HEAT TRANSFER IN THE LAMINAR AND TRANSITIONAL FLOW REGIMES OF SMOOTH VERTICAL TUBE FOR UPFLOW DIRECTION

Bashir A.I. and Meyer J.P.*
*Author for correspondence
Department of Mechanical and Aeronautical Engineering,
University of Pretoria,
Pretoria, 0002,
South Africa.
E-mail : josua.meyer@up.ac.za

ABSTRACT

Due to design limitations or system upgrade, heat exchangers are required to operate in the laminar-to-turbulent transition region in order to achieve a high heat transfer with low pressure drop. The present research investigates the effect of vertical upflow with heating on the single-phase heat transfer in the transitional flow regime of a vertical tube. The experimental setup consists of a swinging test bench which allows for horizontal flow and vertical upflow direction under constant heat flux boundary condition. A smooth circular tube with inner diameter of 5.1 mm and a heated length of 4.52 m was used as the test section with water at Prandtl number ranging between 5 to 7 as working fluid. The experiment covers Reynolds number range of 1 000 to 10 000 at horizontal and vertical orientations of the test section using a squared-edged inlet geometry. For fully developed vertical upflow direction, transition is delayed when compared to horizontal flow direction where the effect of secondary flow increases the laminar flow heat transfer and causes transition to occur much earlier. The width of transition region for vertical tubes is significantly smaller than that of the horizontal tubes.

NOMENCLATURE

c_p	[J/kg K]	Specific heat at constant pressure
D	[m]	Diameter
Gr	[-]	Grashof number
h	[W/m^2 K]	Heat transfer coefficient
j	[-]	Colburn *j*-factor
k	[W/m K]	Thermal conductivity
L	[m]	Length
$\dot{m}$	[kg/s]	Mass flow rate
n	[-]	Constant (number of stations) Eq. (6)
Nu	[-]	Nusselt number
Pr	[-]	Prandtl number
Q	[W]	Heat transfer rate
$\dot{q}$	[W/m^2]	Heat flux
R	[°C/m]	Thermal resistance
Re	[-]	Reynolds number
T	[°C]	Temperature
x	[m]	Distance from inlet

Special characters

μ	[kg/m s]	Dynamic viscosity

Subscripts

b	Bulk
cr	Critical Reynolds number
Cu	Copper
e	Exit
f	Fluid
i	Inlet/ inner
is	Inner wall surface
m	Mean
os	Outer wall surface
s	Surface
w	Wall
o	Outer

INTRODUCTION

Vertical tubes are used for many industrial applications ranging from cooling to thermal systems such as in solar energy collectors, compact heat exchangers, boilers and nuclear reactors. They are mostly used in heat exchangers because of their buoyancy assisting and/or opposing flows in the upflow and downflow directions. Most heat exchangers operate in either a laminar or turbulent flow regime in order to achieve a balance between the heat transfer and the pressure drop. Heat exchangers operating in the turbulent region have high heat transfer and high pressure drop while for those operating in the laminar region, both the heat transfer and the pressure drop are low. The target is to obtain a high heat transfer with minimum pressure drop, thus low pumping power. These can be achieved through a good understanding and selection of an appropriate operating flow regime, configuration and orientation of the heat exchanger. Reynolds [1] performed one of the early researches on fluid flows within tubes in order to differentiate laminar and turbulent flow regimes. This leads to the introduction of a laminar-to-turbulent transition region.

Transitional flow regime is of paramount importance in convective heat transfer, be it free, forced or mixed convection inside different flow passages due to a good compromise between the heat transfer and pressure drop. Although, most engineering textbooks discourage designing and operating a heat exchanger in the transition region, the need to fully understand its behavior becomes necessary as the trend in using transition region increases.

