

HEAT TRANSFER CHARACTERISTICS OF DEVELOPING FLOW IN THE TRANSITIONAL FLOW REGIME OF A SOLAR RECEIVER TUBE

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ABSTRACT

The transitional flow regime has been mostly avoided by designers due to uncertainty and perceived chaotic behaviour. However, changes in operating conditions, design constraints or additional equipment can cause that the flow to move into the transitional flow regime. Previous work done in the transitional flow regime focused on fully developed flow or average measurements of developing and fully developed flow across a tube length and developing flow in the transitional flow regime have not been investigated yet. Therefore, the purpose of this study is to investigate the heat transfer characteristics of developing flow in the transitional flow regime of a solar receiver tube and is work in progress. An experimental set-up was designed, built and validated and heat transfer measurements were taken at a heat flux of 6.5 kW/m² between Reynolds numbers of 500 and 10 000. It was found that the width of the transition region decreased along the tube length and the heat transfer coefficients decreased as the flow approached fully developed flow.

INTRODUCTION

Heat exchangers have a wide range of applications including solar power and engineers need accurate correlations to optimise the design of solar receiver tubes. In the design process they usually have a choice to select between a flow regime that is either laminar or turbulent. The aim is to obtain high heat transfer coefficients and low pressure drops since the pressure drop is related to pumping power and thus operational running cost. Laminar flow provides low pressure drops, but unfortunately low heat transfer coefficients as well, while the opposite is true for turbulent flow. The best compromise between high heat transfer coefficients and low pressure drops is usually in or close to the transitional flow regime between laminar and turbulent flow. Changes in operating conditions, design constraints and additional equipment can also cause that the flow regime is transitional.

NOMENCLATURE

A	[m ²]	Area
c_p	[J/kg.K]	Specific heat at constant pressure
D	[m]	Diameter ¹
EB	[-]	Energy balance
h	[W/m ² K]	Heat transfer coefficient
I	[A]	Current
j	[-]	Colburn j -factor
k	[W/mK]	Thermal conductivity
L	[m]	Length
M	[-]	Axial conduction number
\dot{m}	[kg/s]	Mass flow rate
Nu	[-]	Nusselt number
Pr	[-]	Prandtl number
\dot{Q}	[W]	Heat input
\dot{q}	[W/m ²]	Heat flux
R	[°C/m]	Thermal resistance
Re	[-]	Reynolds number
T	[°C]	Temperature
V	[V]	Voltage
x	[m]	Distance from inlet

Special characters

ρ	[kg/m ³]	Density
μ	[kg/m.s]	Dynamic viscosity

Subscripts

b	Bulk
c	Cross-section
cr	Critical Reynolds number
i	Inlet/ inner
l	Laminar
lre	Low-Reynolds-number-end
m	Mean
o	Outlet/ outer
s	Surface
t	Turbulent

¹ Except when defined differently with a subscript o to indicate outer diameter

Designers are usually advised to avoid the transitional flow regime since the flow is believed to be unstable and chaotic. In this flow regime the flow alternate between laminar and turbulent and turbulent eddies will occur in flashes known as turbulent bursts. This might cause the pressure drop to increase and order of magnitude [1].

According to a recent review paper by Meyer [1], flow in the transitional flow regime has been mainly investigated by Professor Ghajar from Oklahoma State University and his co-workers and Professor Meyer from the University of Pretoria and his co-workers. Ghajar and co-workers investigated the influence of different inlet geometries and focussed on fully developed flow only [2 – 11], except for Tam *et al.* [9] who investigated the isothermal and diabatic friction factors of developing and fully developed flow. Meyer and co-workers considered the average measurements across a tube length, therefore their data contained both developing (laminar and transitional flow regimes) and fully developed (turbulent flow regime) data [12 – 15]. However, the focus of their studies was not on developing flow since the average measurements were used only.

The thermal entrance length is a function of the tube diameter, Reynolds number and Prandtl number. Parabolic plants usually consist of several 4 m receiver tubes. Up to 10 receiver tubes are connected to each other before a bend in the tube occurs. The total heated length would then be 40 m. Therefore, if the thermal entrance length is greater than 30 m, more than 75% of the tube will have developing flow. When a thermal oil with a Prandtl number of 5 is used, the diameter should then be greater than 60 mm. However, when a glycol mixture or oil (with an average Prandtl number of 33) is used, the diameter should only be greater than 9 mm in order to have developing flow in more than 75% of the tube.

