

## COLBURN *J*-FACTOR IN THE TRANSITIONAL FLOW REGIME IN A PLAIN CIRCULAR TUBE WITH TWISTED TAPE INSERT AND SQUARE-EDGE ENTRY

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### ABSTRACT

The flow regime at which a fluid flows is very important in heat transfer engineering and also analogous to the rate of energy consumption in the domestic or industrial applications of heat exchangers. In this article, the influence of twisted tape insert on the heat transfer coefficients in the transitional flow regime was experimentally investigated and reported. A thin typical twisted tape of twist ratio of 5 was inserted into a plain circular copper tube with water as working fluid, square-edge entry and at a constant heat flux boundary condition of  $2 \text{ kW/m}^2$  over a Reynolds number range of 500 to 10 394 and Prandtl number range of 4.32 to 6.72. The transitional flow regime was easily identified and differentiated from laminar and turbulent flow regime by plotting the local heat transfer coefficients in terms of local Colburn *j*-factors against local Reynolds numbers in the fully developed flow region. The local station considered in this study had a length-to-diameter ratio of 141. The results showed that the transitional flow regime commenced at a local Reynolds number of 981 and ended at 1 447 for the tube with twisted tape insert. In the plain tube, which was used for validation, the critical Reynolds number was attained at a local Reynolds number of 3 005 and the transitional flow regime ended at a local Reynolds number of 3 318.

### INTRODUCTION

The heat exchangers in many engineering applications are either operated in plain tubes or in enhanced tubes. The enhanced-tube-operated heat exchangers are emerging because of augmentation in heat transfer of these devices compared to the plain-tube-operated heat exchangers. The enhancement plays a significant role in the overall energy management, efficiency and the overall cost associated with operation and maintenance of the heat exchangers. The improvement of heat transfer in the heat exchangers is achieved by two methods [1]. The first is the active technique, which depends on the application of external power such as electrostatic fields, impingement of jet, surface-fluid vibration etc. The second method is known as the passive technique, this does not require the application of external power, thus making it more advantageous in terms of reduction in energy consumption and overall energy management. This passive method involves the modification of the inner surface of the heat exchanger in order to achieve higher heat transfer. This

inner surface modification or insertion of some turbulators leads to increase in the swirl flow and mixing of the working fluid flowing through the heat exchanger. The passive technique is achieved by inserting either one or a combination of some of the following devices conical-ring [1, 2], single twisted tape inserts, delta-winglet twisted tape [3] and multiple twisted tape [4] etc.

In most cases, the smooth tubes or enhanced tubes with any of the turbulators have been designed to operate in the laminar or turbulent flow regime. The intermediate regime, which is known as the transitional flow regime has been avoided both in design and operation of heat exchangers. This transitional flow regime has received limited research investigation because of insufficient information to clearly identify this regime [5], associated unsteadiness, instability and uncertainty [6] and that the region is unpredictable [7].

However, recent studies arising from the work of Ghajar and his co-workers [8-11] from Oklahoma State University and Meyer and his co-workers at the University of Pretoria, have proven the existence of the transitional flow regime, particularly in smooth tubes [12-14], micro-tubes [15] and enhanced tubes [16].

These previous studies have been able to quantify and characterise using Reynolds numbers, the transitional flow regime and have also investigated and published various means of achieving an early or delayed transition. The results published from these previous studies indicated that the transitional flow regime was influenced by the manner in which the working fluid flowed into the test section. These entry modes could either be the re-entrant inlet, the square-edge inlet, or the bellmouth inlet [8]. Other ways by which the transitional flow regime was influenced include variation in tube diameter [13] and variation in the properties of working fluid [17]. In order to achieve a higher heat transfer coefficients, the transitional flow regime provides an added advantage compared to the laminar flow regime and reduced energy cost compared to the turbulent flow regime.

