Heat Transfer and Pressure Drop Characteristics of Smooth Horizontal Tubes in the Transitional Flow Regime

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The operating conditions of many heat exchangers are in, or close to, the transitional flow regime. However, in this regime, not a lot of design information is available and some design books even recommend to not design heat exchangers to operate in the transitional flow regime. Furthermore, it is known that the type of inlet of heat exchangers influences the transition characteristics. It was therefore the purpose of this study to measure heat transfer and pressure drop characteristics in smooth horizontal tubes using different types of inlets. The types of inlets were hydrodynamically fully developed, square-edged, re-entrant and bellmouth. Experiments were conducted on a 14.48 mm inner diameter horizontal tube in which the water was cooled. Reynolds numbers ranged between 1 000 and 20 000 and Grashof numbers were in the order of 10^5. It was found that, for adiabatic flow, the square-edged inlet delayed transition to Reynolds numbers of around 2 600, while the bellmouth inlet delayed it to about 7 000. However, for diabatic flow, the transition was independent of the type of inlet. Laminar friction factors were much higher than their theoretically predicted values due to the secondary flows increasing the amount of mixing in the tube. Heat transfer measurements showed that transition with water was totally independent of the type of inlet used.

Keywords: *smooth tubes, geometries, transition, flow regime, heat transfer, pressure drop, inlet*

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INTRODUCTION

Heat exchangers have many applications in the domestic and industrial sectors. In the domestic sector it is used in vapour compression heat pumps [1-6] for the heating and cooling of air and hot water, drying of grain, heating of swimming pools, etc. In industry there are many applications, including boilers, cooling towers, evaporators, heating and air-conditioning of buildings, radiators on internal combustion engines, etc. [7]. Many of these heat exchangers are using smooth tubes with a fluid such as water that operates in the transitional flow regime.

Despite much work on transition, and even though it is of considerable importance in determining pressure drop and heat transfer in convective flow, the underlying physics and implications of this phenomenon have prevented complete understanding [8]. According to ASHRAE [9], predictions are unreliable in the transitional flow regime. Cengel [10] mentions that although transitional flow exists for Reynolds numbers between 2 300 and 10 000, it should be kept in mind that in many cases the flow becomes fully turbulent when the Reynolds number is larger than 4 000. It is normally advised when designing heat exchangers, to remain outside the transitional flow regime due to the uncertainty and flow instability in this region. For this reason, little design information is available with specific reference to heat transfer and pressure drop data in the transitional flow regime.

Ghajar and co-worker performed extensive studies on the effect of three different types of inlets on the critical Reynolds number. Three of more than seven articles are referenced here [11-13]. More recent work on the effect of different inlet geometries has been conducted by Mohammed [14]. However, his work was limited to the laminar flow regime for a Reynolds number range of 400 to 1 600. Furthermore, Ghajar and Tamas well as Mohammed, used a constant heat flux boundary condition. This condition does not cool, but heats fluids in a tube, resulting in an irregular wall temperature, as the condition would be with water flowing in the inside of the tubes of a shell-and-tube heat exchanger (like in the case of the current work). More differences between constant wall temperature and constant heat flux results, and why the results are not comparable with this study, are summarised in Olivier [15].

The purpose of this paper is to present alimited review [16-21] of measured heat transfer and pressure drop data in the transitional flow regime of water flowing in a horizontal circular smooth tube while the temperature of the wall remains fairly constant. The constant wall temperature is the same operating condition experienced in water chillers where water is cooled in the inner tubes of a shell-and-tube heat exchanger with refrigerant boiling on the outside of the water tubes.

EXPERIMENTAL SET-UP

A tube-in-tube heat exchanger in a counterflow configuration was used as the test section in an experimental set-up, schematically shown in Fig. 1. Distilled water was used as the working fluid for both streams, with the inner fluid being hot and the annulus fluid being cold. The inner inlet tube temperature was 40-45ºC and the annulus inlet temperature was 20ºC. Heating of the test fluid of the inner tube was done by means of a secondary flow loop containing water from a large reservoir. The temperature in this reservoir was maintained at approximately 60ºC by means of an electric heater, while the reservoir for the annulus water was cooled with a chiller.

The test fluid was pumped through the system with two electronically-controlled positive displacement pumps. The two pumps were installed in parallel and were used in accordance with the flow rate requirements. The cold water loop was connected to a second large reservoir, which again was connected to a chiller. The water was circulated through the system via an electronically-controlled positive displacement pump. Coriolis flow meters were used to measure the mass flow rates.

For three of the four different types of test sections, the flow first went through a calming section, as shown schematically in Fig. 2, before it entered the test section. The purpose of the calming section was two-fold: firstly, to remove any unsteadiness in the flow and to ensure a uniform velocity distribution and, secondly, to house three of the four different types of inlets under investigation.

