HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF A HORIZONTAL ANNULAR PASSAGE IN THE TRANSITIONAL FLOW REGIME

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ABSTRACT

Due to tube enhancements being used to improve process efficiencies, heat exchangers are starting to operate in the transitional flow regime. Unfortunately, heat transfer and pressure drop performances in this flow regime is un-explored for many heat transfer geometries. In this preliminary study, experiments were conducted on the annular passage of a horizontal concentric counter-flow tube-in-tube heat exchanger operated with water for non-fully developed flow associated with a standard inlet geometry type. The annular diameter ratio, defined as the inner wall diameter over the outer wall diameter, was 0.386. An approximate uniform wall temperature on the inner annular wall surface was considered for varying annular mass flow rates that covered all flow regimes. Both heated and cooled cases of the annulus were examined. It was found that the heat transfer and pressure drop characteristics, based on the hydraulic diameter are different from those in circular tubes. The transition from laminar to turbulent flow, based on the Nusselt number, appeared to occur earlier than based on the friction factor. Nusselt numbers for heated annulus case, based on Reynolds number, were high than that for the cooled case. Conversely, the friction factors were higher for the cooled annulus case than for the heated case, while the adiabatic friction factors were the lowest.

INTRODUCTION

Heat exchanger design guidelines normally advise that designs should be done either for the laminar or for the turbulent flow regimes. However, design constraints and energy requirements have often lead to heat exchangers operating outside their design parameters. These parameters often involve the heat exchanger operating in the transitional flow regime [1]. Unfortunately, heat transfer and pressure drop performances in the transitional flow regime is un-explored for many heat transfer applications including that of annular passages of tube-in-tube heat exchangers, one of the most common heat exchanger types.

The ratio of the inner tube's outer diameter to the outer tube's inner diameter, known as annular diameter ratio, has been reported to have an influence on both heat transfer and friction factor characteristics in such a heat exchanger type [2].

NOMENCLATURE

A_s	[m^2]	Surface area
c_p	[J/kg.K]	Specific heat at constant pressure
D	[m]	Diameter
f	[-]	Friction factor
h	[W/m^{2}K]	Convection heat transfer coefficient
k	[W/m.K]	Thermal conductivity
L_{dp}	[m]	Pressure drop length
L_{hx}	[m]	Heat exchange length
$\dot{m}$	[kg/s]	Mass flow rate
n	[-]	Number of thermocouples
Nu	[-]	Nusselt number
Re	[-]	Reynolds number
Δp	[kPa]	Pressure drop
$\dot{Q}$	[W]	Heat transfer rate
T	[°C]	Temperature
V	[m/s]	Average velocity

Special characters

ρ	[kg/m^3]	Density
μ	[kg/ms]	Dynamic viscosity

Subscripts

0	Outer tube inner wall
1	Inner tube outer wall
b	Bulk property
h	Hydraulic
i	Inner tube
ii	Inner tube inlet
io	Inner tube outlet
iw	Inner wall
j	Index number
$LMTD$	Logarithmic mean temperature difference
o	Annular passage
oi	Annular passage inlet
oo	Annular passage outlet
x	Place holder

Analytical and numerical solutions for heat transfer and fluid behaviour for fully developed flow with pure forced convection heat transfer in the laminar flow regime have been available for many years, whereas those for turbulent flow are normally calculated from empirical equations based on experimental data. According to Gnielinski [3], correlations for heat transfer and friction factors in the turbulent flow regime are found to be inconsistent with each other. Some correlations for turbulent flow in annular passages are listed in [2] and [4].

Prinsloo *et al.* [4] investigated the heat transfer and pressure drop characteristics in the turbulent flow regime of the annuli of horizontal tube-in-tube heat exchangers. Among other findings, they observed that for the same inlet water temperature the heated annulus had larger Nusselt numbers, thus transferring more heat than the cooled annulus. However, a heated annulus had a smaller friction factor compared to a cooled one. This was partly ascribed to the influence of wall and bulk fluid temperatures and the associated fluid properties.

A convenient way to observe the transition between the laminar to turbulent flow regimes, is by considering a plot of either the Nusselt number or the friction factor against the Reynolds number. For flows without free convection influence, the onset of turbulence occurs at approximately Re = 2300 for circular smooth tubes with fully developed flow. Zhipeng [5] considered the transition region as metastable and complicated. By summarizing results of earlier research work and by using a fluid flow model which is valid for all flow regimes, Abraham *et al.* [6] proposed a friction factor and Reynolds correlation for the entire range of Reynolds numbers which smoothly bridged between the flow regimes.

