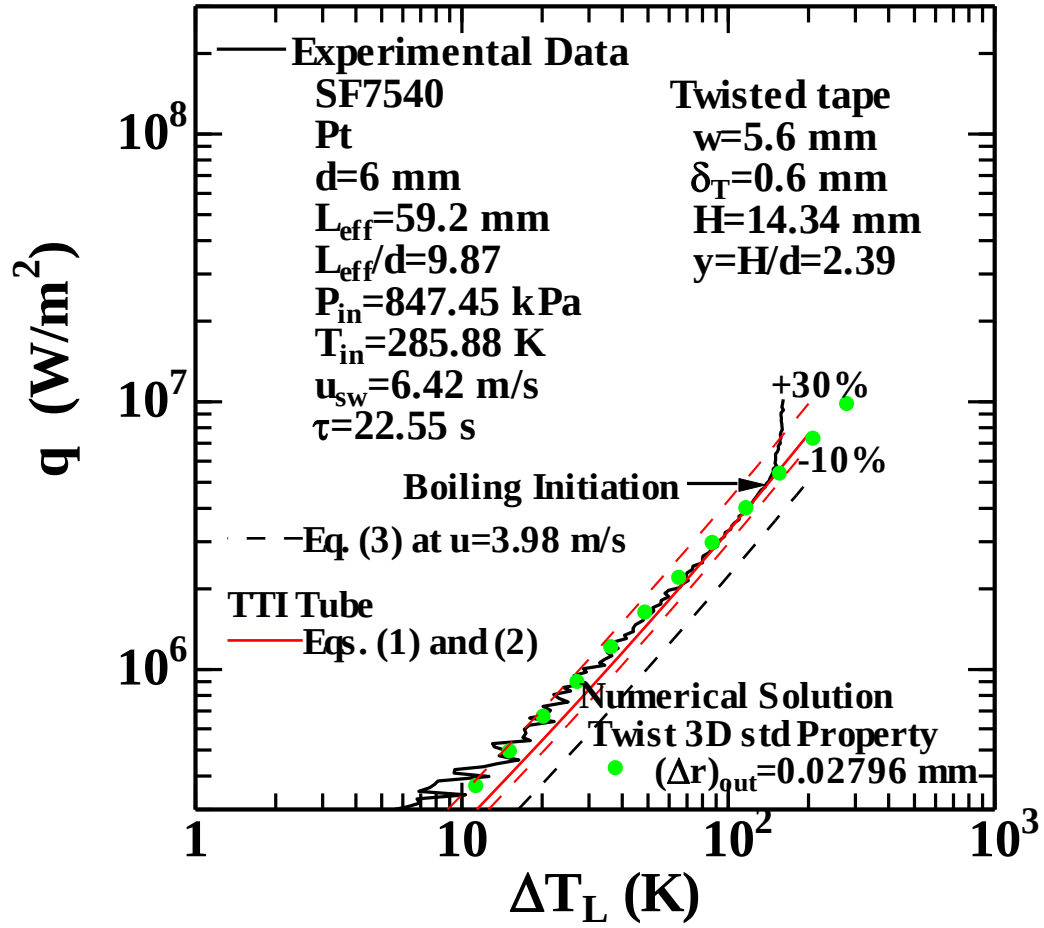


Fig. 12 Relationship between q and ΔT_L with circular tube of $d = 6$ mm and $L_{eff} = 59.2$ mm with twisted tape with $y = 2.39$, $u_{sw} = 6.42$ m/s, and $P_{in} = 847.45$ kPa.