The calming section geometry was based on work conducted by Ghajar and Tam [11] and consisted of a 5° diffuser, which increased from a diameter of 15 mm to 140 mm. This specific angle was chosen to prevent flow separation from the diffuser wall. Three screens were placed behind the diffuser with an *open-area ratio* (OAR) of 0.31. The OAR is the ratio of the area occupied by the holes to the total area occupied by the entire screen. A honeycomb section, with an OAR of 0.92, followed the screens. A wire mesh, with the wires measuring a diameter of 0.8 mm and the OAR being 0.54, was placed in front and behind the honeycomb. Another fine wire mesh was inserted between the last honeycomb mesh and the test inlet. This mesh had a wire diameter of 0.3 mm and an OAR of 0.17.

Downstream, and connected to the calming section, three different inlets (Fig. 3) could be housed, namely a *square-edged*, *re-entrant* and a *bellmouth* inlet. These inlets are also shown in Fig. 2 as items a, b and c respectively. The calming section was designed in such a way that the inlets could easily be interchanged.

The square-edged inlet is characterised by a sudden contraction of the flow. This is a typical situation encountered in the header of a shell-and-tube exchanger. The re-entrant inlet makes use of the square-edged inlet, except that the tube slides into the inlet by a tube diameter. This would simulate a floating header in a shell-and-tube heat exchanger. The third type of inlet is the bellmouth, which is characterised by a smooth contraction and has a contraction ratio of 8.8. The shape of the bellmouth was calculated by using the method suggested by Morel [22]. The use of a bellmouth is thought to help with the reduction of fouling, although practical application thereof is uncommon in heat exchangers.

The fourth type of inlet used, was a fully developed inlet which did not make use of the calming section. This inlet had an inner diameter similar to that of the test section. The length of the fully developed inlet was determined in terms of the suggestion by Durst *et al.* [23], which required a minimum length of 120-tube diameters. To ensure that this minimum length was met, an inlet of 160-tube diameters was chosen. In the case of adiabatic tests, cooling started after this inlet section while, for the other inlets, cooling started after the calming section.

All the insulated test sections (Fig. 4) were operated in a counterflow configuration and were manufactured from hard-drawn copper tubes. The total length of each test section was approximately 5 m. The tubes that were tested had a measured outside diameter of 15.88 mm and inner diameter of 14.482 mm. The inner diameter was measured with a micrometer and the average of several measurements was used, as the tube was not perfectly round.

The annulus mass flow was high, which ensured that the wall temperature of the inner tube remained relatively constant for most experiments. The annulus inner diameter of specifically 20.7 mm was chosen so that the space between the annulus and the inner tube would be small, ensuring high flow velocities and thus turbulent flow in the annulus. This small space further ensured that the annulus had a small thermal resistance compared to that of the inner tube. To prevent sagging and the outer tube touching the inner tube, a capillary tube was wound around the outer surface of the inner tube at a constant pitch of approximately 60º. This also promoted a rotational flow velocity inside the annulus, producing a higher heat transfer coefficient and thus a lower thermal resistance than that of the inner tube.

A full experimental uncertainty analysis was performed on the system, using the method suggested by Kline and McClintock [24]. Uncertainties for the calculated Nusselt numbers were less than 2%. For the friction factor, uncertainties were less than 12% for low Reynolds numbers (< 1 000), but less than 3% for Reynolds number of approximately 15 000.

DATA REDUCTION

The inner tube's average heat transfer coefficient was obtained by making use of the overall heat transfer coefficient and the sum of the resistances, calculated as follows:

$$\alpha_i = \frac{1}{A_i}\left[\frac{1}{UA} - R_w - \frac{1}{\alpha_o A_o}\right]^{-1}$$

(1)

The UA can be obtained by means of the overall heat transferred and the log-mean temperature difference, calculated from the inlet and outlet temperatures of the inner tube and annulus:

$$UA = \frac{\dot{Q}_i}{T_{lmtd}}$$

(2)

The heat transfer rate in the inner tube was used, as its uncertainties were much lower than that of the annulus and was calculated as:

$$\dot{Q}_i = \dot{m}_i C_p \Delta T$$

(3)

The specific heat values were obtained from IAPWS [25], which is based on the fluid temperature.