In the enhanced tubes, particularly with the use of twisted tape inserts, previous investigations have concentrated on the heat transfer augmentation in the laminar and turbulent flow regimes, with no discussion on the establishment of the transitional flow regime. In the laminar flow regime, Hong and Bergles [18], reported the augmentation of heat transfer with

twisted tape inserts by heating a tube flowing with water over a range of Reynolds number of 83-2 460 and ethylene glycol over a Reynolds number range of 13-390 at a constant heat flux boundary condition. In the turbulent flow regime, Promvong and Eiamsa-ard [2], investigated the behaviour of heat transfer in a tube with conical-ring and twisted-tape inserts of twist ratios of  $\gamma = 3.75$  and  $7.5$  over the range of Reynolds numbers of 6 000 to 26 000. The results achieved an enhancement efficiency of 1.96 from the combination of the two passive devices investigated as compared to using individual device alone. The heat transfer augmentation with air as working fluid over the Reynolds number range of 5 132 to 24 989, in a plain tube with twisted tape of twist ratios of  $\gamma = 2 - 4$ , inserted separately into the inner surface of the tube wall was presented in the work of Bas and Ozceyhan [19]. The results showed that the Nusselt numbers increased as the twist ratios and the clearance between the tape and the inner wall of the tube decreased. In the experimental investigation of Manglik and Bergles [20], covering a wide range of Reynolds numbers, the investigators suggested curve fitting of data in the laminar and in the turbulent in order to identify the transitional flow regime. The authors also provided Nusselt number correlations for the laminar and turbulent flow regimes separately.

The only study available in the transitional flow regime with the use of twisted tape insert was a work-in-progress carried out by Abolarin and Meyer [21], which characterised this flow regime using Nusselt numbers against Reynolds numbers. The transitional flow regime was reported as the diversion of the laminar flow regime from that of the turbulent flow regime as the Reynolds numbers varied.

From the above, it can be concluded that, a better approach to the identification of the transitional flow regime is needed. The approach used in this study characterised the heat transfer coefficients in terms of Colburn  $j$ -factor instead of using the Nusselt numbers. The Colburn  $j$ -factor, is a function of the Nusselt number, Reynolds number and Prandtl number.

In this experimental investigation, the heat transfer in a plain circular copper tube equipped with twisted tape insert of twist ratio 5 and square-edge inlet is presented. In this experiment, water was used as the working fluid and at a constant heat flux boundary condition of  $2 \text{ kW/m}^2$ . The results are presented in terms of the local Colburn  $j$ -factors over a range of local Reynolds numbers of 500 to 10 394 in order to clearly identify the transitional flow regime in the fully developed flow region. Prior to the commencement of the experiment with the twisted tape insert, a set of experiment was conducted for the plain tube. This was carried out in order to validate the experimental setup and compare the results with available correlations.

## NOMENCLATURE

$A_c$	[m <sup>2</sup> ]	Cross sectional area
$C_p$	[J/kg.K]	Specific heat capacity at constant pressure
$D$	[mm]	Diameter of the test section
$h$	[W/m <sup>2</sup> .K]	Heat transfer coefficient
$j$	[-]	Colburn $j$ -factor
$k$	[W/m.K]	Thermal conductivity
$L$	[m]	Length
$\dot{m}$	[kg/s]	Mass flow rate
$Nu$	[-]	Nusselt number
$Pr$	[-]	Prandtl number

$\dot{q}$	[W/m <sup>2</sup> ]	Heat flux
$Re$	[-]	Reynolds number
$RTD$	[°C]	Resistance temperature detector
$T$	[°C]	Temperature
$U$	[m/s]	Velocity
$x$	[m]	Axial distance in flow direction
$\gamma$	[-]	Twist ratio
Greek symbol		
$\rho$	[kg/m <sup>3</sup> ]	Density of the working fluid
$\mu$	[kg/m.s]	Dynamic viscosity
Special characters		
—	[-]	Average
Subscripts		
$b$		Bulk
$e$		Exit
$h$		Heating
$i$		Inlet, inner
$m$		Mean or average
$s$		Surface