Since the early 1990's to date, Ghajar and co-workers [2] at Oklahoma State University and Meyer and co-workers [3] at University of Pretoria have been working, on the experimental analysis of heat transfer and pressure drop in the transitional flow regime of smooth, enhanced and micro tubes using water, nanofluids and glycol mixtures as working fluids. They investigated the effect of heating, configuration and inlet geometries in the transitional flow regime under uniform heat flux as well as uniform wall temperature boundary conditions. However, all their transitional flow regime works were conducted within horizontal channels with no vertical tubes examined as reviewed by Meyer [4].

The behaviour of laminar heat transfer in vertical tubes differs from that of horizontal tubes. The heat transfer rate in vertical tubes is a strong function of either the upflow or downflow directions [5]. For assisting flow (upflow with heating or downflow with cooling), natural convection assists forced convection and enhances the heat transfer, while for opposing flow (upflow with cooling or downflow with heating), natural convection resists forced convection and impairs the heat transfer [6]. Early research on laminar and turbulent mixed convection heat transfer in vertical tubes was conducted by Eckert and Diaguila [7]. Petukhov *et al.* [8] reviewed different experimental and theoretical studies on combined free and forced (mixed) convection in vertical tubes. Metais and Eckert [9] developed a flow regimes maps to separate the forced, mixed and free convection heat transfer for laminar, transition and turbulent flow regimes within vertical tubes under constant wall temperature and constant heat flux boundary conditions. Jackson *et al.* [10] compared different theoretical and experimental works by other researchers on mixed convection heat transfer for laminar and turbulent flows within vertical tubes, without a transition region.

Mohammed [11] as well as Mohammed and Salman [12] studied the effect of flow direction and tube inclination on the surface temperatures of laminar mixed convection heat transfer in vertical tubes. A decrease in laminar heat transfer for vertical tubes with upflow observed when compared to horizontal tubes. Wang *et al.* [13, 14] showed that transition was delayed with decreasing Prandtl number and increasing temperature difference between the inlet and exit temperatures in a vertical heated rectangular channel. Behzadmehr *et al.*[15] performed an experimental analysis on the onset of laminar-turbulent transition mixed convection in vertical tube using air as working fluid at three different Reynolds numbers 1 000, 1 300 and 1 600. They also studied low Reynolds numbers mixed convection in laminar and turbulent regions in vertical tubes under uniform heat flux boundary condition [16]. However, their transition work concentrated more on the onset of transition only, not the entire transition region between laminar and turbulent regions. To the best of our knowledge there is insufficient information available in the literature on the behavior of heat transfer in the transitional flow regime of vertical tubes.

This research aims to investigate the effect of vertical upflow direction on single-phase heat transfer in the laminar and transition regions of smooth vertical tubes under constant heat flux condition.

EXPERIMENTAL SETUP

The overall experimental setup consists of a water flow loop, calming section, and the experimental test section placed on a rotating test bench. The water flow loop circulates from a 500 ℓ storage tank using a gear pump through the flow meters to the calming section and experimental test section. After the experimental test section, the flow then returns back to the storage tank where it is cooled down by a chiller unit.

Figure **1** shows the overall experimental setup. Water was used as working fluid throughout the experiment and was maintained at a temperature of 20^{0}C inside a storage tank using a chiller unit attached to it. An Ismatec® BVP-Z gear pump with a maximum flow rate of 420 ℓ/h was connected to the storage tank using flexible horses in order to avoid transmitting vibration from the pump to the test section. The pump was used to circulate the water in the experimental system through a filter. The pump was automatically controlled from an in-house LabVIEW programme on a Personal Computer to set the required mass flow rate. Next to the gear pump is a bypass valve used to allow the flow back to the storage tank thereby increasing the back pressure on the pump with increase in pump speed. Two Micro Motion Coriolis flow meters with accuracy of 0.05% were used afterwards to measure the mass flow rate of the water from the pump to the experimental test section. One Coriolis flow meter was used at a time depending on the mass flow rate requirements.