The annulus heat transfer coefficient was calculated by means of the annulus bulk temperature and the average of the inner-tube and outer-wall temperature measurements:

$$\alpha_o = \frac{\dot{Q}_i}{A_o \left(T_{bo} - T_{wo}\right)}$$

(4)

The bulk and average wall temperatures were obtained by making use of the trapezium rule to "integrate" across the whole length of the tube. Throughout the tests, the annulus flow rate was kept as high as possible, as this reduced the thermal resistance of the annulus, thereby reducing its influence in Eq. (1), and hence decreasing the equation's overall uncertainty. On

average, the annulus thermal resistance was only 6% of the value of the inner tube. All fluid properties were obtained from Wagner and Pruß [26]. Experimental data were only captured once an energy balance of less than 1% was achieved. At low inner-tube Reynolds numbers (< 6 000), this requirement was not met due to the high annulus flow rate and its uncertainty. However, tests were conducted as checks, by substantially lowering the annulus flow rate. These tests proved that the heat transfer error in the inner tube at low Reynolds numbers was indeed less than 1%.

The Darcy-Weisbach friction factors were determined by:

$$f = \frac{2\Delta p D_i}{\rho u^2 L_{\Delta p}} \qquad (5)$$

The bulk fluid properties used for the calculation of the Reynolds numbers, Prandtl numbers, etc., were calculated, not as the average between the water inlet and outlet temperatures, but estimated from:

$$T_i = \frac{\dot{Q}_i}{\alpha_i} + T_{wi} \qquad (6)$$

RESULTS

Adiabatic friction factor
In Fig. 5, the adiabatic friction factors are given for different types of inlets for laminar, transitional and turbulent flow. In the laminar flow regime, the friction factors are very close to the Poiseuille relation (*f = 64/Re*) and are not influenced by the type of inlet used. However, the type of inlet influences the critical Reynolds number. The re-entrant type of inlet has the lowest critical Reynolds number, followed by the fully developed, square-edged type and then the bellmouth type of inlet. In general, it shows that the "smoother" the type of inlet, the longer transition can be delayed, and that transition can be delayed much longer than the traditional critical Reynolds number of approximately 2 100 to 2 300.

Diabatic friction factor
During diabatic conditions, there are two additional factors which cause an increase in friction factors when compared to adiabatic flow. Firstly, the viscosity difference between the bulk of the fluid and the wall [27] which influences the wall shear stress, and secondly, secondary flow [28] which is caused by the temperature difference – also because of the temperature difference between the wall and bulk fluid.

The diabatic friction factors for the different types of inlets are shown in Fig. 6, with regard to the laminar, transitional and turbulent flow regimes., The Poiseuille equation for laminar adiabatic flow is also indicated in this figure, as well as the correlation of Filonenko [29] in the turbulent flow regime, which is also given by Lienhard and Lienhard [30] and more commonly used in the heat transfer correlation of Gnielinski [31].

In general, the friction factors for the laminar flow region were independent of the type of inlet used but, on average, they were 35% higher than predicted by the Poiseuille relation. Even with a viscosity correction, the prediction only improved by 4%. This increase in friction factor can be attributed to the secondary flow effects, with data from Nunner [32] showing similar results. Tam and Ghajar [28] also noted this increase and found that it was

dependent on the heating rate. This implies that, since the friction factor was proportional to the wall shear stress, which, in turn, was proportional to the velocity gradient at the wall, secondary flows distorted the velocity profile in such a way that the velocity gradient near the wall was much greater. This would then give rise to the higher friction factors. Many numerical and experimental studies that have been performed show this distortion [33-35].

Furthermore, the type of inlet does not affect the critical Reynolds number. For all the different types of inlets, the critical Reynolds number was approximately 2 100 to 2 300. Also, the type of inlet did not affect the friction factor in the turbulent flow regime. Turbulent flow results correlated fairly well with the viscosity ratio correction, although it would seem as if full turbulence is only reached at Reynolds numbers above 15 000, where the predictions of Filonenko [29] converge with the measurements. Unfortunately, the range of data was limited in such a way that this could not be confirmed by taking measurements at Reynolds numbers greater than 15 000.

Nusselt numbers
Fig. 7 shows the Nusselt numbers for a fully developed hydrodynamic boundary layer as inlet but with the thermal boundary layer still developing. This is typically what happens in chiller tubes, as fully developed thermal flow takes longer to fully developed than the length (5 m) of the test section that was used. It shows that, in the laminar flow regime, the Nusselt numbers varied between approximately 10 to 15, which are much higher than the theoretical predictions [10] of 3.66 for a constant wall temperature and 4.36 for a constant heat flux. This is attributed to the buoyancy-induced secondary flow in the tube which was a result of the difference in density at the centre and the wall of the tube. The correlations of Oliver [36] and that of Shome and Jensen [37], which were designed to incorporate these natural convection effects, predicted the laminar data from a Reynolds number of 1 000 to 2 100 to within 10% and 7.5% respectively.

It seems as if the transition occurs at a Reynolds number of about 2 500. However, pressure drop measurements as well as variation in temperature measurements by the same authors [16-21] on the same experimental set-up, indicated that transition starts at a Reynolds number of 2 100 and ends at approximately 3 000. The kink in the Nusselt numbers indicates the end of transition.