## EXPERIMENTAL SETUP

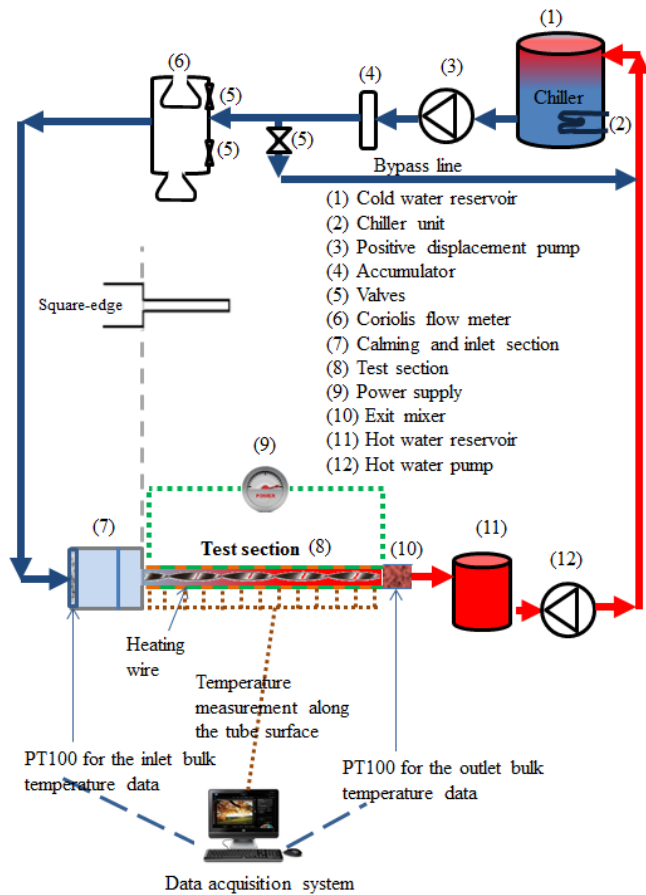
The schematic diagram of the experimental loop is shown in Figure 1. This facility consisted of a well insulated 1 000  $\ell$  water reservoir (1), which was cooled by a 15 kW chiller unit (2), a positive displacement pump (3), a 1  $\ell$  standard valve accumulator (4), control valves (5), two coriolis flow meters (6), a calming section (7), the test section (8), the power supply unit (9), an exit mixer (10), a hot water reservoir (11) and a hot water pump (12) with a maximum capacity of 270  $\ell/h$ .

The cold water used as working fluid was circulated at a temperature of about 20°C from the 1 000  $\ell$  reservoir (1) and the temperature was regularly maintained at 20°C by a 15 kW chilled unit (2) connected to this reservoir. This water was circulated through the entire loop by a positive displacement pump (3) with a maximum volume flow rate of 1 344  $\ell/h$ . Connected to this pump was a 1  $\ell$  standard valve accumulator (4) with a maximum volume flow rate of 14  $\ell/h$ . This bladder accumulator was equipped with fluid section and a gas section containing Nitrogen. The fluid section was connected to the positive displacement pump to enable the bladder accumulator draw the working fluid when the pressure was increased and the gas within the accumulator was compressed. As the pressure dropped, the gas that was compressed, expanded and forced the fluid stored in the accumulator to circulate. The purpose of this bladder accumulator was to reduce pulsation associated with the electronically operated positive displacement pump and to ensure constant pressure process as the working fluid was circulated. A bypass (5) valve was installed ahead of the accumulator to achieve a low flow rate of the working fluid in the experimental loop.