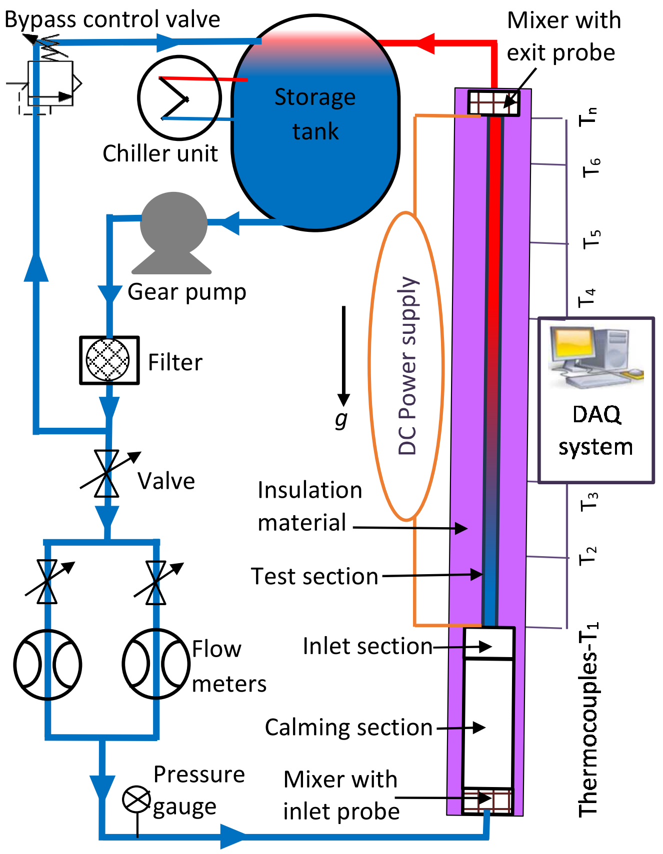

Figure 1 Schematic layout of the overall experimental facility.

Before the water enters test section through the calming section, an inlet mixer attached to the entrance of the calming section was used in order to achieve uniform bulk temperature at the inlet of the test section.

After the mixer, the water passed through a calming section connected to the inlet section of the experimental test section to ensure a uniform velocity distribution in the test section. The calming section design used was similar to that of Ghajar and Tam [2] with inlet contraction ratio of 33.5. A square-edged inlet geometry was used in this analysis where sudden contraction is achieved at the test section inlets. After the test section, another exit mixer was used and the flow then returns back to the storage tank. Bakker *et al.*[17] static mixer design was used to ensure proper mixing of the water before measuring the inlet and exit fluid bulk temperatures.

The experimental test section consists of a smooth hard drawn copper tube of inner diameter 5.1 mm and an outer diameter of 6.3 mm. The total length of the test tube with calming section was approximately 5.6 m with a heated test section length of about 4.52 m and maximum length to diameter ratio (*x/D*) of 886. A theoretical fully developed length of 1 m can be achieved with this length at a Reynolds number of 2 300 and a Prandtl number of 6. This length was dedicated for the fully developed flow analysis in the test section. Armaflex® insulation material of thickness 80 mm and thermal conductivity of 0.034 W/m K was used to insulate the test section from the environment. Using a simple one-dimensional heat transfer calculation, the maximum relative heat loss was calculated to be 1.4%.

Twenty-one thermocouple stations were designated at closer intervals near the entrance and exit (fully developed region), and at wider intervals downstream of the tube. For each station, three thermocouples were used with one thermocouple placed at the top and bottom of the tube and another at 90^0 position for station 1, 3, 5, etc., and another at 270^0 position for station 2, 4, 6, etc. Two Pt-100 temperature probes with accuracy of 0.06% were inserted in the inlet and exit mixers each in order to measure the average inlet and exit fluid bulk temperatures respectively. For a constant heat flux boundary condition experiment, the heat flux was maintained over the heated length of the test section using a DC power supply though the heating wires. Two T-type constantan heating wires of diameters 0.38 mm were connected in parallel to the DC power supply and coiled to the test section skipping the thermocouple junctions.