After transition, if the results are compared with the Colburn [38] correlation as modified by Sieder and Tate [27], all the turbulent regime data (Re > 3 000) are predicted on average to within 12%, although the equations are actually only valid for Reynolds numbers greater than 10 000. For Reynolds numbers greater than 5 000, the correlation predicted the data on average to within 1%, with root mean square deviations of 5%, thus validating the experimental set-up for turbulent flow.

Fig. 8 compares the Nusselt numbers for different types of inlets. It is apparent that the inlet geometry has no influence on the results. In fact, transition for all inlets occurs at the same Reynolds number, that is, it starts at 2 100 and ends at 3 000. The reason for this is that the secondary flows in the tube suppress the growth of the hydrodynamic boundary layer to such a degree that the fluid is fully developed, and hence transition occurs at the fully developed inlet's transition point. This might, however, only be unique to water or low Prandtl number fluids, as other researchers have found transition to be dependent on the inlet profile when using a water-glycol mixture together with heat transfer [12]. Furthermore, it is evident from

the data that transition from laminar to turbulent flow is not sudden, but a smooth transition between the two regimes.

In this instance, the inlet profile also has no influence in the laminar regime, again showing that the natural convection in the tube dominates the flow. Turbulent results further show that there is very little variation in the Nusselt numbers for the various inlet profiles, indicating that the results in this region are independent of the inlet.

CONCLUSIONS

More heat transfer and pressure drop data are needed, since modern heat exchanger equipment start operating in the transitional flow regime as the water mass flow through the tubes is being decreased. With the little data available on transition for the cooling of the fluid, designing for this regime is difficult, therefore necessitating the current research.

Single-phase smooth tube pressure drop and heat transfer measurements for water were conducted in horizontal circular smooth tubes. An experimental set-up consisting of a tube-in-tube counterflow heat exchanger with the cooling of water as the working fluid was used to obtain measurements within the transitional flow regime. Four different types of inlet geometries were used, i.e. hydrodynamical fully developed, square-edged, re-entrant and a bellmouth. Single-phase smooth tube heat transfer and friction factor results in the transition regime were presented. The effect of the different types of inlets was also investigated.

According to the adiabatic friction factor results, it was found that transition from laminar to turbulent flow was strongly dependent on the type of inlet used. The smoother the inlet, the more transition was delayed. Results for the bellmouth inlet showed the largest delay, with transition only occurring at a Reynolds number of approximately 7 000.

On the contrary, diabatic friction factor results showed that transition was independent of the type of inlet. Laminar friction factors were, however, much higher than the values predicted by the Poiseuille relation. This was also attributed to the natural convection flows influencing the boundary layer to such a degree that the shear stress at the tube wall was higher than normal.

Laminar heat transfer results were much higher than their theoretically predicted values due to the secondary flows increasing the amount of mixing in the tube. Furthermore, heat transfer measurements showed that transition with water was totally independent of the type of inlet used and that transition for all the different types of inlets occurred at the same Reynolds number. This was due to the buoyancy-induced secondary flows suppressing the inlet disturbance.

NOMENCLATURE

A	area, m^2
C_p	constant pressure specific heat, J/kg.K
D	diameter, m
f	friction factor (Darcy-Weisbach)
L	length, m
$\dot{m}$	mass flow rate, kg/s
p	pressure, Pa
u	average fluid velocity, m/s
$\dot{Q}$	overall heat transfer rate, W
R	thermal resistance, K/W
T	temperature, K
U	overall heat transfer coefficient, W/m^2.K

Greek Symbols

α	heat transfer coefficient, W/m^2.K
ρ	density, kg/m^3

Subscripts

b	*bulk*
i	inner tube, inside
lmtd	log-mean temperature difference
o	outside, annulus
Δp	pressure drop length
w	wall