It can therefore be concluded that the flow in solar receiver tubes will be developing, rather than fully developed. In order to optimise the design of solar receiver tubes, designers require more insight in the heat transfer characteristics of developing flow. Therefore, the purpose of this study is to investigate the heat transfer characteristics of developing flow in the transitional flow regime in a smooth horizontal solar receiver tube.

EXPERIMENTAL SET-UP

The experimental set-up is shown in Figure 1 and consisted of a closed water loop which circulated the test fluid from the storage tank through the test section and back using a positive displacement pump. Water was used as the test fluid and the temperature of the storage tank was kept at approximately 20 °C. Flow pulsations were introduced into the system due to the pump, therefore an accumulator was installed prior to the flow meters to dampen the pulsations. This ensured a constant pressure at the inlet of the test section.

A bypass valve was inserted between the accumulator and the flow meters to allow the water to flow back into the tank. The bypass valve was also used to increase the pump speed for a specific flow rate, since the pulsations decreased with increasing pump speed. The valve positions were continuously adjusted to minimise the flow pulsations for all the measurements since the stability of flow is crucial when studying transitional flow.

Two Coriolis mass flow meters with different capacities were installed in parallel to measure the mass flow rates. The flow meters were used according to the flow rate requirements to minimise the uncertainty of the mass flow measurements. After the flow meters, the fluid flowed through the calming section to the test section and back into the storage tank.

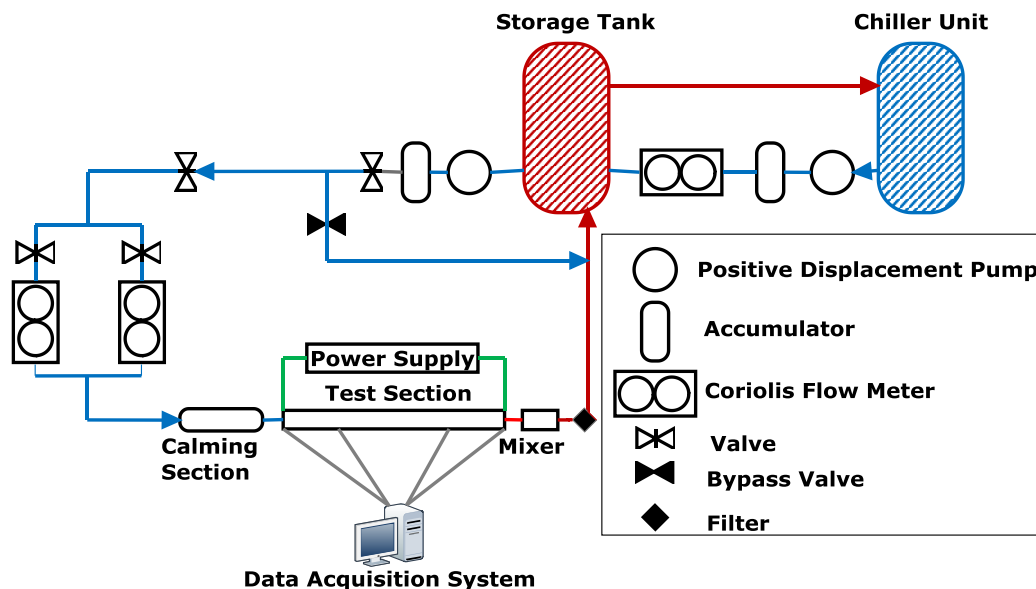


Figure 1 Schematic of experimental set-up used to conduct heat transfer and pressure drop measurements. Water was circulated from the storage tank through the test section and back using a pump. An accumulator was used to dampen the flow pulsations and the mass flow rate was measured using Coriolis flow meters. The temperatures, pressures and flow rates were recorded using a data acquisition system.