The test section was placed on a 6 m long Tectra® aluminium profile bench pivoted at the centre and supported at both ends so that it can swing around an angle of 90^0 in the upflow direction. This test bench was then placed on a rigid test bench structure of 3 m height made from Tectra® aluminium profile designed to prevent any wobbling of the test section. Damping pads were used between the beam and the test bench to prevent transmitting vibration to the test section from the equipment and floor.

The data-capturing was undertaken using a National Instruments® Data Acquisition (NI-DAQ) system. A Labview program was designed to integrate all the DAQ system hardware used and a separate program was used for the analysis. The experiment runs in horizontal and vertical upflow directions of the complete test section for validation and comparison purposes.

DATA REDUCTION

All fluid properties were evaluated at the fluid bulk temperature. The bulk temperature, T_b was calculated from the measured fluid inlet and exit temperatures as:

$$T_b = \frac{(T_i + T_e)}{2} \tag{1}$$

For the fully developed region under consideration, the following equation was used to estimate the fluid bulk/local mean temperatures;

$$T_m = T_i + \frac{(T_e - T_i)x}{L} \tag{2}$$

where x is the distance from the tube inlet.

The average Reynolds number was determined from the measured mass flow rate $\dot{m}$, inner tube diameter D_i, and the fluid viscosity, μ as:

$$Re = \frac{4\,\dot{m}}{\pi\, D_i\, \mu} \tag{3}$$

The heat flux applied to the fluid, $\dot{q}_f$, was estimated using the heat transfer rate, $\dot{Q}_f$, defined as $\dot{Q}_f = \dot{m}\, c_p\, (T_e - T_i)$ and the tube inner surface area;

$$\dot{q}_f = \frac{\dot{Q}_f}{\pi\, D_i\, L} \tag{4}$$

The resistance through the wall, R_w was determined from:

$$R_w = \frac{ln(D_o/D_i)}{2\,\pi\, k_{Cu}\, L} \tag{5}$$

where k_{Cu} is the thermal conductivity of the copper tube. The average outer wall surface temperature, T_{os} for all the thermocouple stations considered in the fully developed region was determined using the trapezoidal rule, with T_{os_n} been the average temperature of the thermocouples per station.

$$T_{os} = \frac{1}{2(n-1)}\left(T_{os_1} + 2T_{os_2} + 2T_{os_3} + \cdots + 2T_{os_{n-1}} + T_{os_n}\right) \tag{6}$$

The inner wall surface temperatures, T_{is} was calculated using the thermocouple measurements on the outer wall surface temperatures, T_{os}, heat transferred to the fluid and the wall resistance as:

$$T_{is} = T_{os} - \dot{Q}_f\, R_w \tag{7}$$

Since the thermal conductivity of copper is very high, the wall resistance, R_w was found to be negligible and hence, assumed that $T_{is} \approx T_{os}$ from Eq. (7).

The average heat transfer coefficient, h, was determined from the heat flux, $\dot{q}_f$, inner wall surface temperature, T_{is} and bulk fluid temperature, T_b as:

$$h = \frac{\dot{q}_f}{T_{is} - T_b} \tag{8}$$

The average and local Nusselt numbers were determined based on the heat transfer coefficient and thermal conductivity of the fluid as:

$$Nu = \frac{h\ D}{k} \tag{9}$$

$$Nu(x) = \frac{h(x)\ D}{k(x)} \tag{10}$$

The average Colburn j-factor used to compare the relationship between the Reynolds number and heat transfer coefficient while taking into account the variation of the fluid Prandtl number was calculated from:

$$j = \frac{Nu}{Re\ \ Pr^{\frac{1}{3}}} \tag{11}$$

UNCERTAINTY ANAYLSIS

The uncertainty analysis of the test section was performed using the procedure proposed by Moffat [18] and Dunn [19]. All uncertainties were estimated within a 95% confidence interval. All the instruments used have manufacturer-specified accuracy as fixed error and two times the standard deviation of 400 samples of data captured as random error. The uncertainties of the inlet and exit Pt-100 thermal probes were calculated within 0.034% while that of the thermocouple was 0.1%. The Reynolds number uncertainty was approximately constant at 1.1% in the turbulent region and a maximum uncertainty of 1.6% in the laminar region for the lowest Reynolds number. The minimum and maximum uncertainty of the Nusselt number and Colburn j-factor was found to be 3.3% in laminar region and 10% in the turbulent region respectively. Due to fluctuation of the temperature measurements inside the test section within transition region, a maximum Nusselt number and Colburn j-factor uncertainty of 16.2% was obtained.

VALIDATION

The heat transfer results were validated in the laminar and turbulent flow regimes at horizontal orientation of the smooth circular tube. The validation uses local and average Nusselt numbers in the fully developed region.

The laminar experimental data were first validated using the forced convection experiment where the theoretical forced convection Nusselt number is 4.36 and then mixed convection experiments. Forced convection heat transfer experiment was performed using low heat flux of about 277 W/m^2, Reynolds number of 664 and a Prandtl number of 5.79. This condition is within the laminar forced convection region based on the Ghajar and Tam [20] flow regime map. The result of local Nusselt numbers as a function of axial location is shown in Figure **2**, where forced convection heat transfer was achieved in the fully developed region with average Nusselt number of 4.39.

Figure **2** shows the entrance and fully developed flow Nusselt numbers where the Nusselt numbers decreases along tube length showing that the flow is developing from tube inlet up to *x/D* of 298. From *x/D* of 357, the Nusselt numbers become relatively constant and this shows that the flow is thermally fully developed.

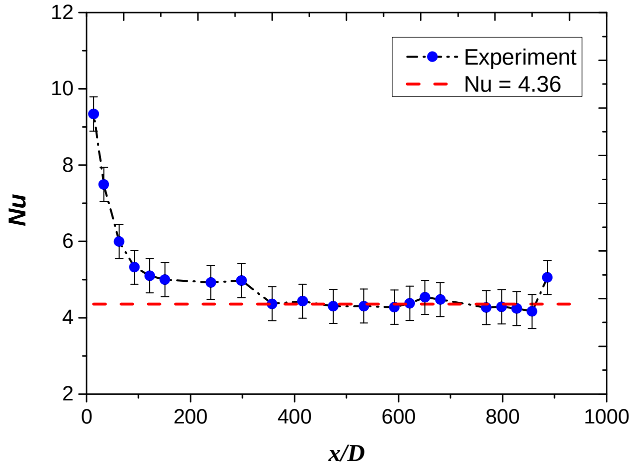

Figure 2 Local Nusselt numbers as a function of axial location for forced convection condition at Reynolds number of 664 and a heat flux of 277 W/m^2.

The fully developed local Nusselt numbers were compared with forced convection Nusselt number of 4.36 and shows good agreement with an average deviation of 2.35% and a maximum deviation of 4.5%. The last thermocouple station at *x/D* = 886 shows an increase in heat transfer caused by the upstream effects from the exit mixer and is excluded in the analysis because the deviation is higher than the uncertainty as shown in Figure **2**.

The laminar mixed convection validation experiment was performed by increasing the heat flux from 277 W/m^2 where forced convection is achieved to a heat flux of 6 kW/m^2 between Reynolds number of 1 000 to 3 000. As the heat flux increases, the laminar heat transfer increases. The fully developed local Nusselt numbers at a heat flux of 6 kW/m^2 deviated with an average of 70% from the forced convection Nusselt number of 4.36. This shows that secondary flow was developed within the flow and enhances the laminar heat transfer which indicates mixed convection heat transfer. The average laminar Nusselt number results in the fully developed flow were compared with fully developed Morcos and Bergles [21] correlation for constant heat flux condition. The data show excellent agreement with Morcos and Bergles [21] correlation with an average deviation of 1.6% and a maximum deviation of 2.6%.