The working fluid then flowed into either of the two coriolis flow meters (6) connected in parallel. The working fluid flowed through the flow meter with a maximum capacity of 108  $\ell/h$  during the low mass flow rate, while the fluid flowed through the bigger flow meter with a maximum volume flow rate of 2 180  $\ell/h$  during the higher mass flow rate measurements. The bypass valve (5) connected to each of the flow meters was either fully opened when the flow meter was being used or closed when the flow meter was not being used.

Upon leaving the flow meter, the working fluid flowed through the calming section (7) with an inner diameter of 0.2 m and overall length of 0.716 m and an inlet section of a length of 0.254 m unto which the square-edge inlet configuration was installed. Prior to the calming section was an inlet mixing well, where a resistance temperature detector (*RTD*) was installed to measure the bulk inlet temperature of the working fluid.

The fluid flowed through the inlet section, and then through the square-edge inlet and to the test section (8). This test section was made up of a plain circular copper tube of inner diameter of 19 mm, thickness of 1.5 mm and a length of 5.27 m. The twisted tape inserted with a twist ratio of 5, was fabricated from a 1 mm copper plate, with a width of 18 mm and pitch of 90 mm. The length of the twisted tape insert was the same as that of the test section.



**Figure 1** Schematic diagram of the heat transfer experimental loop

On this test section was installed a total of 21 thermocouple stations. Each station had a total of four (4) thermocouples. This translated to a total of 84 thermocouples on the entire test section. On the outer surface of the test section was closely wound round, two heating wires with wire diameter of 0.81 mm over a length of 4.8 m. These heating wires provided the constant heat flux boundary condition during the experimentation. These heating wires were connected to a 1.5 kW power supply (9). The water

flowing through the test section was heated by the heat that the copper tube received from the electrical heating wires.

The working fluid then flowed through the test section and exited through a mixer (10) placed at the end of the test section. In this mixer was a copper plate baffle which provided the homogenous mixing of the hot fluid leaving the test section. The temperature of the hot fluid leaving the test section was then measured by a second *RTD* installed in the exit mixer.

The exited hot fluid flowed into a 1 000 l reservoir (11) and was then circulated by another positive displacement pump with a maximum volume flow rate of 270 l/h to the cold reservoir (1) equipped with the chiller unit (2).

This process was repeated by varying the mass flow rate of the fluid flowing through the cold-water pump (3). The mass flow rates ranged from laminar to transition and then to turbulent flow regime. This covered a range of Reynolds number of 500 to 10 394 and range of Prandtl number of 4.32 to 6.72.

The data obtained during the experiments was logged using a data acquisition system unto which all the thermocouples, *RTDs* and flow meters were connected and stored in a personal computer.

On this set-up, two experiments were conducted. The first was the plain tube experiments, in which the working fluid was circulated through the plain copper tube and the second experiment was carried out by inserting the twisted tape inserts in to the plain tube.

The uncertainties of all the instrumentations were determined, as the square root of the sum of the square of the bias and precision. The bias being the uncertainty of the instrumentation as provided by the manufacturers and the precision evaluated from the method of Dunn [22]. The precision was evaluated based on the standard deviation of the 400-sample size of data logged for each of the instrumentation and the student's *t* variable at a 95% confident level. The maximum uncertainties of the *RTDs* and thermocouples were 0.06°C and 0.124°C. the uncertainties of the Reynolds number varied from 0.5% to 0.3% and Colburn *j*-factor were 0.489% and 5.48% at the minimum and maximum mass flow rate respectively.

## DATA REDUCTION

The data reduction presented in this study was for one of the stations in the fully developed flow region. Upon comparing the local heat transfer coefficients all the 21 thermocouple stations on the surface of the test section, only the first thermocouple station exhibited an entrance region characteristics. From the second station, the values of the heat transfer coefficients were relatively the same along the axial direction of the flow. The results also showed that, except for the first station, the temperatures of all other twenty (20) stations increased linearly as expected for a fully developed flow region. From these two analyses, it was concluded that the flow from the second station till the end of the tube was fully developed. This conclusion is consistent with the results of the smooth and enhanced tube carried out in this study.