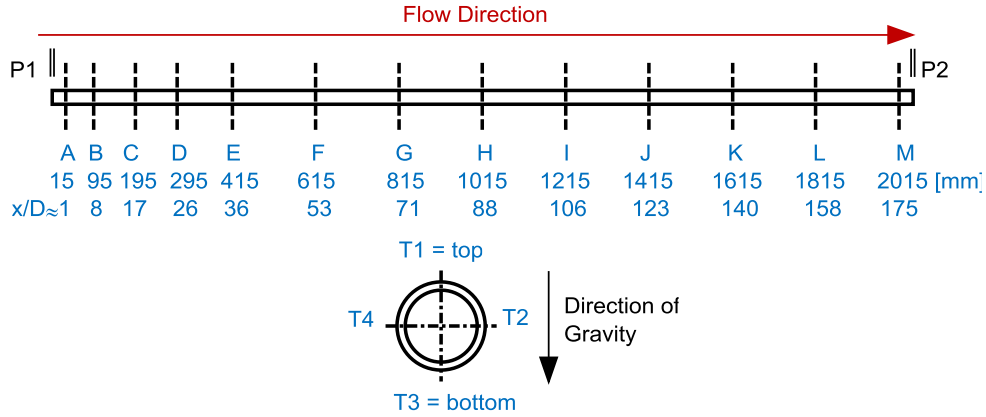


Figure 2 Schematic representation of test tube indicating the two pressure taps, P1 and P2, as well as the 13 thermocouple stations, A - M. A cross-sectional view of the test tube is also included to indicate the four thermocouples spaced around the periphery of the tube.

The mass flow rates were controlled by frequency drives that were connected to the pump, therefore the required flow rate was obtained by increasing or decreasing the pump speed. The frequency drives were also connected to a personal computer via the data acquisition system. A Labview program was used to record the data points and a MATLAB program was used to read the measured raw data and process the results.

A square-edged inlet is characterised by a sudden contraction from the calming section diameter to the test section diameter and was used in this study. The test section was manufactured from hard-drawn copper tubes with an inside diameter and length of 11.52 mm and 2 m, respectively. The tube roughness was measured with a hand-held roughness tester TR200 to be approximately 0.455 μm to 0.508 μm . The relative roughness was approximately 0.00004 and therefore for all practical purposes, the tube was considered as a smooth tube. The tube was insulated with 150 mm thick insulation with a thermal conductivity of 0.034 W/m.K and the maximum heat loss was calculated to be less than 1%.

T-type copper-constantan thermocouples were used to measure the surface temperatures at the desired locations. Thirteen thermocouple stations were spaced along the tube length, as shown in Figure 2. Four thermocouples were used at each thermocouple station to investigate possible circumferential temperature distributions caused by secondary flow. One thermocouple was placed inside the calming section to measure the inlet water temperature. Another thermocouple station, consisting of four thermocouples, was located in the mixing section to measure the outlet water temperature. The thermocouples were soldered to the test section by first drilling a 1.8 mm depression into the tube. Flux and solder were inserted into the depression and heated up to the melting point. The thermocouple was then inserted into the depression and the heat was removed in order for the tube to cool down. The thermocouples were checked to ensure good contact with the tube. In-situ calibration was done once the test section was built completely since the properties of the thermocouple junction may change when soldering it to the tube. The thermocouples were calibrated using PT-100 probes at the inlet

of the calming section, outlet of the mixing section and in the thermal bath.

To obtain a constant heat flux boundary condition, two constantan wires with a diameter of 0.38 mm were coiled around the test tube. Two heating wires were connected in parallel to decrease the resistance and current flowing through each wire, in comparison to using a single wire.

To ensure a uniform outlet temperature, a mixer was inserted after the test section to mix the water in the tube. The mixer design was based on work done by Bakker *et al.* [16].

DATA REDUCTION

For tube flow, the Reynolds number was calculated using the following equation:

$$Re = \frac{\dot{m}D}{\mu A_c} \quad (1)$$

where \dot{m} is the mass flow rate inside the tube, D is the inner tube diameter, μ is the dynamic viscosity and A_c is the cross-sectional area of the tube.