The fully developed turbulent flow heat transfer results were compared with the correlations of Gnielinski [22] and Ghajar and Tam [2] at a heat flux of 6 kW/m^2 between Reynolds numbers of 3 5000 and 10 000. The data show that there is an average deviation of 11.2% and a maximum deviation of 20.6% from Gnielinski [22] correlation. However, with Ghajar and Tam [2] correlation, the data under-predict their correlation with an average deviation of 13.4% and a maximum deviation of 16.9%. This is due to the fact that the Ghajar and Tam [2] correlation is for higher Prandtl numbers. Hence, the maximum deviation of the turbulent experimental data from all the correlations were within 20.6%. Therefore, the laminar and turbulent results compare very well with the literature and these give confidence that the system and procedure used for the measurement and analysis of the heat transfer in transition region are accurate and validated.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Figure **3** shows the results of fully developed heat transfer in terms of average Nusselt number and Colburn *j*-factor as a function of Reynolds number in the horizontal and vertical orientations of the test section under the same flow conditions. The experiment runs from Reynolds number of 1 000 to 10 000, first with the test section in horizontal orientation at a heat flux of 6 kW/m^2 and then rotates the test section to vertical upflow orientation for comparison purpose.

In the laminar region, the horizontal flow heat transfer in Figure **3(a)** was dominated by mixed convection heat transfer as the Nusselt number is much higher than the predicted forced convection Nusselt number of 4.36 for constant heat flux condition. It indicates that the effect of secondary flow increases the laminar heat transfer in the horizontal tube and the Nusselt number was in the range of 7.41 ~ 7.57. For the vertical upflow direction, the laminar Nusselt number is much closer to forced convection Nusselt number with an average deviation of 13%. This shows that the flow was dominated by forced convection heat transfer and the effect of secondary flow is negligible. It also indicates that the buoyant motion assisting the flow for vertical upflow with heating (in Figure **3(a)**) was also negligible up to a Reynolds number of 1 500 examined. The Colburn *j*-factor in Figure **3(b)** was used to account for the effect of variation of fluid Prandtl numbers in the heat transfer and shows that as the Reynolds number increases the Colburn *j*-factor decreases for both the horizontal and vertical tubes in the laminar region. Again, the Colburn *j*-factors for horizontal tube are higher than that of a vertical tube and are almost parallel to each other, confirming fully developed forced and mixed convection heat transfer in vertical and horizontal tubes respectively.

As mentioned in the literature, most of the transitional flow regime experiments were generally preformed in horizontal tubes where the buoyancy force acting perpendicular to the tube axis causes buoyancy induced flow and the heat transfer is a function of Reynolds number, Prandtl number, and Grashof number. It is therefore challenging to perform experiments in the absence of buoyancy induced flow (where $Nu = f\ (Re,\ Pr)$) within the transitional flow regime of horizontal tubes. One way of achieving this is to use a vertical tube where the buoyancy force acts on the axial direction and no secondary flow exist.

Figure **3** shows that for vertical upflow direction, transition is delayed when compared to a horizontal tube. This shows that the effect of secondary flow associated with a horizontal tube causes transition to occur much earlier. The start of transition critical Reynolds number, Re_{cr}, for horizontal flow in Figure **3(b)** was found to be 2 633 and for vertical upflow direction was 2 814. Hence, transition is delayed by Reynolds number difference of about 181. This indicates that the start of transition is a strong function of Grashof number.

