HEAT TRANSFER IN THE TRANSITIONAL FLOW REGIME INSIDE SMOOTH TUBES WITH TWISTED TAPE INSERTS

Abolarin S.M. 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

A higher heat transfer at relatively low energy cost in heat exchangers used in various engineering applications is essential towards achieving sustainable energy management in the industries. This is achievable by enhancing and operating heat exchangers in the transitional flow regime as compared with the laminar flow regime. The Nusselt number results of an experimental investigation on transition in a smooth tube equipped with twisted tape insert of twist ratio 5 are presented in this study. This experiment was performed under a constant heat flux boundary condition of 2 kW/m^2, using water as working fluid over a range of $500 \leq \mathrm{Re} \leq 11\,000$ and $3.9 \leq \mathrm{Pr} \leq 6.7$. In the enhanced tube transition from laminar to turbulent occurred at a range of Reynolds numbers of 750 to 2 082, while the transition in smooth tube commenced and ended at a Reynolds number of 2 700 and 3 187 respectively. The commencement of the transitional flow regime inside the tube with twisted tape insert was observed to be earlier as compared with smooth tube counterpart. The heat transfer coefficients with twisted tape insert tube are also higher than the one in smooth tube in the all the flow regimes. This investigation makes data available for both designers and researchers of heat exchangers. Particularly in the transitional flow regime, the heat transfer results are higher at low Reynolds numbers. This increase is very useful in the industries where higher heat transfer through heat exchangers at low energy consumption and cost are required.

INTRODUCTION

Heat exchangers over the years are found in many engineering applications and research studies. Some of these include domestic services [1], industrial processes such as process and electrochemical plants [2], power station using fossil and renewable energy source like concentrated solar power stations [3], food processing [4] to mention a few.

Heat exchangers today have been designed to operate as either smooth or enhanced tubes mainly in the laminar and turbulent flow regimes, while the transitional flow regime has been disregarded during the design and operation of heat exchangers. However, recent research is now encouraging the design and operation of heat exchangers in the transitional flow regime both in smooth and enhanced tubes.

Enhanced heat exchangers as compared to smooth tubes have been identified to be accompanied with higher heat transfer coefficients in the laminar and turbulent flow regimes at relatively low costs particularly when passive enhancement techniques are employed.

In order to achieve industrial goals where the need to exchange heat from one medium to another is encountered; improving the heat transfer of such exchange is important towards realizing improved heat transfer and low costs. This means that enhanced heat exchangers, for instance with twisted tape inserts are needed to achieve these goals.

The transitional flow regime occurs in between laminar and turbulent flow regimes. Research investigation of this regime has received very few attention because of limited or inadequate understanding [5], unreliable prediction [6] and inherent uncertainty and instability [7] amongst many others.

Laminar and turbulent flow regimes have received and still receiving research attention in various forms, however, very few computational and experimental studies have being carried out to demonstrate the transitional flow regime in the operation of heat exchangers. Making more data available in this regime will help designers and operators of heat exchangers understand the transitional flow regime. This data availability is very fundamental because the heat transfer coefficients in the transitional flow regime are higher than that of the laminar flow regime. This increased heat transfer provides improved performance of the heat exchanger and ensures better energy performance. This means that most industries that are still operating heat exchangers in the laminar flow regime can now operate these devices in transitional flow regime that will help achieve a higher heat transfer at lower pressure drops as compared with the turbulent flow regime.

There are two institutions that have dedicated attention into experimental investigations in the transitional flow regime. The first is Ghajar and his co-workers from Oklahoma State University [8-12], and the second is the research work of Meyer and his students at University of Pretoria [1, 13-17].

These researchers have identified and published various factors that influence transitional flow regime in circular tubes. Some of these influencers include tube diameters [1], three different inlet configurations, namely re-entrant, square-edge and bell-mouth [9, 12, 18, 19] and working fluids [16]. Each of these inlets influence the critical Reynolds numbers from laminar to turbulent.

The use of twisted tape insert as a passive enhancement technique was pioneered by Hong and Bergles [20] who carried out an experimental investigation in the laminar regime with viscous fluid, under constant heat flux boundary condition.

Other empirical studies include the work of Watanabe, et al. [21] which use air as working fluid and Bandyopadhyay, et al. [22] which used oil as working fluid. Computational investigations in the laminar flow regime include the work of Date [23] and Date and Gaitonde [24]. The setback with the numerical studies is the idealization and assumption to simplify governing equations to solvable conditions; by this many terms are neglected.