The properties of water were determined using the thermophysical correlations for liquid water [17] at the bulk fluid temperature for the average values and at the mean fluid temperature for the local values. A constant heat flux boundary condition was applied to the tubes, thus the temperature of the water increased linearly. The bulk fluid temperature was the average of the inlet (obtained from a thermocouple inside the calming section) and outlet (obtained from the thermocouples inside the mixing section) temperatures of the fluid:

$$T_b = \frac{T_i + T_o}{2} \quad (2)$$

The cross-sectional area of the tube was calculated as follows:

$$A_c = \frac{\pi}{4} D^2 \quad (3)$$

The electrical energy input remained constant, resulting in a constant heat flux. The heat flux was determined from the following equation:

$$\dot{q} = \frac{\dot{Q}_{electric}}{A_s} = \frac{VI}{\pi DL} \quad (4)$$

The thermal conductivity of copper is 401 W/m.K, which is very high. The thermal resistance across the tube wall was calculated using the following equation:

$$R_{tube} = \frac{\ln\left(\frac{D_o}{D}\right)}{2\pi Lk} \quad (5)$$

The thermal resistance and heat input was known, therefore the temperature difference across the tube wall was calculated as follows:

$$\Delta T = \dot{Q}R_{tube} \quad (6)$$

The thermal resistance was calculated to be 7.9×10^{-6} . Therefore when a heat flux of 6.5 kW/m^2 was applied to the tube; the temperature difference across the tube wall was approximately $0.0036 \text{ }^\circ\text{C}$. Since this temperature difference is very small, the temperature measured at the outer surface of the tube was used as the temperature on the inside of the tube. The average surface temperature of the tube was obtained by averaging the temperature of all the thermocouples stations.

The average heat transfer coefficient was then determined from the following equation since the heat flux, surface temperature, and bulk fluid temperature were known:

$$h = \frac{\dot{q}}{(T_s - T_b)} \quad (7)$$

The local heat transfer coefficients were calculated using the following equation:

$$h(x) = \frac{\dot{q}}{(T_s(x) - T_m(x))} \quad (8)$$

The average of the four thermocouples at a station was used as the surface temperature at that thermocouple station, $T_s(x)$. The mean fluid temperature, $T_m(x)$ was the temperature of the water near the centre line of the tube and was found by using the gradient of the line joining the inlet and outlet temperatures of the fluid:

$$T_m(x) = \left(\frac{T_o - T_i}{L}\right)x + T_i \quad (9)$$

Finally, the Nusselt number was determined as follows:

$$Nu = \frac{hD}{k} \quad (10)$$

where k is the thermal conductivity of the water obtained using the thermophysical equations of liquid water. The heat rate to the water, $\dot{Q}_w = \dot{m}c_p(T_o - T_i)$, was compared to the

electric power of the power supply by using the following energy balance:

$$EB = \left| \frac{\dot{Q}_{electric} - \dot{Q}}{0.5(\dot{Q}_{electric} + \dot{Q})} \right| = \left| \frac{VI - \dot{m}c_p(T_o - T_i)}{0.5(VI + \dot{m}c_p(T_o - T_i))} \right| \quad (11)$$

The energy balance at the higher Reynolds number ($Re = 10\,000$) was approximately 6% due to the small temperature differences between the inlet and outlet temperatures of the test fluid. However, an energy balance of less than 1.5% was obtained when the Reynolds number was lower than 4 000.

The heat transfer results were also investigated in terms of the Colburn j -factor to account for the variation in the fluid Prandtl number. The Colburn j -factor is:

$$j = \frac{Nu}{RePr^{1/3}} \quad (12)$$

UNCERTAINTIES

The method suggested by Dunn [18] was used to calculate the uncertainties of the test section and all the uncertainties were calculated within the 95% confidence interval. The Reynolds number uncertainty remained approximately constant at 1% for all Reynolds numbers. The Nusselt number and Colburn j -factor uncertainties remained approximately constant at 4.6% for Reynolds numbers below 6 000 and increased slightly at Reynolds numbers greater than 6 000. This slight increase was due to the temperature differences between the inlet and outlet fluid temperatures, as well as between the surface and fluid, which decreased with increasing Reynolds number. Both Nusselt number and Colburn j -factor uncertainties were slightly higher (approximately 5%) during transition ($Re \approx 2\,300$) due to the temperature fluctuations which occurred inside the tube.