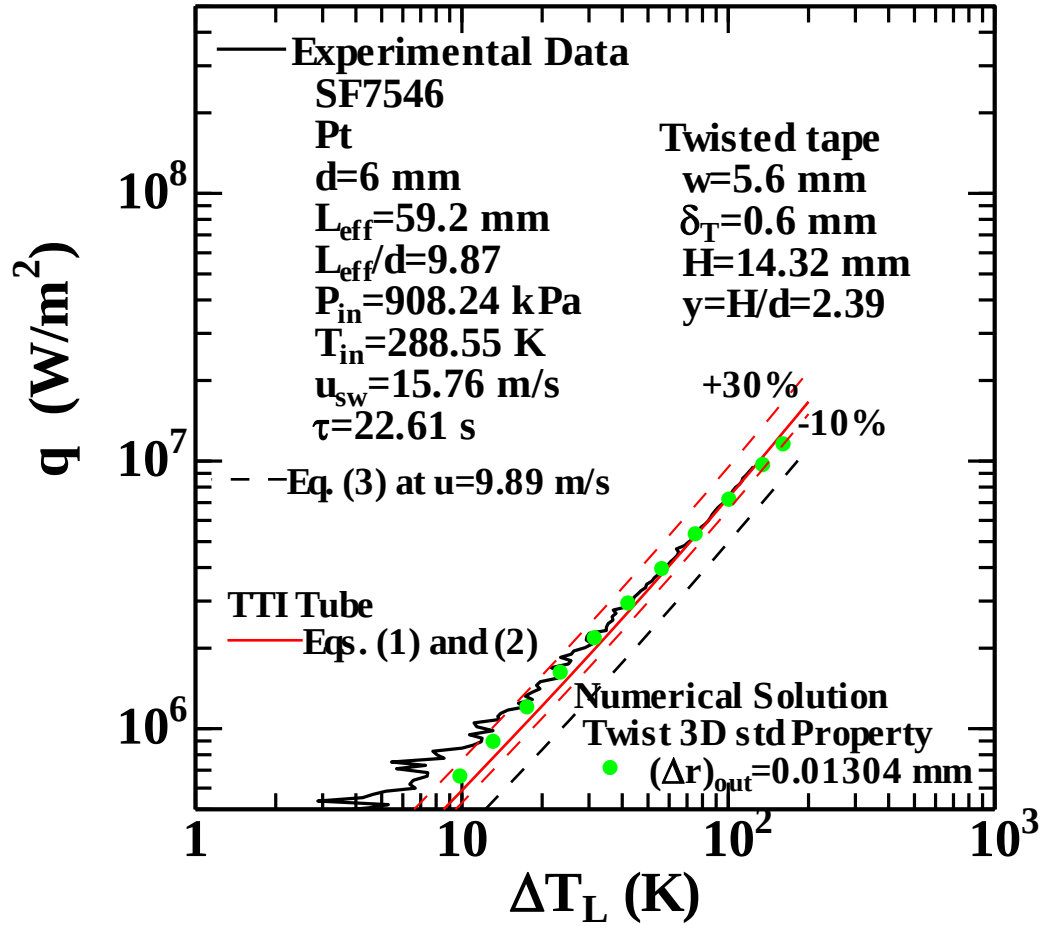


Fig. 13 Relationship between q and ΔT_L for circular tube with $d = 6$ mm and $L_{eff} = 59.2$ mm for twisted tape with $y = 2.39$, $u_{sw} = 15.76$ m/s, and $P_{in} = 908.24$ kPa.

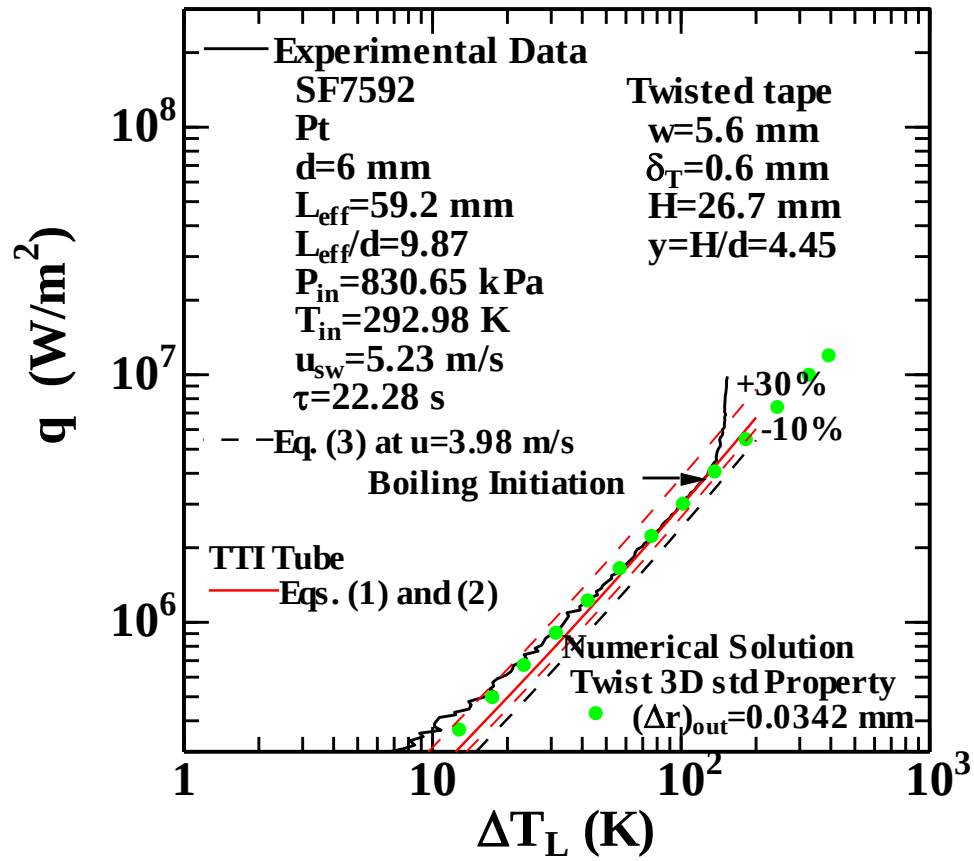


Fig. 14 Relationship between q and ΔT_L for circular tube with $d = 6$ mm and $L_{eff} = 59.2$ mm for twisted tape with $y = 4.45$, $u_{sw} = 5.23$ m/s, and $P_{in} = 830.65$ kPa.