In the turbulent flow regime, some studies have also investigated the use of twisted tape inserts under constant heat flux boundary condition and range of Reynolds numbers of 4 000 to 25 000. Some of these included the work of Eiamsa-ard, et al. [25] who used air as working fluid, Bas and Ozceyhan [26] with air as working fluid and Seemawute and Eiamsa-ard [27] with water as working fluid.

The investigation carried out by Manglik and Bergles [28], under constant temperature boundary condition suggests curve fitting data from laminar to turbulent in order to identify the transition region. The work suggests that transition occur before the Reynolds number of 10 000 is reached. .

The focus of this paper is to integrate the inlet sections that have been identified to influence transition in smooth circular tube into heat exchanger tube enhanced with twisted tape inserts. The results present the influence of twisted tape inserts inside a smooth horizontal tube with a square-edge inlet using water as working fluid.

NOMENCLATURE

A_c	[m^2]	Cross sectional area
C_p	[J/kg.K]	Specific heat capacity at constant pressure
D	[mm]	Diameter of the test section
h	[W/m^2.K]	Heat transfer coefficient
L	[m]	Length
$\dot{m}$	[kg/s]	Mass flow rate
$\dot{q}$	[W/m^2]	Heat flux
T	[°C]	Temperature
U	[m/s]	Velocity
x	[m]	Distance
Greek symbol		
ρ	[kg/m^3]	Density of the working fluid
μ	[kg/m.s]	Dynamic viscosity
Dimensionless variable		
Nu	[-]	Nusselt number
Re	[-]	Reynolds number
Special characters		
$-$	[-]	Average
Subscripts		
b		Bulk
e		Exit
f		Fluid
h		Heating
i		Inlet
m		Mean or average
s		Surface

EXPERIMENTAL SETUP

The test section consisted of a single 5 m heat exchanger tube of copper material with an inner diameter of 19 mm and outer diameter of 22 mm. The geometry of the twisted tape inserted in the heat exchanger for enhancement was fabricated with a copper plate of thickness of 1 mm, width of 18 mm and pitch of 90 mm. This means that the twisted tape has a twist ratio of 5. The flow in this study was hydrodynamically fully developed in the laminar regime and both hydrodynamically and thermally fully developed in the turbulent flow regime; since the hydrodynamic and thermal entrance lengths are equal when the flow is turbulent. This means that, in the laminar and transition regions, the thermocouples between 2.35 m and 4.8 m were used for the calculations. However, when the flow was turbulent, the thermocouples across the entire test sections were used for the calculations. The schematic diagram of the test loop is presented in Figure 1.

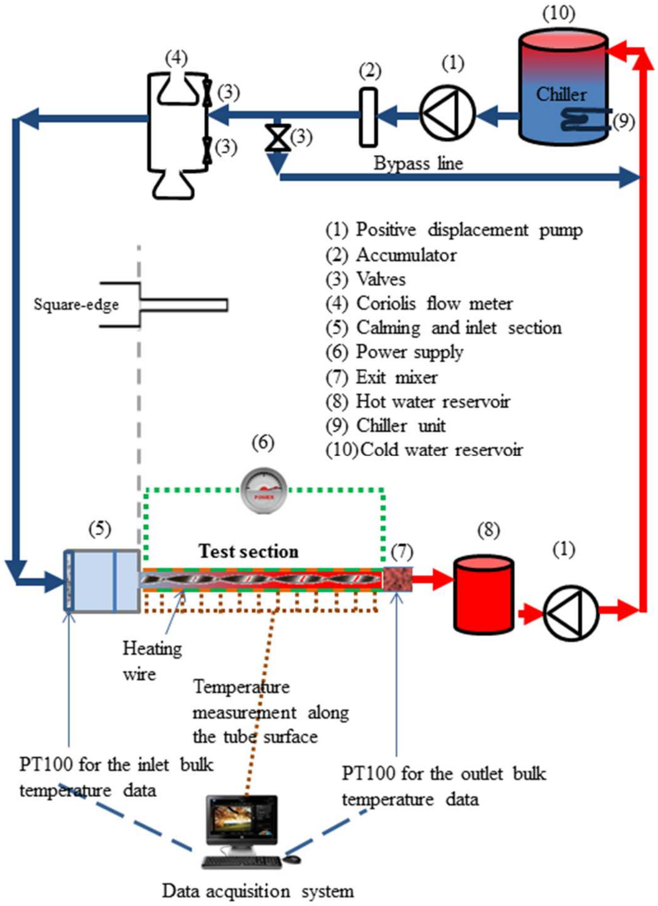

Figure 1 Schematic diagram of the heat transfer experimental loop

Water was used as the working fluid, over a wide range of Prandtl numbers of 3.9 to 6.7 and Reynolds numbers of 500 to 11 000. The inlet temperature of the working fluid was maintained at an average of 20ºC. The working fluid was circulated with two positive displacement pumps, one with a maximum volume flow rate of 1 344 ℓ/h, which was used to pump the working fluid into the test section from a 1 000 ℓ cold reservoir equipped with a 15 kW chiller unit and the other pump with a maximum volume flow rate of 270 ℓ/h which was used to circulate the fluid from the exit of the test section to a chiller. A 1 ℓ standard valve accumulator with a maximum volume flow rate of 14 ℓ/h equipped with bladders containing air was installed to assist in the reduction of pulsation associated with the electronically controlled positive displacement pump and to ensure constant pressure process.