Practically, most heat exchangers do not operate with a flow that is fully developed, and in some cases an influence of free convection heat transfer may exist, which in turn creates secondary flow. Lu and Wang [7] investigated experimentally the characteristics of a non-fully developed flow with secondary flow in a narrow annulus of hydraulic diameter and pressure drop length of 6.16 mm and 1410 mm, respectively. The results showed different characteristics from those of a fully developed flow and pure forced convection heat transfer. They realised that the flow characteristics can be related to the liquid temperature difference at the inlet and outlet of the annulus. The influences of temperature difference are significant in the laminar flow regime while none were observed in turbulent flow regime. The flow transition (based on the friction factor) from laminar to turbulent in their investigation occurred in a low Reynolds number range from 1100 to 1500, while fully turbulent convective heat transfer for heated water was achieved at a Reynolds number range from 800 to 1200. Lu and Wang [8] also carried out experiments to investigate heat transfer characteristics of water flow in a narrow annulus of hydraulic diameter and pressure drop length of 4.12 mm and 1500 mm, respectively. The results for convective heat transfer were similar to those in [7].

Many researchers have also investigated the flow and heat transfer characteristics in the transitional flow regime for micro channels. Jiang *et al.* [9] observed that friction factors in micro channels were smaller than that in conventional-sized channels. Peng *et al.* [10] found that the critical Reynolds number for transition from laminar to turbulent in micro channels to occurred at 200 – 700 and for fully developed flow at Reynolds numbers of 400 – 1500. Dirker *et al.* [11] investigated the effects of different types of inlets in rectangular microchannels. They found that the critical Reynolds number and the transitional behaviour in terms of heat transfer and friction factors were influenced significantly by the inlet types. Mala and Li [12] measured the transition at Reynolds number of 500 – 1500 in micro-tubes.

On the basis of the previous research, some of which is mentioned above, preliminary experiments were conducted to investigate the characteristics of friction factor and heat transfer for a heated and cooled annulus. Specific attention was given to a transitional regime. The experiments were carried out on non-fully developed flow with secondary flow.

EXPERIMENTAL SETUP

Figure 1, shows the schematic layout of the experimental facility that includes test section. The facility consisted of two closed loop water systems. The lay-out depicted is that of a heated annulus set-up. By switching the connectors at the test section inlets and outlets between the hot and cold loops, either cooled or heated annular cases could be investigated depending on the test requirements. Adiabatic cases could be considered by passing water from one flow loop through both the inner tube and annular passage of the test heat exchanger.

The hot water loop was supplied by a 1000 litre reservoir (item R1) fitted with a 36 KW electrical resistance heater. The hot water was circulated by a positive displacement pump, CB620 (item P1) with a delivery range of 0.5 – 3.87 kg/s. Since flow rates, much less than what the pump could handle were required, a bypass valve (item RV1) was utilized to assist control the flow rate of water. An accumulator (item A1) was installed next to the pump to arrest pulsations that were created by the pump. Flow rates were measured using a Coriolis flow meter with an effective range of 0 – 1.833 kg/s (item M1). To avoid loose particles settling in the test section, and thereby compromising results, a filter (item F1) was installed in the loop.

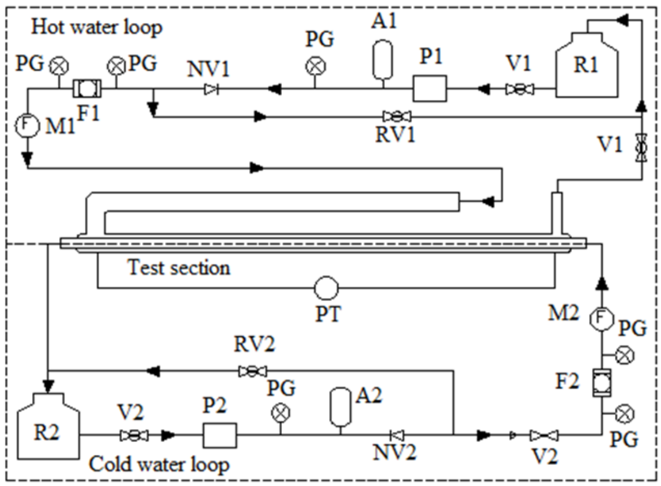

Figure 1 Schematic diagram of experimental facility.