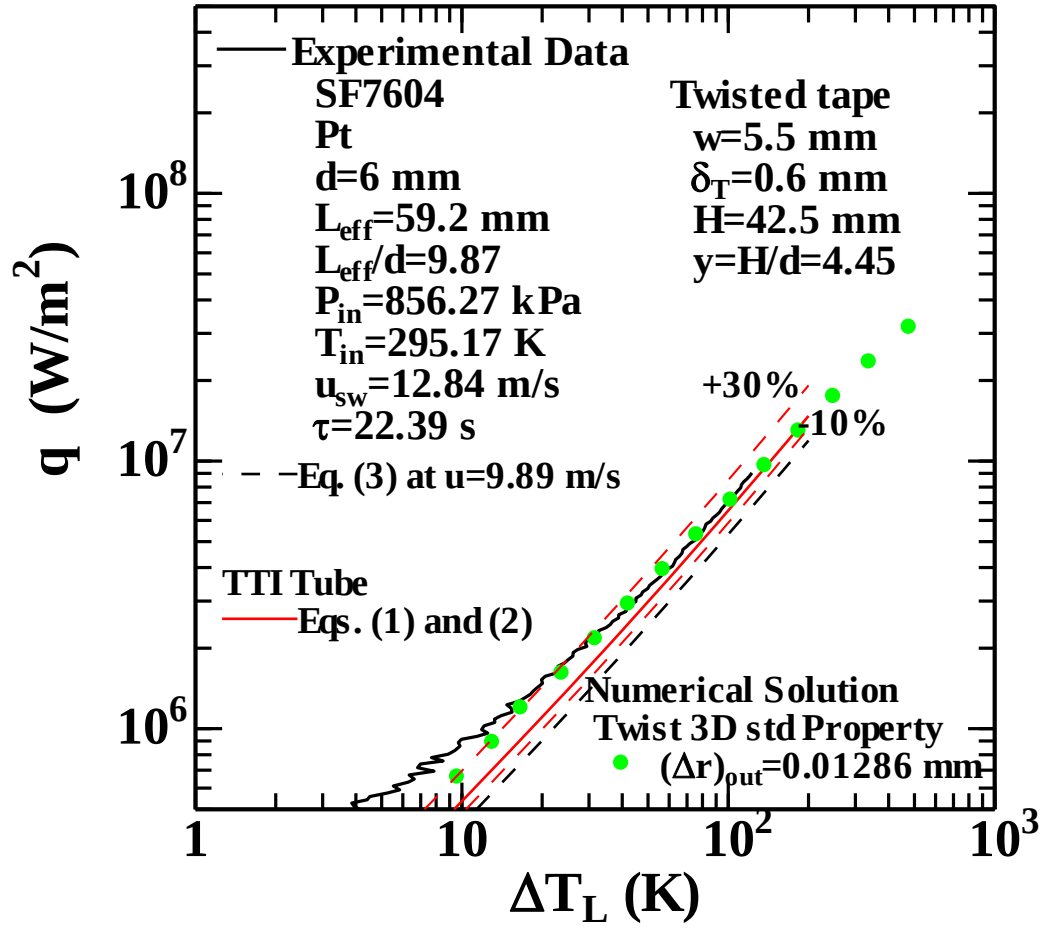


Fig. 15 Relationship between q and ΔT_L for circular tube with $d = 6$ mm and $L_{eff} = 59.2$ mm for twisted tape with $y = 4.45$, $u_{sw} = 12.84$ m/s, and $P_{in} = 856.27$ kPa.