The results of the station with a length-to-diameter ratio of  $x/D_i = 141$  is presented and reported in this study. The heat transfer results from this station symbolise the results of other stations within the fully developed flow region.

The local mean bulk temperature at this station was evaluated using

$$T_m(x) = T_i + \left[ \frac{T_e - T_i}{L_h} \right] * x \quad (1)$$

where,  $L_h$ , is the heated length of 4.8 m, and,  $x$ , was the axial distance from the inlet of the test section to the station being considered.

All the fluid properties used during the data reduction were calculated by substituting the values of the mean bulk temperature into the equations of Popiel and Wojtkowiak [23] for each of the local dynamic viscosity, the local thermal conductivity, the local density, the local specific heat capacity and the local Prandtl number.

The local Reynolds number at the  $x/D_i = 141$ , was obtained from the measured mass flow rate, tube inner diameter, fluid dynamic viscosity and the cross-section area of the test section as

$$Re(x) = \frac{\dot{m}D_i}{\mu * A_c} \quad (2)$$

The, local heat transfer coefficient, which was a function of the heat flux of the working fluid and the difference between the local surface temperature and the mean bulk temperature, was determined using

$$h(x) = \frac{\dot{m}C_p(x)(T_e - T_i)}{\pi D_i L * [T_s(x) - T_m(x)]} \quad (3)$$

The local Nusselt number was calculated using

$$Nu(x) = \frac{h(x) * D_i}{k(x)} \quad (4)$$

The local Colburn  $j$ -factor was then calculated for each of the mass flow rates as

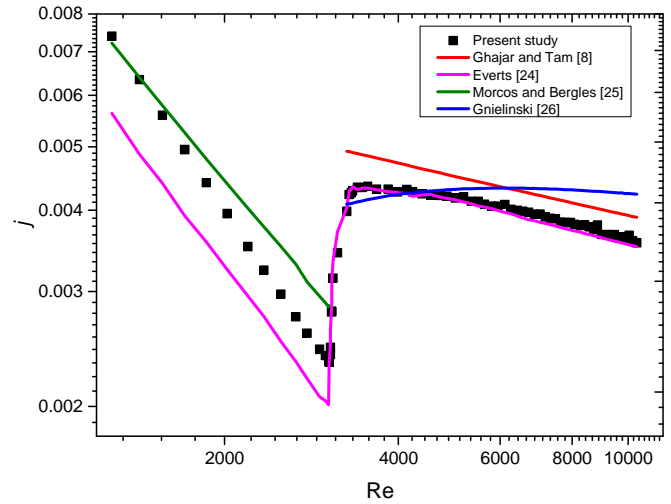
$$j(x) = \frac{Nu(x)}{Re(x) * [Pr(x)]^{1/3}} \quad (5)$$

## VALIDATION WITH LITERATURE

The local Colburn  $j$ -factors against local Reynolds numbers of the present study at the heat flux of  $2 \text{ kW/m}^2$  are presented in this section and are compared with available smooth tube correlations in the laminar and turbulent flow regimes as shown in Figure 2.

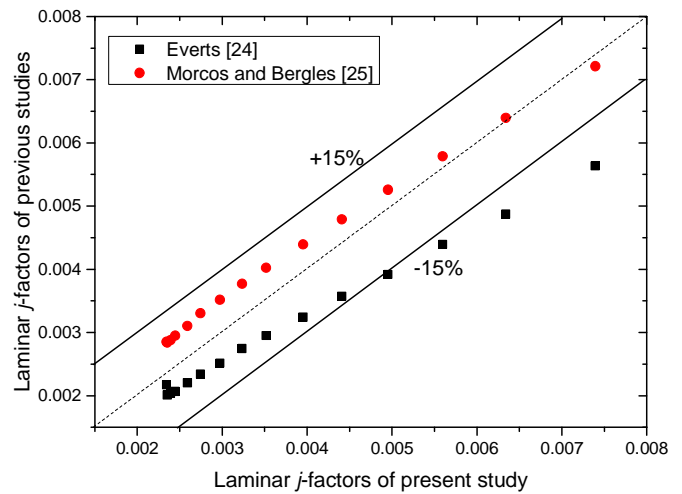
These results are presented for the selected station with length-to-diameter ratio  $x/D_i = 141$  and are validated with 74 data points over the Reynolds number range of 1 276 to 10 348, covering the laminar, transitional and turbulent flow regimes.