Two coriolis flow meters were installed in parallel before the test section. One flow meter has a maximum volume flow rate of 108 ℓ/h; this was used during measurement at low mass flow rate. The other has a maximum volume flow rate of 2 180 ℓ/h; this was used during the high mass flow rate measurement.

Leaving the flow meter, the working fluid flows through a pipe and then into the calming section and square-edge inlet section and then entered the test section. The water leaves the test section back into the 1 000 ℓ cold water reservoir.

The inlet and exit temperatures of the working fluids were measured using two PT100 probes. The inlet PT100 was installed at the mixing well of the calming section, the exit PT100 was installed at the exit mixing well after the test section. To reduce heat loss, the test section was adequately insulated with armaflex insulating material. The overall thickness of insulation used was 0.144 m. With this insulation thickness, the heat loss during the experimentation was less than 1%.

On the test section was installed a total of 84 T-type thermocouples used to measure the surface temperatures of the test section at different position along the tube. These thermocouples were installed at 21 equally spaced locations along the length of the heat exchanger. The stations were equally spaced at a distance of 250 mm and each station contained four thermocouples circumferentially installed at an angle of 90º to each other.

The constant heat flux boundary condition was achieved by heating two electrically insulated wires with a wire diameter of 0.81 mm connected in parallel and wound round the outer surface of the smooth copper tube.

A comprehensive uncertainty analysis was carried out on the flow meters, PT100s and thermocouples installed on the test section for the experimentation and data reduction. This was calculated as the square root of the sum of the square of the bias and the precision. The bias was taken as the instrumentation uncertainty provided by manufacturers, while the precision was calculated according to the suggestions of Dunn [29]. The precision was calculated for each of the instrumentation from a sample data containing 400 data points as the product of standard deviation of the points and student't variable at a confidence level of 95%. The uncertainties of the inlet and outlet PT100s are 0.03ºC and 0.03ºC respectively. The uncertainties at the minimum and maximum Reynolds numbers are 0.46% and 0.26%. The minimum and maximum uncertainties of the Nusselt numbers are 2.19% and 5.42%.

DATA REDUCTION

The Reynolds numbers in the heat exchanger was calculated from the specified mass flow rate measured from the flow meter, the inner diameter of the tube, the dynamic viscosity of the working fluid obtained at the fluid bulk temperature and the cross sectional area of the tube. The Reynolds number is expressed as

$$Re = \frac{\dot{m}_f D_i}{\mu_f A_c} \tag{1}$$

The properties of the working fluid were calculated using the average (bulk) temperatures from the inlet and outlet PT100 probes. A linear variation in the temperature of the working fluid from inlet to outlet was obtained when the diabatic measurements were conducted. This is in line with previous studies [7]. The bulk temperatures in the laminar and transition region were calculated using

$$T_b = T_i + \left[\frac{T_e - T_i}{L_h}\right] * x \tag{2}$$

The fully developed bulk temperatures in the turbulent flow regime were calculated as the average of the inlet and outlet temperature using

$$T_b = \left[\frac{T_e + T_i}{2}\right] \tag{3}$$

The laminar flow regime has longer hydrodynamic entrance length compared to the turbulent flow regime. In this study, the hydrodynamically fully developed laminar flow was achieved at a distance of about 2.35 m from the entrance of the test section. After 2.35 m the velocity profile was fully developed and remained unchanged. The turbulent flow regime was achieved at a distance of about 0.19 m from the inlet of the test section, which is negligible compared to the total length of the test section, therefore the entire test section was used for the calculations. As a result, the bulk temperatures in the laminar and transitional flow regimes were calculated using the equation of a straight line shown in Eqn. 2, which evaluates the mean fluid temperature between the distance of 2.35 m and 4.8 m measured from the inlet of the test section. In the turbulent flow regime, the bulk temperatures as shown in Eqn. 3 were calculated as the arithmetic average of the mean fluid temperatures at the inlet and outlet of the test section.