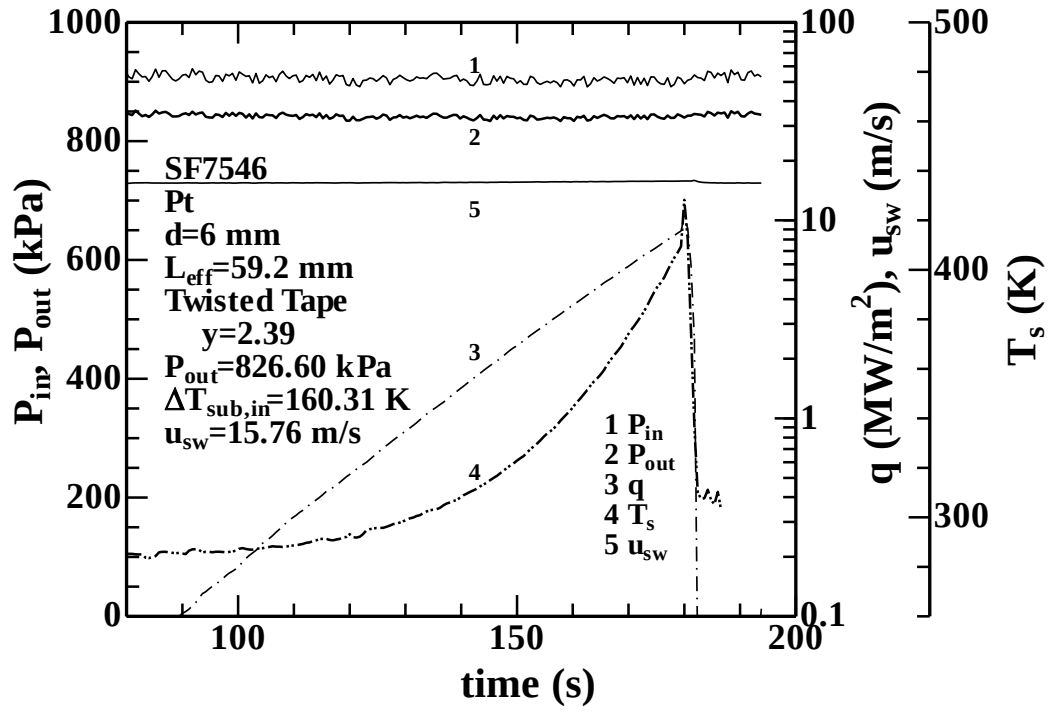


Fig. 16 Time variations in P_{in} , P_{out} , q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 2.39 at $P_{out}=826.60$ kPa, $\Delta T_{sub,in}=160.31$ K, $u_{sw}=15.76$ m/s and $\tau=22.61$ s.

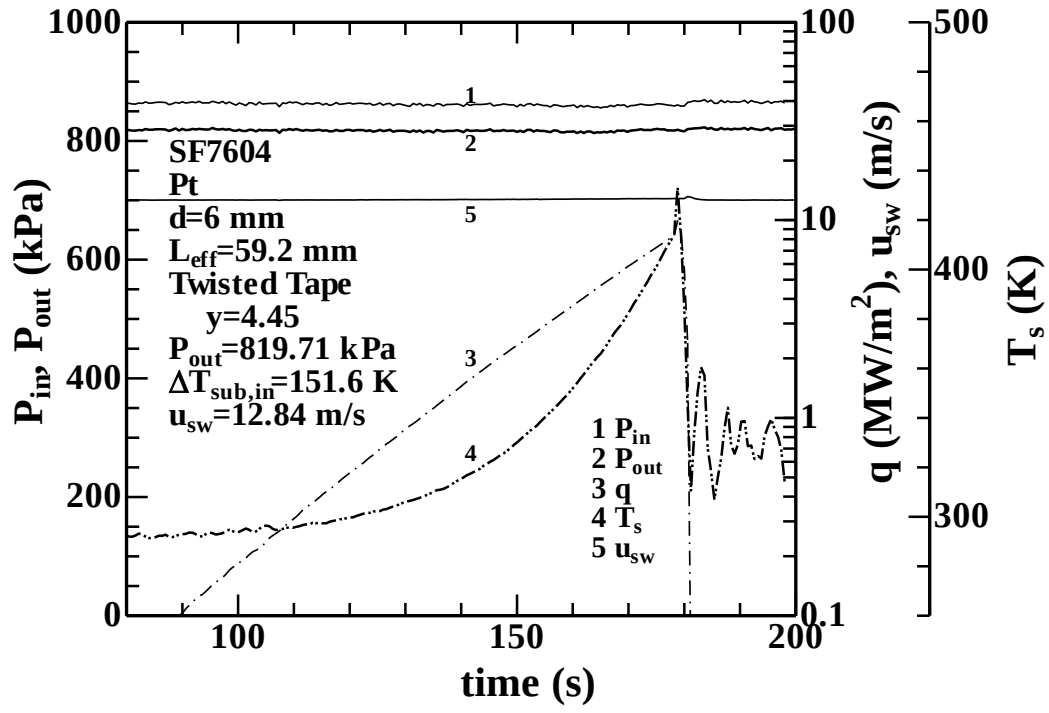


Fig. 17 Time variations in P_{in} , P_{out} , q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 4.45 at $P_{out}=819.71$ kPa, $\Delta T_{sub,in}=151.6$ K, $u_{sw}=12.84$ m/s and $\tau=22.39$ s.