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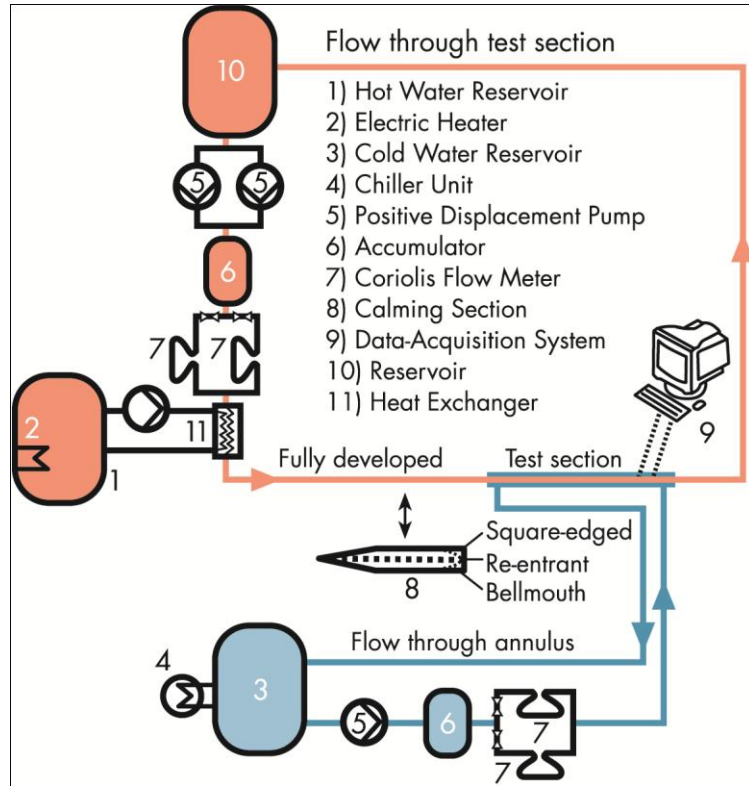
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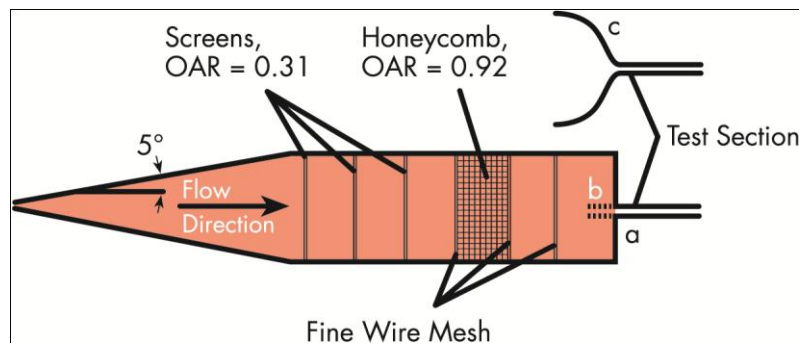Fig. 1:   Schematic layout of the experimental set-up