VALIDATION

The average laminar Nusselt numbers obtained between Reynolds numbers of 600 and 1 900 when a heat flux of 6.5 kW/m^2 was applied, were used for the laminar validation. The results were compared with the correlation of Ghajar and Tam [2] and the average deviation over the whole laminar Reynolds number range was 5%. To validate the turbulent Nusselt numbers, the Reynolds number was varied between 4 000 and 10 000 and a heat flux of 14 kW/m^2 was applied. The average Nusselt numbers correlated well with the experimental data of Meyer *et al.* [19] for turbulent flow and the average deviation was less than 2%.

RESULTS

Previous studies focussed primarily on fully developed flow or average measurements along a tube length, therefore this study was devoted to the heat transfer characteristics of developing flow. The Nusselt numbers and Colburn j -factors were calculated at each thermocouple station (Figure 2) and compared with each other.

Figure 3 contains the Nusselt numbers along the test section at a heat flux of 6.5 kW/m^2 . The Reynolds number was varied

between 500 and 10 000 to ensure that the whole transitional flow regime, as well as sufficient parts of the laminar and turbulent flow regimes, was covered. From Figure 3 it follows that the Nusselt numbers at the thermocouple station close to the inlet of the test section ($x/D = 1.3$) were significantly higher than at the other thermocouple stations. This is to be expected as the thermal boundary layer at this station is thinner than at the other stations. The local heat transfer coefficients were a maximum at the inlet of the test section and then decreased along the tube length as the flow approached fully developed flow.

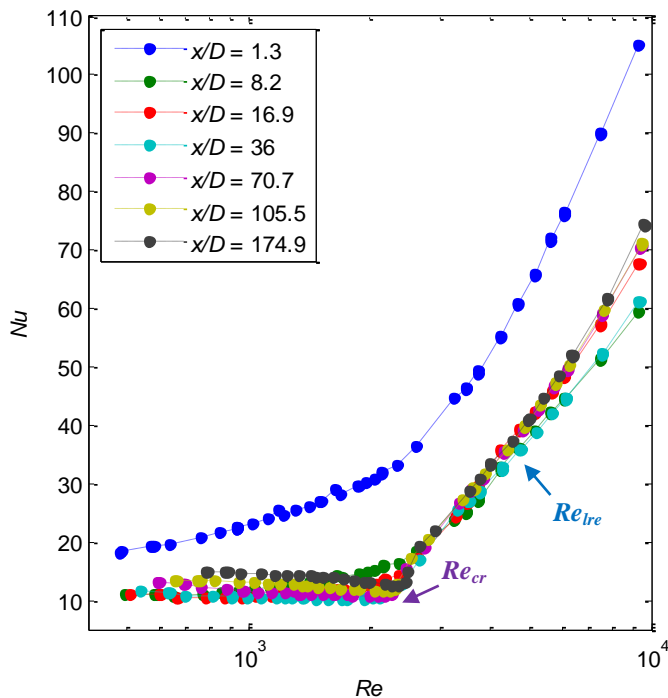


Figure 3 Comparison of Nusselt numbers at different x/D locations along the test section for Reynolds numbers between 500 and 10 000 at a heat flux of 6.5 kW/m^2

However, from Figure 3 it follows that between $x/D = 36$ and $x/D = 174.9$, the Nusselt numbers in the laminar flow regime increased along the tube length. This is due to the effects of secondary flow which exists due to the temperature difference between the fluid near the surface and the fluid outside the thermal boundary layer. Near the inlet of the test section, the thermal boundary layer was very thin and the secondary flow effects were suppressed. As the thermal boundary layer increased along the tube length, there was more “room” for secondary flow and the heat transfer coefficients and Nusselt numbers increased.

At $x/D = 1.3$ and $x/D = 8.2$, the Nusselt numbers increased gradually with increasing Reynolds number and a clear distinction between the different flow regimes could not be made. At $x/D = 16.9$, the laminar and turbulent flow regimes could be identified. The laminar flow regime is where the Nusselt numbers formed an approximate straight line between Reynolds numbers of 400 and 1 300, while the turbulent flow regime is where the Nusselt numbers formed a diagonal line

between Reynolds numbers of 6 000 and 10 000. The different flow regimes can be better identified between $x/D = 70.7$ and $x/D = 174.9$. Between Reynolds numbers of 500 and 2 300 the flow was laminar and the Nusselt numbers remained approximately constant. At a Reynolds number of approximately 2 300 (Re_{cr}), the Nusselt numbers began to increase and deviate from the horizontal line, which indicates the start of the transitional flow regime. At $x/D = 174.9$, the gradient of the Nusselt numbers changed at a Reynolds number of approximately 4 500 (Re_{lre}) which indicates the end of the transitional flow regime. The low-Reynolds number-end and turbulent flow regimes followed after a Reynolds number of approximately 4 500, however no clear distinction between these two flow regimes could be made.