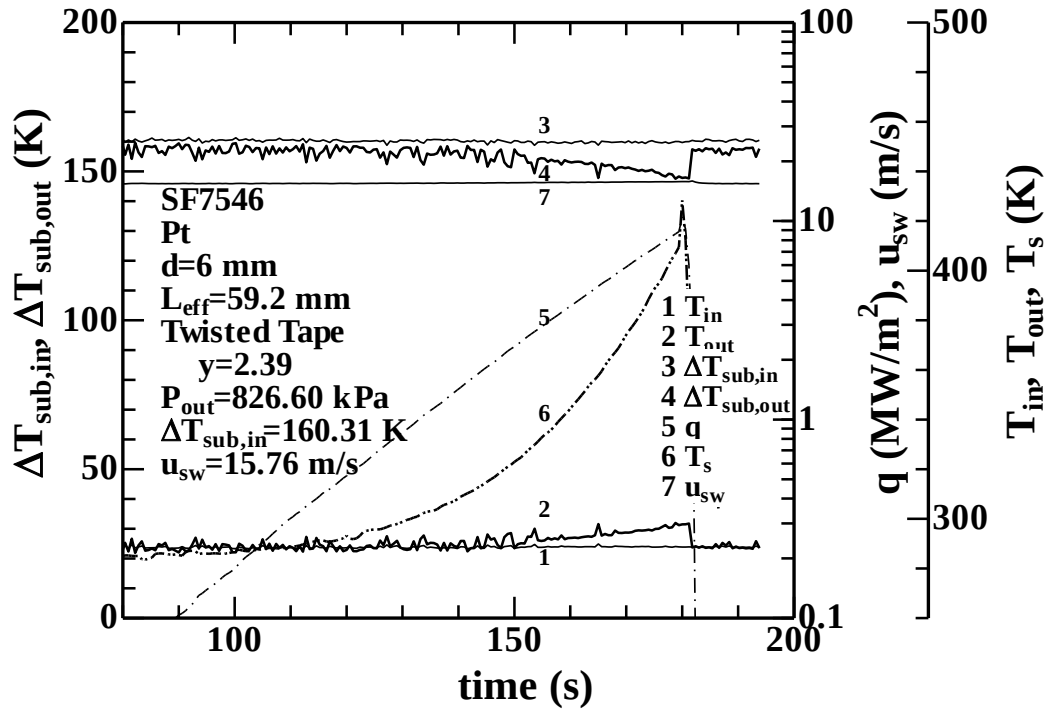


Fig. 18 Time variations in T_{in} , T_{out} , $\Delta T_{sub,in}$, $\Delta T_{sub,out}$, q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 2.39 at $P_{out}=826.60$ kPa, $\Delta T_{sub,in}=160.31$ K, $u_{sw}=15.76$ m/s and $\tau=22.61$ s.

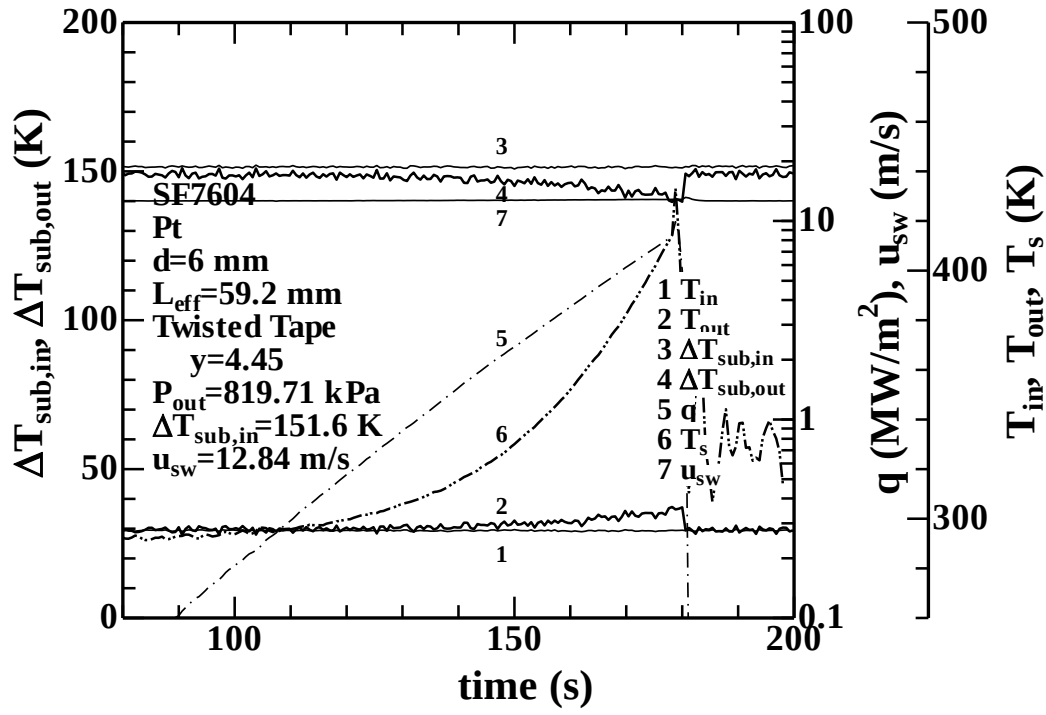


Fig. 19 Time variations in T_{in} , T_{out} , $\Delta T_{sub,in}$, $\Delta T_{sub,out}$, q and T_s for Pt test tube of $d=6$ mm and $L_{eff}=59.2$ mm with the twisted tape of the twist ratio, y , of 4.45 at $P_{out}=819.71$ kPa, $\Delta T_{sub,in}=151.6$ K, $u_{sw}=12.84$ m/s and $\tau=22.39$ s.