Fig. 2: Calming section with the different types of inlet configurations: a) square-edged, b) re-entrant and c) bellmouth
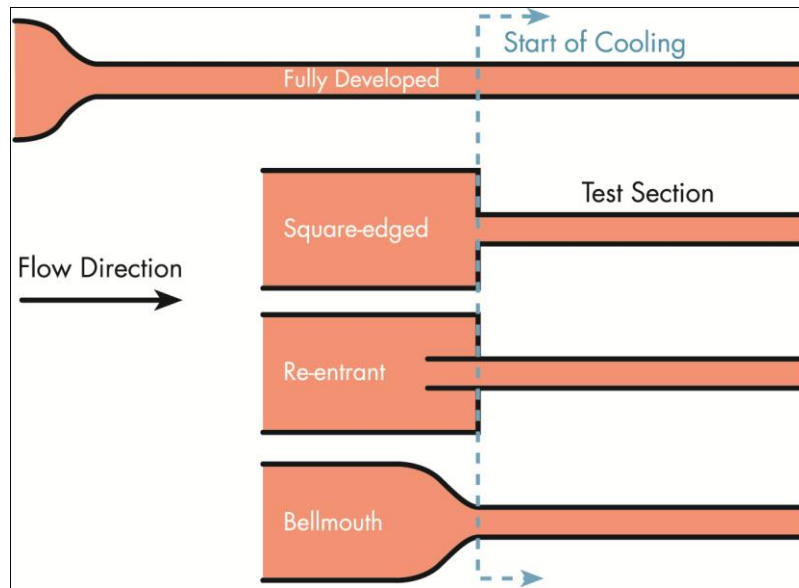
Fig. 3: Illustration of the different inlet geometries used together with the calming section shown in Fig. 2
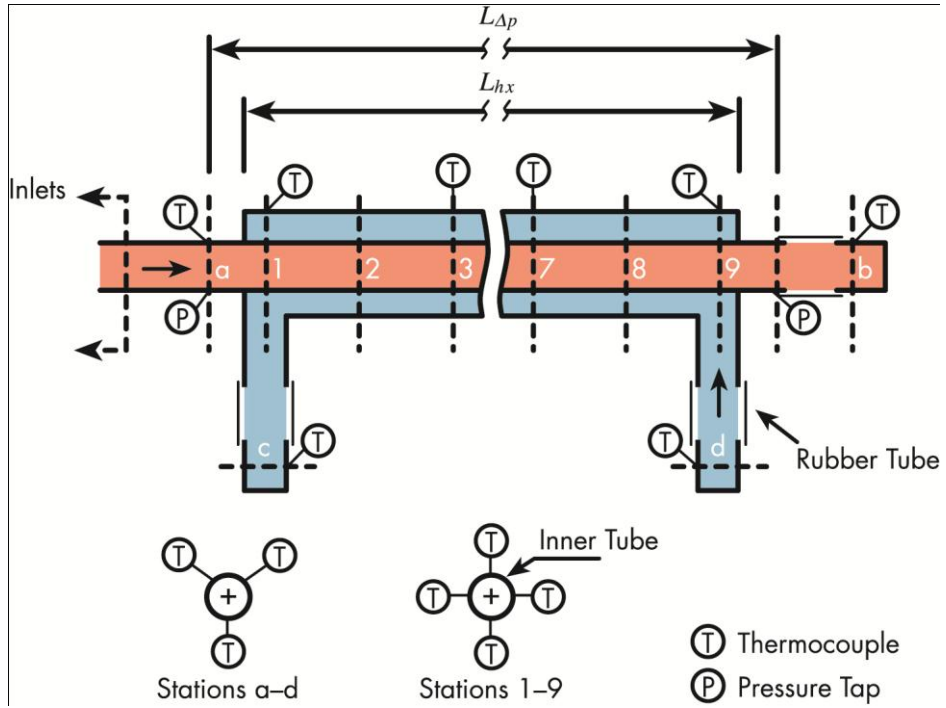
Fig. 4:   Schematic layout of the test section. The inlets refer to the different inlet geometries of Fig. 3.
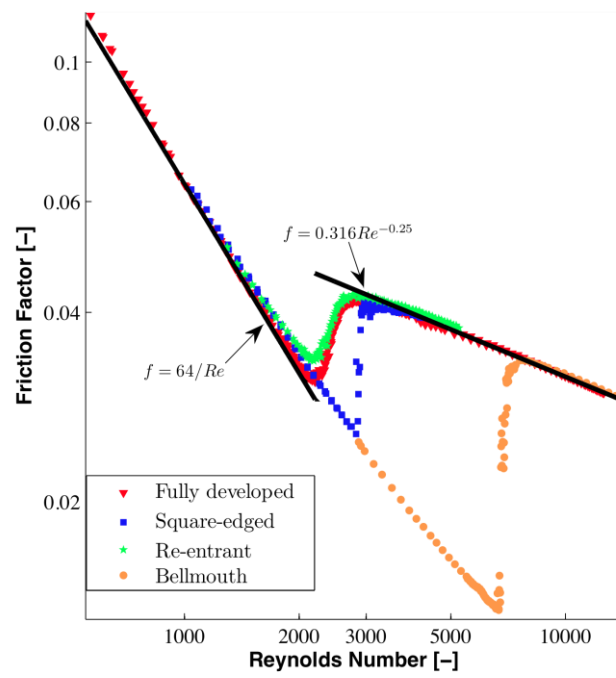
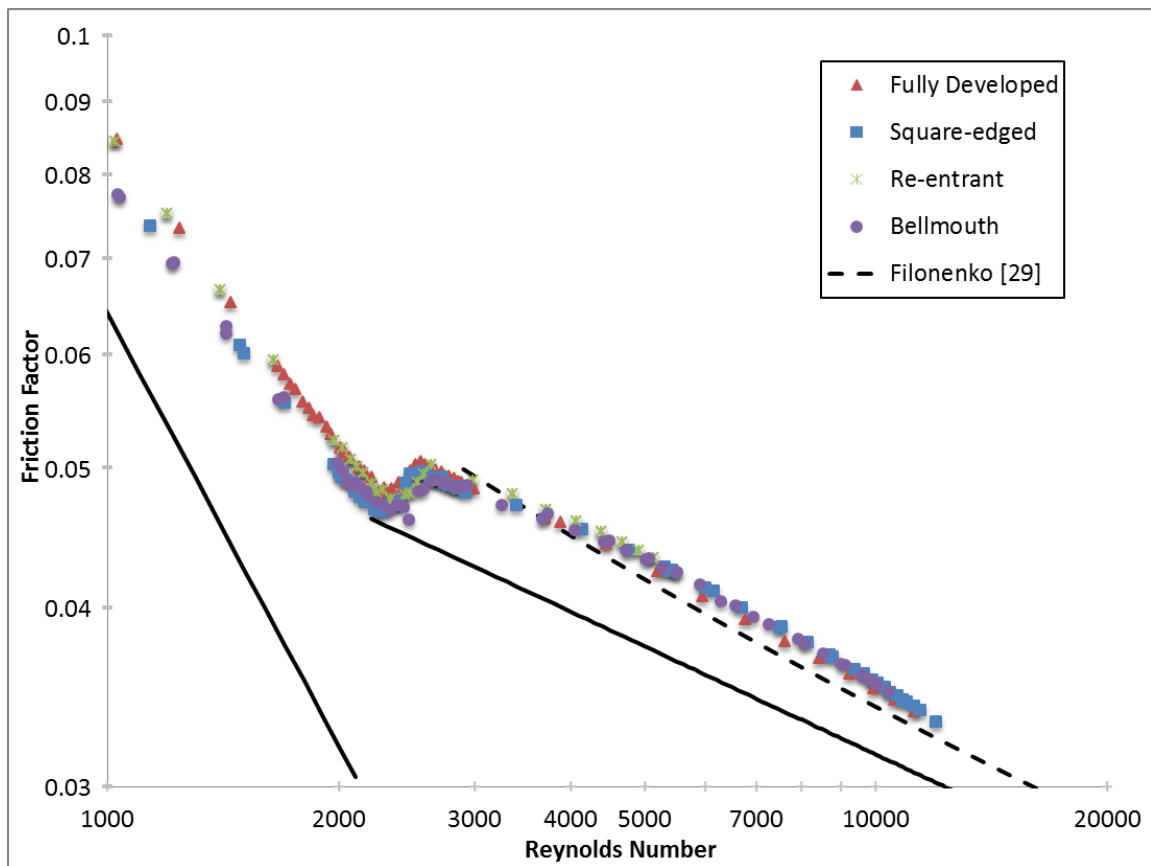Fig. 5:   Adiabatic friction factors for the smooth tube with various types of inlets