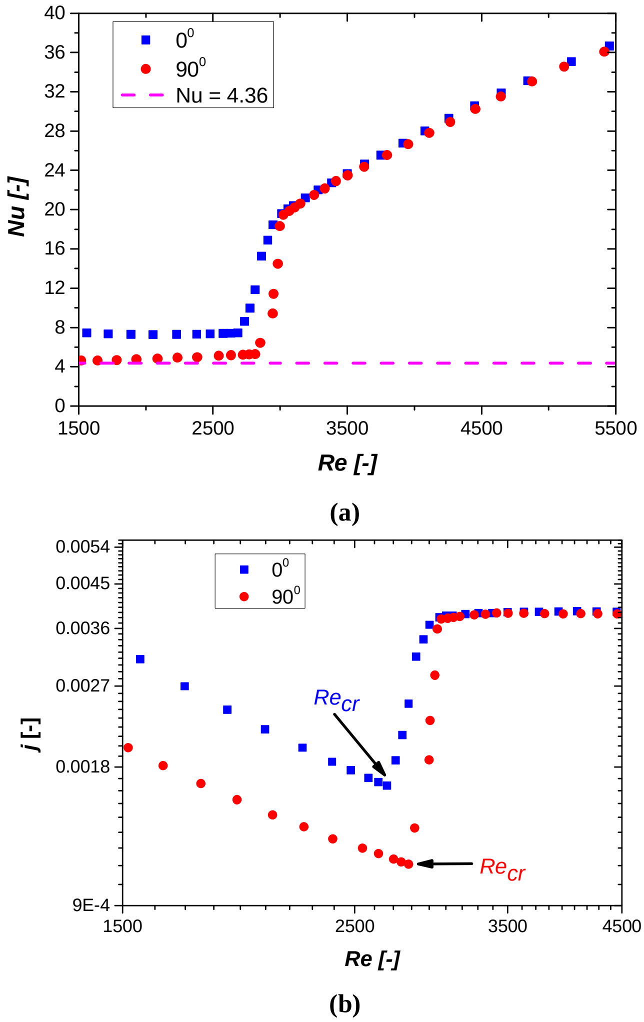

Figure 3 Comparison of average heat transfer results as a function of Reynolds number for horizontal (0^0) and vertical (90^0) upflow orientations in terms of: **(a)** Nusselt number and **(b)** Colburn *j*-factor, at heat a flux of 6 kW/m^2.

The end of transition in Figure **3(b)** where the flow enters low-Reynolds-number-end region for both horizontal and vertical tubes orientation occurs at approximately the same Reynolds number of about 3 057 and 3 067 respectively. This indicates that the effect of secondary flow in the horizontal tube decreases in the transition region from laminar region (where it is high) to turbulent region (where it is negligible). Therefore, buoyancy has a negligible effect on the end of transition region and low-Reynolds-number-end region. Again, the width of transition region (i.e. the difference between the beginning and end transition critical Reynolds numbers) for vertical tubes is much smaller than that of horizontal tubes due to delay in start of transition as shown in Figure **3**.

For the turbulent flow regime, the results (in Figure 3(**a**)) show no difference in heat transfer between the horizontal and vertical upflow direction because the turbulent motion of the fluid completely suppresses the effect of buoyancy under uniform heat flux condition. Therefore, secondary flow has no significant effect on turbulent flow heat transfer in both horizontal and vertical tubes.

CONCLUSIONS

This article presented a preliminary result of fully developed heat transfer in the laminar and transition region of smooth horizontal and vertical tubes, in an attempt to provide more comprehensive data and information on the behaviour of heat transfer in the transitional flow regime of vertical tubes. Reynolds number range of 1 000 to 10 000 was used covering the complete transition region at a constant heat flux of 6 kW/m^2. It was found that, as the effect of secondary flow becomes negligible in vertical tubes with upflow direction, the laminar flow heat transfer converges to forced convection and transition from laminar to turbulent region was delayed when compared to horizontal tubes. The width of transition region for vertical tube was much smaller than that of horizontal tube and this showed that buoyancy effects strongly influenced the start of transition critical Reynolds number with negligible effect on the end of transition. Therefore, due to these differences in the start of transition between horizontal and vertical tubes, the need to fully investigate the effect of flow direction at different inclination angles on heat transfer in the transitional flow regime is recommended for an optimized vertical and inclined heat exchanger design.

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