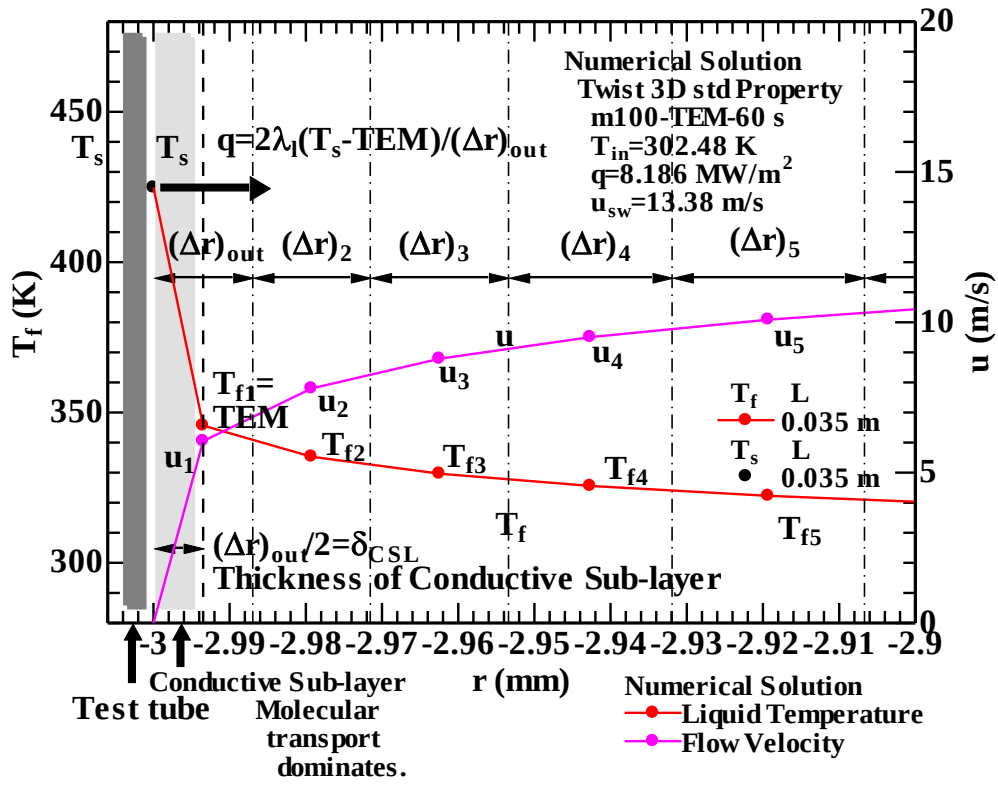


Fig. 20 Liquid temperatures in the conductive sub-layer for circular tube with $d = 6\text{ mm}$ and twisted tape insert.