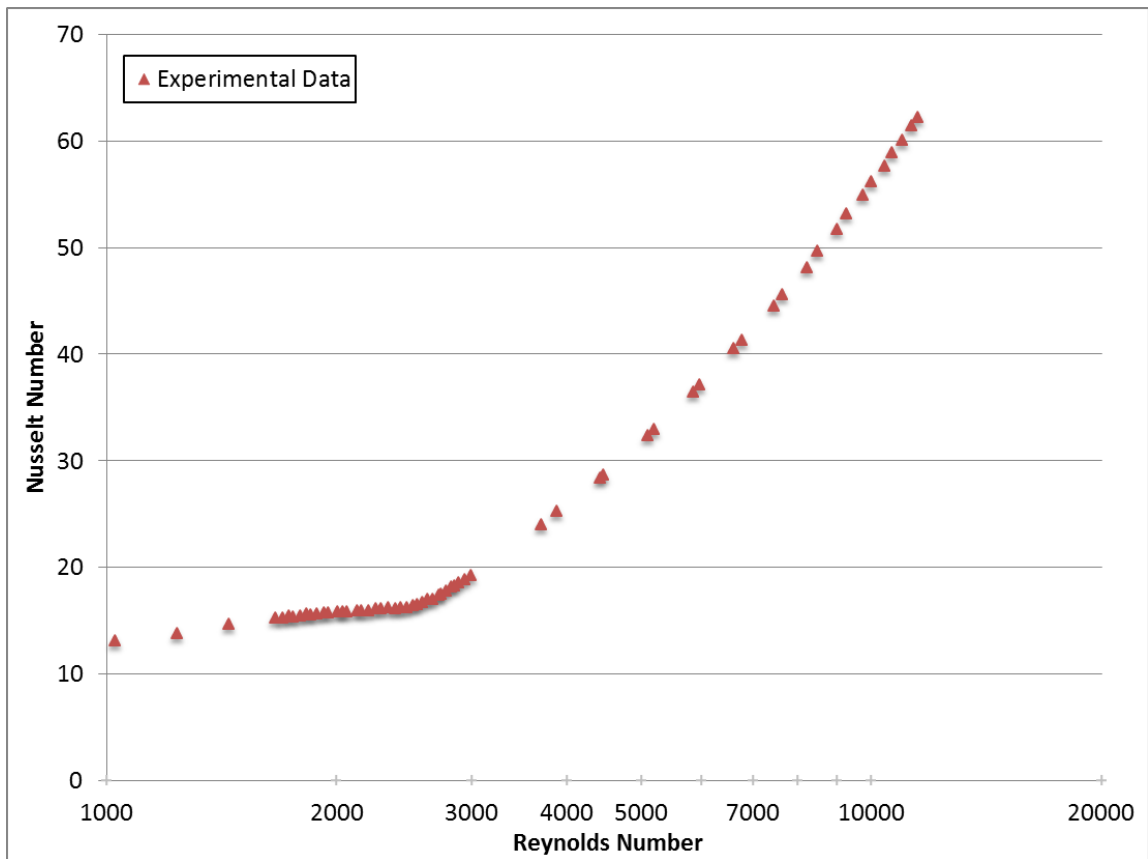Fig. 6: Diabatic friction factors for the smooth tube with various types of inlets