The properties of the working fluid in all these three regimes were calculated according the mathematical expressions published by Popiel and Wojtkowiak [30].

Then the average heat transfer, which is a function of the heat flux of the working fluid and the difference between the average temperature on the surface of the test section, calculated using trapezoidal rule and the bulk temperate of the fluid. This is expressed as

$$\bar{h} = \frac{\dot{q}_f}{\left[\widehat{T}_s - T_b\right]} = \frac{\dot{m}_f C_{pf}(T_e - T_i)}{\pi DL * \left[\widehat{T}_s - T_b\right]} \tag{4}$$

Eqn. (4) expresses the average heat transfer coefficient obtained as a result of the balance between the heat transfer from the electrical heating wire and the heat received by the working fluid.

Then the average Nusselt number was then calculated as

$$\overline{Nu} = \frac{hD_i}{k_f} \tag{5}$$

VALIDATION WITH PREVIOUS STUDIES

The Nusselt numbers in the laminar flow regimes were compared with the correlations of Ghajar and Tam [8], Ghajar and Zurigat [10], Morcos and Bergles [31] and the experimental data of Everts [14].

In the turbulent flow, the results were compared with the work of Blasius [7], Ghajar and Tam [8], Everts [14] and the fully developed correlation of Gnielinski [32].

The average Nusselt numbers of the present data at the heat flux of 2 kW/m^2 are compared with the available correlations as shown in Figure 2. These Nusselt number data were validated with 66 data points. The Reynolds numbers vary from 1 300 to 11 048 spanning the laminar, transitional and turbulent flow regimes.

The average Nusselt numbers for hydrodynamically fully developed region in the laminar regime were calculated using bulk temperature in Eqn. (2) for the properties of the working fluid. This bulk temperature was calculated at a distance of 3.578 m from the entrance of the tube and the average surface temperature was calculated using a total of 44 thermocouples spread across 11 stations along the length of the test section. This decision was made to eliminate any effect of developing flow or thermal boundary condition. The correlations of Ghajar and Tam [8] and experimental data of Everts [14], have average deviations of about -25% and -12% respectively from the present experimental data, while the average Nusselt numbers predicted by correlation of Morcos and Bergles [31] are higher than the present data an average of 13% deviation. The present result is in excellent agreement with the correlation of Ghajar and Tam [9] with slight over prediction of 7%.

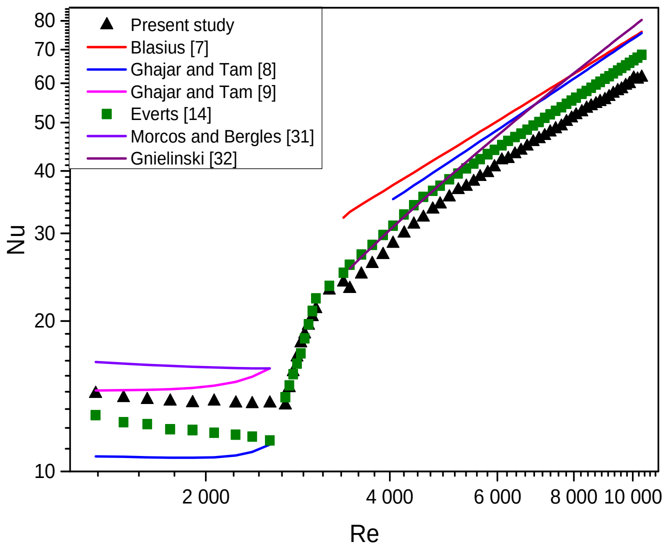

Figure 2 Comparison Nusselt number with previous correlations for smooth circular tube with square-edge inlet in the fully developed region for the heat flux of 2 kW/m^2

The properties of the working fluid in the transitional flow regime were also calculated using Eqn. (2). The transition range of Reynolds number with the square edge-inlet is from 2 700 to 3 187.

Since the flow was fully developed in the turbulent flow regime, the properties of the working fluid were calculated using the bulk temperature written in Eqn. (3). The average surface temperature was calculated using the total of 84 thermocouples spread along 21 stations on the test section. The range of Reynolds numbers in the turbulent flow regime was from 3 187 to 11 050. The Nusselt number experimental data of Everts [14] and the correlation of Gnielinski [32] predicted the present data well in the turbulent region. A maximum deviation of about 8% was recorded with these corrections. The deviation increased as the Reynolds number increased. An average of 20% deviation was reported with the correlations of Ghajar and Tam [8], this is envisaged because a high Prandtl number fluid was used for the development of this correlation.