Figure 4 contains the Colburn j -factors for Reynolds numbers between 500 and 10 000 at different x/D locations along the test section at a heat flux of 6.5 kW/m^2 . Similar to the Nusselt number results, the Colburn j -factors decreased with increasing x/D near the inlet of the test section, but then increased with increasing x/D due to the effects of secondary flow. Although the width of the transition region seemed to be constant, it was found that it decreased slightly with increasing x/D .

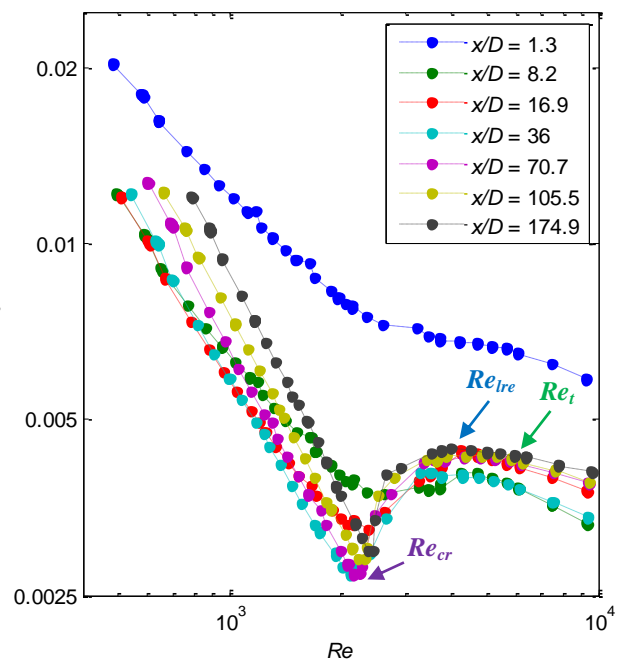


Figure 4 Comparison of Colburn j -factors at different x/D locations along the test section for Reynolds numbers between 500 and 10 000 at a heat flux of 6.5 kW/m^2

Between Reynolds numbers of 500 and 2 000 (depending on the value of x/D) the Colburn j -factors formed a straight diagonal line, which indicated that the flow is laminar. As x/D was increased, the Reynolds number at which the Colburn j -factors began to deviate from this line increased, which implied that the laminar flow regime was extended and transition was delayed. The transition region started at the Reynolds number that corresponds to the point (Re_{cr}) where the first abrupt

change in the Colburn j -factor occurs and the Colburn j -factors began to increase. The end of the transition region and the start of the low-Reynolds-number-end regime is at the Reynolds number where the gradient of the Colburn j -factors was approximately zero (Re_{tre}). Between Reynolds numbers of 6 200 and 10 000 (depending on the value of x/D) the Colburn j -factors formed a straight diagonal line, which indicated the turbulent flow regime (Re_t). The low-Reynolds-number-end regime is between the end of the transitional regime and the start of the turbulent flow regime.

Overall it can be concluded that the width of the transition region decreased slightly with increasing values of x/D . Although Ghajar and Tam [4] investigated fully developed flow, the authors found that the width of the transition region increased with increasing values of x/D . Therefore, heat transfer characteristics of developing and fully developed flow are significantly different and engineers need more insight in to the heat transfer characteristics of developing flow in order to optimise the design of solar receiver tubes.

CONCLUSION

This paper presented the heat transfer data for developing flow in solar receiver tubes in the transitional flow regime. It was found that the width of the transition region decreased with increasing values of x/D and the heat transfer coefficients decreased as well. It was concluded that the heat transfer characteristics of developing flow differed significantly from fully developed flow, therefore accurate heat transfer correlations for developing flow in the transitional flow regime are required. In order to be able to develop heat transfer correlations for developing flow, the whole developing flow region should be investigated up to the point where the flow becomes fully developed. It is recommended that future work should include measurements along a longer test section.

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