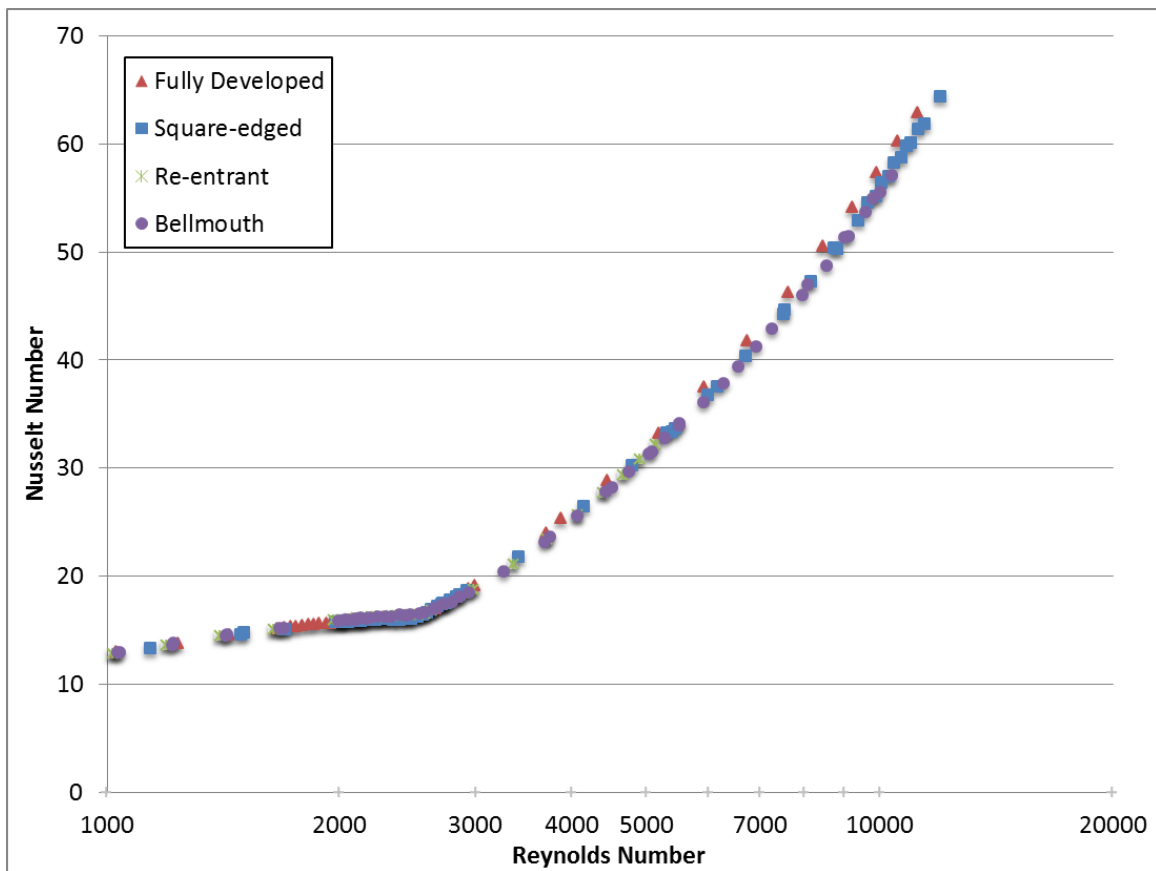Fig. 7:   Heat transfer results for the fully developed smooth tube

Fig. 8:   Heat transfer results for the smooth tube for various types of inlets

Josua Meyer obtained his BEng (cum laude) in 1984, MEng (cum laude) in 1986, and his PhD in 1988, all in Mechanical Engineering from the University of Pretoria. He is registered as a professional engineer. After his military service (1988-1989), he accepted a position as Associate Professor in the Department of Mechanical Engineering at the Potchefstroom University in 1990. He was Acting Head and Professor in Mechanical Engineering before accepting a position as Professor in the Department of Mechanical and Manufacturing Engineering at the Rand Afrikaans University in 1994. He was Chairman of Mechanical Engineering from 1999 until the end of June 2002, after which he was appointed Professor and Head of the Department of Mechanical and Aeronautical Engineering at the University of Pretoria from 1 July 2002. At present, he is the Chair of the School of Engineering. He specialises in heat transfer, fluid mechanics and thermodynamic aspects of heating, ventilation and air-conditioning. He is the author and co-author of more than 350 articles, conference papers and patents and has received various prestigious awards for his research. He is also a fellow or member of various professional institutes and societies such as the South African Institute for Mechanical Engineers, South African Institute for Refrigeration and Air-Conditioning, American Society for Mechanical Engineers, American Society for Air-Conditioning, Refrigeration and Air-Conditioning, and is regularly invited to be a keynote speaker at local and international conferences. He has also received various teaching and exceptional achiever awards. He is an associate editor of *Heat Transfer Engineering.*

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