RESULTS

The results of the average Nusselt numbers of the heat exchanger tube enhanced with twisted tape of twist ratio of 5 at the heat flux of 2 kW/m^2 is presented in Figure 3. The result presented shows the data points in the laminar, transitional and turbulent flow regimes in conjunction with previously developed correlations in the laminar and turbulent regimes.

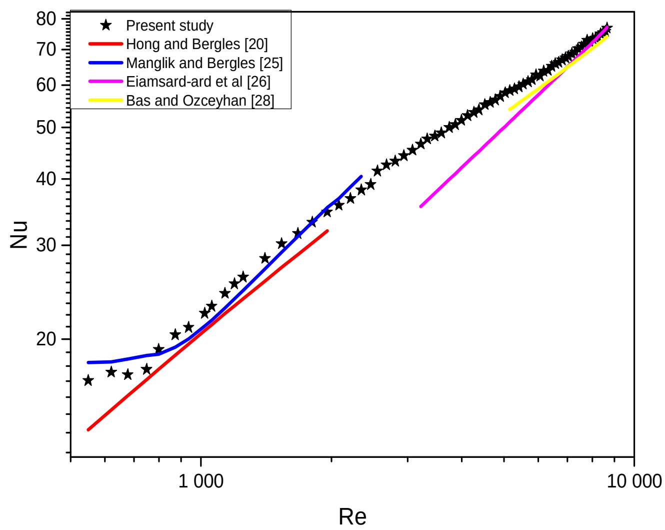

Figure 3 Average Nusselt numbers as a function of Reynolds number in heat exchanger equipped with twisted tape insert of twist ratio 5 and square-edge inlet in the fully developed region for the heat flux of 2 kW/m^2.

In the laminar flow region, the Nusselt numbers are compared with the correlations of Manglik and Bergles [28] and Hong and Bergles [20]. This region was identified by the asymptotic nature of the data. This means that the Nusselt numbers tend to approach the same value and curve tends to be relatively horizontal, as also found in the case of smooth tube. This asymptotic curve is also obtained from the correlation of Manglik and Bergles [28] carried out under constant wall temperature boundary condition and this correlation over predicted the Nusselt number in the laminar region by an average deviation of 6%. However, the Nusselt numbers of the present study is higher that the correlation of developed by Hong and Bergles [20] carried out on laminar flow regime. This correlation did not show the asymptotic curve. The Nusselt numbers in the laminar flow regime with the twisted tape insert is also higher than that of the smooth tube. The increase in the present results is due to higher heat transfer area and swirl mixing in the heat exchanger.

The transition from laminar is very obvious from Figure 3, the transition commenced at a Reynolds number of 750. However, the transition was prolonged until the Reynolds number of 2 082. The Nusselt numbers in this transitional flow regime with twisted tape insert are also higher than that of the smooth tube counterpart.

In the turbulent flow regime, the Nusselt numbers results were compared with the correlations of Eiamsa-ard, et al. [33] and Bas and Ozceyhan [26]. At the highest Reynolds number considered the Nusselt numbers were fairly the same. However, as the Reynolds numbers reduces, the Nusselt numbers of this present are significantly higher than that predicted by these two correlations. The Nusselt numbers are also observed to diverge from these two correlations.

The commencement of the transitional flow regime with the use of the square-edge inlet on the heat transfer was observed to be earlier as compared with smooth tube results. The transition region of the heat transfer results of the present study was slightly earlier compared to the available correlation.

CONCLUSION

An empirical investigation on the effect of square-edge inlet configuration on the critical Reynolds numbers of heat exchanger enhanced with twisted tape insert has been carried out in this study. The range of Reynolds numbers considered was from laminar to turbulent. The heat flux boundary condition of 2 kW/m^2 has been investigated with water as working fluid. The results show that the beginning and ending of the transitional flow regime from the laminar to turbulent was earlier as compared with smooth tube. The transitional flow regime with the heat exchanger equipped with twisted tape inserts with twist ratio of 5 was within the range of Reynolds numbers of 750 to 2 082 and is found to occur earlier compared with transitional flow regime in smooth tube heat exchanger, which commenced at the Reynolds number 2 700 and progressed up to a Reynolds number of 3 187.

A higher heat transfer with minimum expense is vital for improved productivity in heat exchangers. One of the ways to achieve a higher heat transfer coefficient is to design and operate heat exchangers in the transitional flow regime. As could be observed from the present results, the heat transfer coefficient with twisted tape insert is higher in the transitional flow regime than in the laminar flow regime and is also higher than the smooth tube. This is a double advantage.

Designing and operating existing or new heat exchangers in the transitional flow regime with twisted tape inserts will help in the advancement of energy management and more productivity.

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