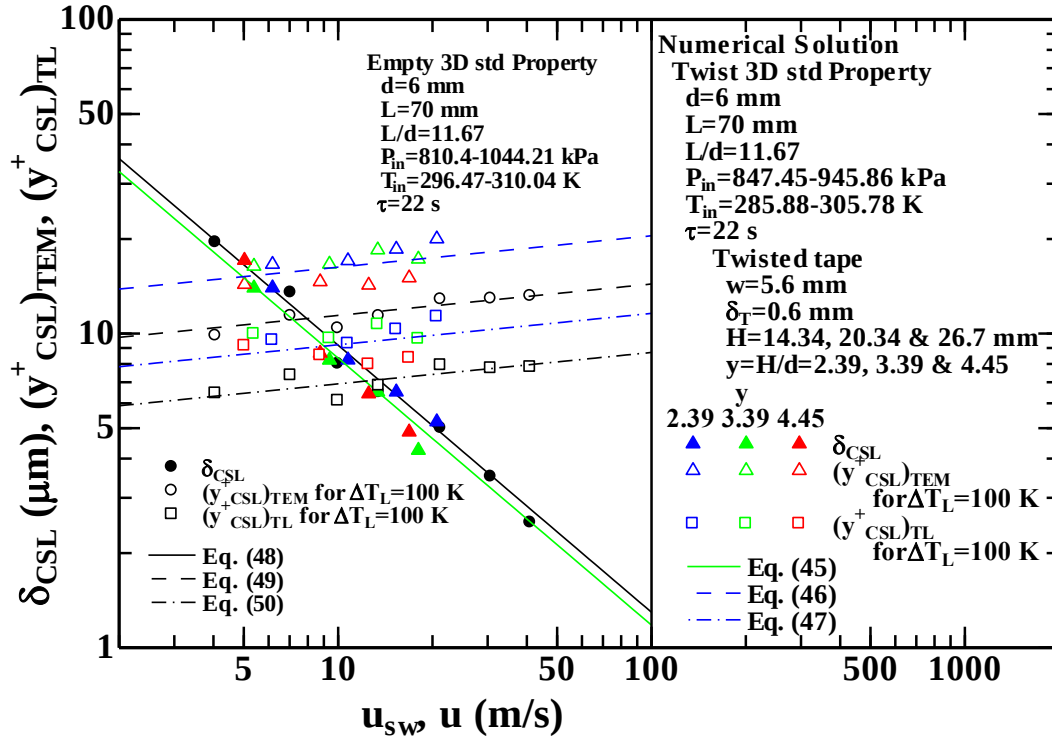


Fig. 21 Relationships between δ_{CSL} , $(y_{CSL})_{TEM}$, and $(y_{CSL})_{TL}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 and u_{sw} at $\Delta T_L = 100$ K.

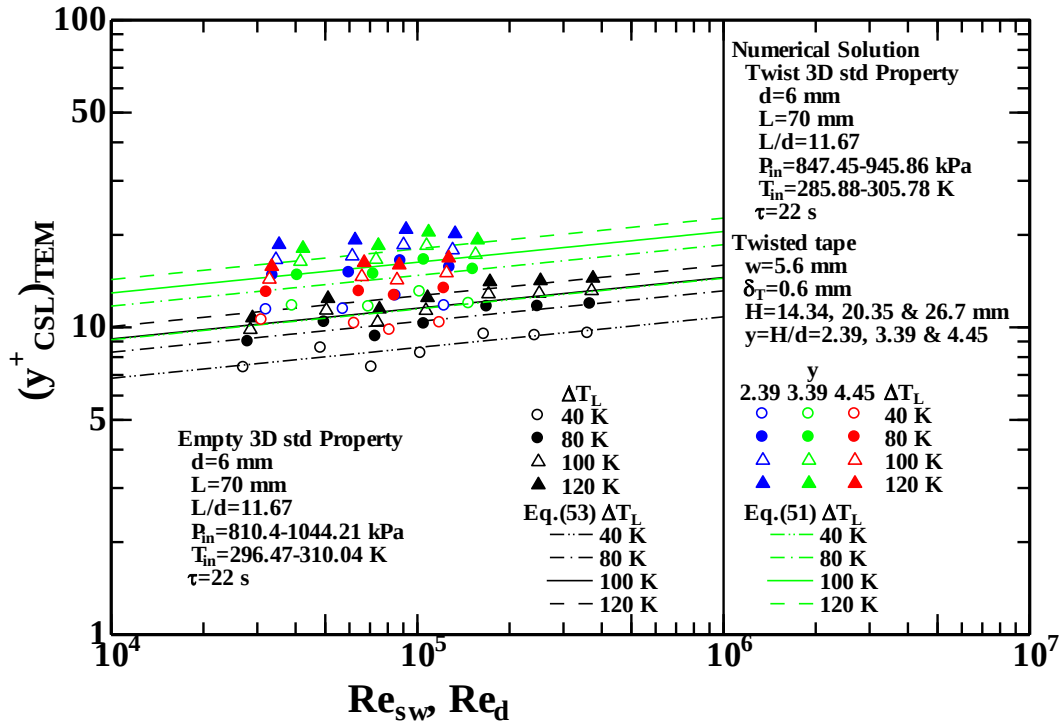


Fig. 22 Values of $(y^+_{CSL})_{TEM}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 and Re_{sw} at $\Delta T_L = 40, 80, 100$, and 120 K.