The local Colburn  $j$ -factors comparison in the laminar flow regime are presented in Figure 3. The equation of Everts [24] deviated below the results of the present study by an average of 15%. Also the correlation of Morcos and Bergles [25] predicted slightly higher with an absolute average deviation of 13%.



**Figure 2** The variation of the local Colburn  $j$ -factor results with the local Reynolds numbers in a plain tube at the station with length-to-diameter ratio of  $x/D_i = 141$  and heat flux of  $2 \text{ kW/m}^2$ .

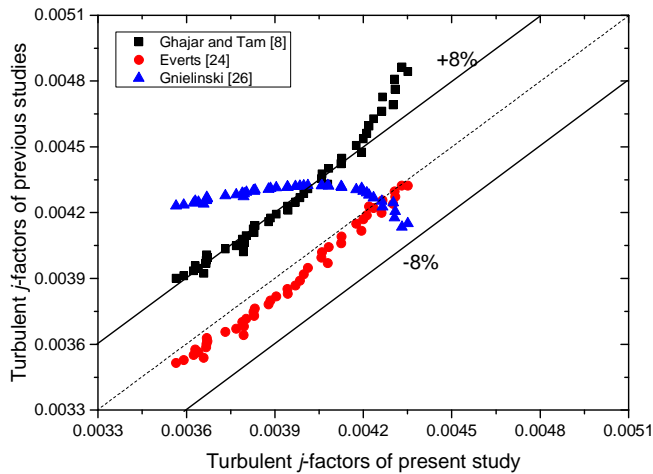
The transitional flow regime in this plain tube showed the critical Reynolds number as 3 005 and the end of transitional flow regime as 3 318.



**Figure 3** Comparison of the percentage deviation of the local Colburn  $j$ -factors against the local Reynolds numbers in the plain tube with previous studies in the laminar flow regime

In the turbulent flow regime, the percentage deviation of the present experimental data with the previous correlations are shown in Figure 4. The results showed that the present experimental data agreed well with these correlations. As presented in Figure 4, the heat transfer results evaluated using the correlation of Ghajar and Tam [8] was on an average higher than the present study by 8%. The equation of Everts [24] agreed excellently well with the present experimental data with a maximum deviation of about 4%. As the Reynolds numbers

increased, the heat transfer results of Gnielinski [26] deviated from the present study over a range of -4% to +18%.



**Figure 4** Comparison of the percentage deviation of the local Colburn  $j$ -factors against the local Reynolds numbers in the plain tube with previous studies in the turbulent flow regime

As shown in Figures 2, 3 and 4 the local heat transfer results are in good agreement with literature in the laminar and turbulent flow regimes [8, 24-26]. From the above, it can be concluded that the present experimental setup used for the plain tube provided accurate results of the heat transfer and therefore can be relied on for the enhanced tube presented in the next section.

## RESULTS AND DISCUSSION

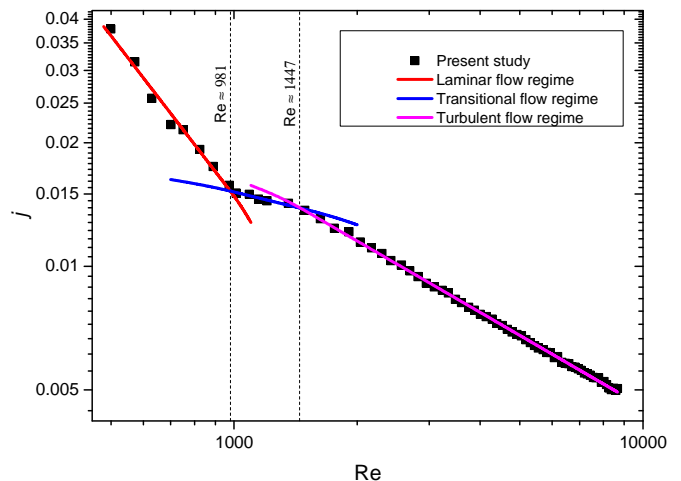
The local heat transfer results at the selected station with the length-to-diameter ratio of  $x/D_i = 141$  in the plain tube heat exchanger enhanced with typical twisted tape insert of twist ratio of  $\gamma = 5$  is presented in this section for a constant heat flux boundary condition of  $2 \text{ kW/m}^2$ . These results are presented in terms of Colburn  $j$ -factors in order to appropriately identify the three flow regimes of laminar, transitional and turbulent.

The local Colburn  $j$ -factors as plotted against the local Reynolds numbers for each of the three flow regimes are shown in Figure 5. The data points within each of the flow regimes have been grouped and characterised by a way of linear curve fitting.

The results showed that in the laminar flow regime only 8 data points were captured. The Colburn  $j$ -factors in the laminar flow regime increased very steeply as the Reynolds numbers reduced. These data points have been captured over a range of Reynolds numbers of 500 and 973. The gradient of the linear curve fitting in the laminar flow regime was evaluated as -1.30.

The transitional flow regime, which is the focus of this work, has also been clearly marked by the diversion of the data points within this regime from the data points in the laminar flow regime and that of the turbulent flow regime. Also used is the linear curve fitting placed between the end of the laminar flow regime and the beginning of the turbulent flow regime. The beginning of the transitional flow regime has been chosen to be the point of intersection of the laminar and transitional curve fittings and the end of the transitional flow regime as the point

of intersection of the transitional flow regime curve fitting and the curve fitting that characterised the turbulent flow regime. The critical Reynolds number, which indicates the commencement of the transitional flow regime was evaluated as 981 as shown in Figure 5. This transitional flow regime progressed until a Reynolds number of 1447. This transitional range of Reynolds number was achieved earlier compared with that of the plain tube. A lower negative slope was obtained from the linear curve fitting of data within this flow regime. The gradient of the line was -0.244. This indicates that the Colburn  $j$ -factors increased very slowly as compared to the other flow regimes as the Reynolds numbers reduced.



**Figure 5** The variation of the local Colburn  $j$ -factors with the local Reynolds numbers in a plain tube with twisted tape insert of twist ratio of  $\gamma = 5$  and at a station of  $x/D_i = 141$ .

In the turbulent flow regime, the Colburn  $j$ -factors reduced as the Reynolds numbers increased. The linear curve fitting on this regime had a gradient of -0.578. This indicated that the steepness of this regime is lower compared to the laminar and higher compared to the transitional flow regime.

## CONCLUSION

Heat transfer experiments in a plain tube heat exchanger have been carried out and discussed in this study with particular emphasis on the identification of the transitional flow regime. Two experiments were conducted.

The first was for the plain tube which was used for the purpose of validation. The transitional flow regime for this plain tube commenced at a Reynolds number of 3005 and ended at a Reynolds number of 3318.

The second experiment was carried out by inserting a twisted tape of twist ratio of  $\gamma = 5$  in to the plain tube. The heat transfer results, evaluated in terms of the Colburn  $j$ -factors showed that the transitional flow regime commenced at a Reynolds number of 981 and ended at a Reynolds number of 1447.

The results also indicated that each of the flow regimes could be accurately curve fitted linearly when plotted on a log-log scale.



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