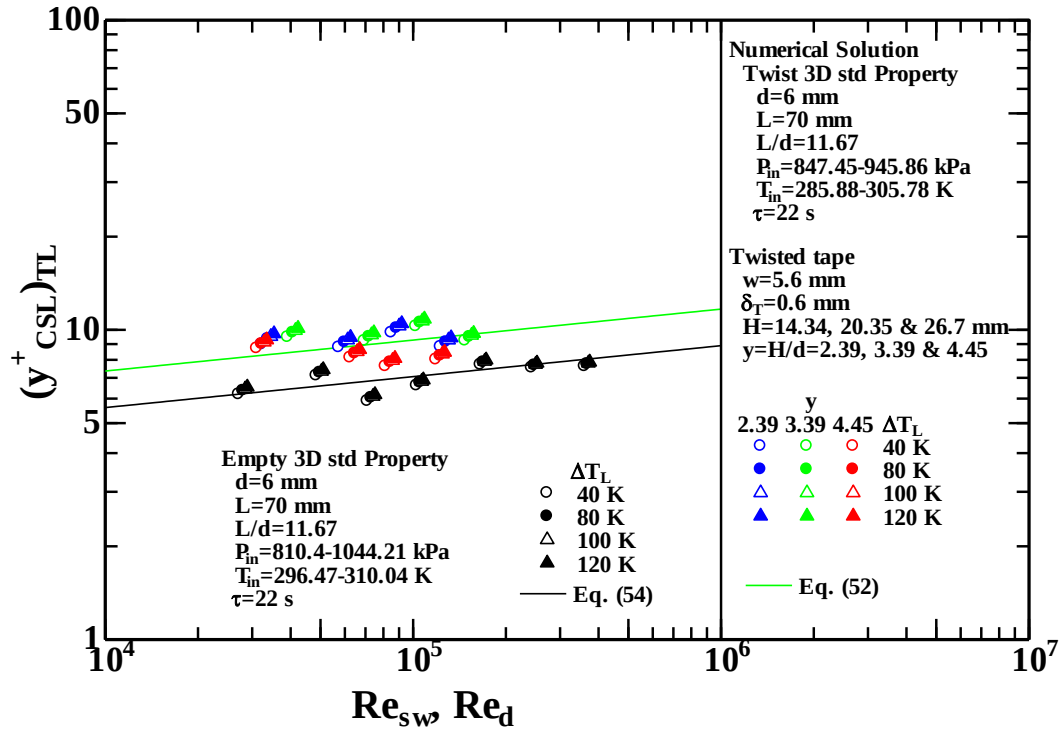


Fig. 23 Values of $(y^+_{CSL})_{TL}$ for circular tubes with twisted tapes with $y = 2.39, 3.39$, and 4.45 and Re_{sw} at $\Delta T_L = 40, 80, 100$, and 120 K.

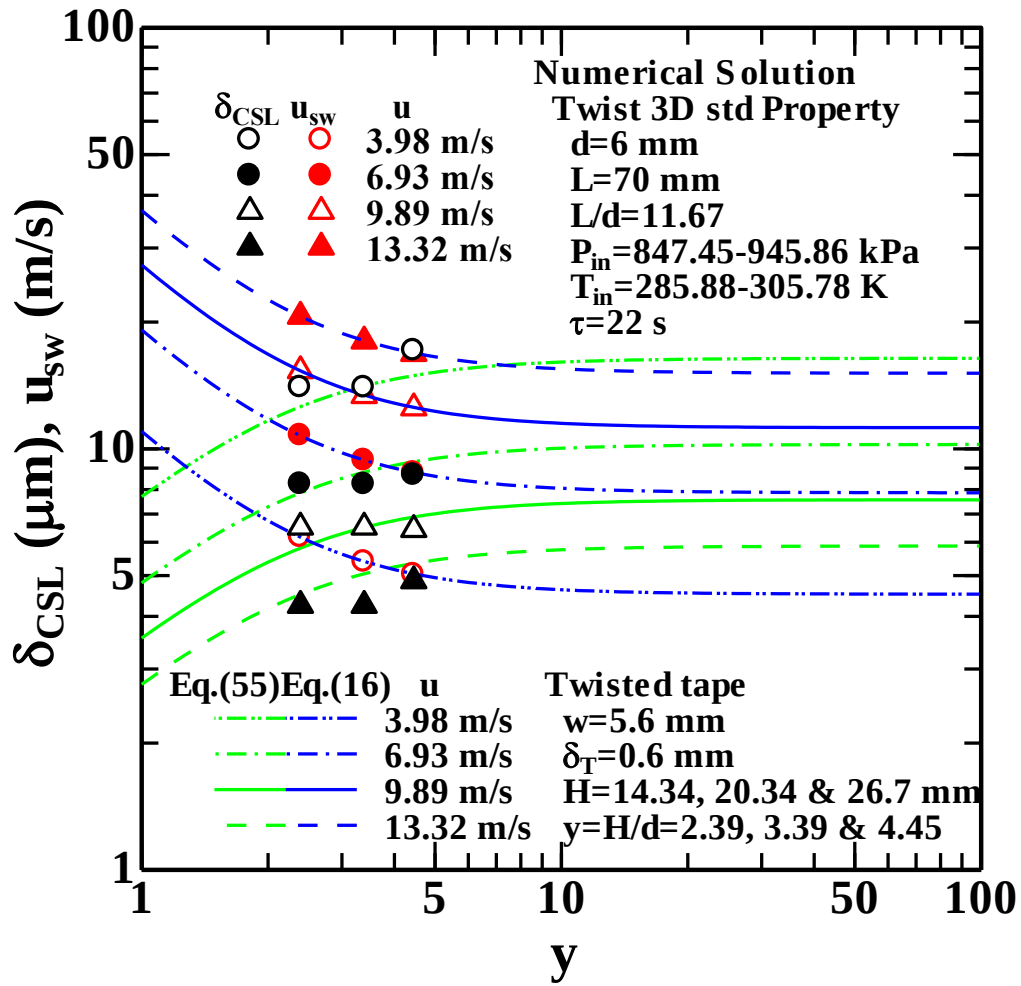


Fig. 24 Relationship between δ_{CSL} and u_{sw} for numerically solved turbulent heat transfer and y at $u = 3.98, 6.93, 9.89,$ and 13.32 m/s.