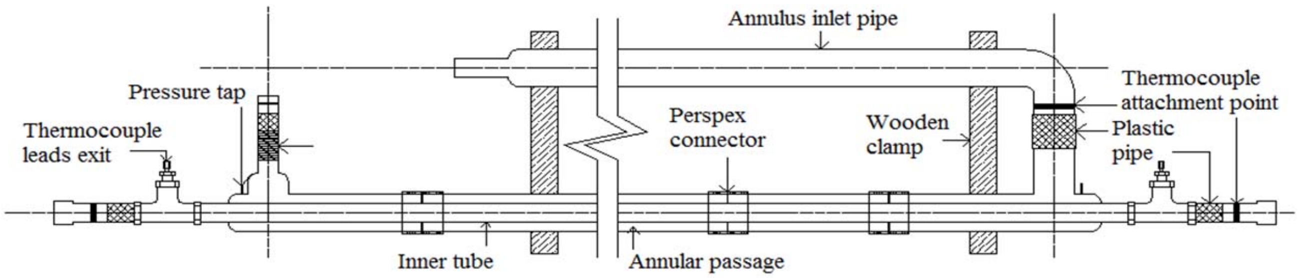

Figure 2 Tube-in-tube heat exchanger test section.

The cold water loop was very similar to the hot water loop. The cold water, however, had a 45 KW chiller unit connected to a 5000 litre reservoir (item R2). A SP4 pump (item P2) with a delivery range of 0.032 – 0.775 kg/s was utilized to circulate the water. A Coriolis flow meter (item M2) with a range of 0 – 0.604 kg/s was used to obtain the mass flow rate. Other components like an accumulator (item A2) and a filter (item F2) were also fitted in this loop.

In addition, both loops were fitted with relevant pressure relief valves, pressure gauges, non-return valves (items NV1 and NV2) and appropriate pipes and pipe fittings. Temperature, pressure drop and water flow rate readings were captured using National Instruments data acquisition system using Lab-view software.

The test section, represented in Figure 2, was a tube-in-tube heat exchanger with an annular diameter ratio of 0.386 and a length of 5.5 m. The tubes were made of hard drawn copper. The inner tube had inner and outer diameters of 11.18 mm and 12.7 mm, respectively while for the outer tube these diameters were 32.9 mm and 35 mm, respectively.

Special care was taken with the inlet of the annular passage. The annular inlet geometry was that of a 90° T-section fitting, similar to that found in most practical applications. The T-section was preceded by an adiabatic inlet length which ran parallel to the main test section length and which was connected to the heat exchanger by an elbow with a mid-pipe radius of 25 mm. The inlet length was 4.2 m long with an inner diameter of 32.9 mm and was clamped to the outer tube of the heat exchanger by three wooden clamps. It was designed to ensure repeatable flow inlet conditions at the inlet of the heat transfer section by producing fully developed hydrodynamic flow before the elbow.

To ensure concentricity of the annular passage, the inner tube was supported by hypodermic needles (0.8 mm in diameter) at eight equally spaced axial positions. Each support position had four equally spaced needles, held in place on the outer annular wall in thick-walled Perspex connectors. The outer tube was therefore made up of nine copper sections linked to each other via carefully manufactured Perpex connectors, such that the outer wall of the annulus was smooth and straight.

Inlet and outlet fluid temperatures for the inner tube were measured at adiabatic measuring stations each consisting of a short copper length equipped with four thermocouples connected 90° apart. Thermally, these measuring stations were insulated from the heat exchanger by means of rubber hoses.

The inlet and outlet temperatures of the annular passage fluid were measured in a similar manner as for the inner tube, except that each measuring station was equipped with eight thermocouples to reduce the effective measurement uncertainty to 0.0376°C. A mixing section was placed before the outlet temperature measuring station to avoid thick boundary layers and to ensure that the correct water temperature was captured.

In order to obtain heat transfer coefficients, the test section was equipped with a large number of inner wall thermocouple measuring positions. Two T-type thermocouples, each with measurement uncertainty of 0.106 °C, were inserted at each of the nine equally spaced stations along the length of the inner tube. In order to keep the annular passage, which was the focus of this study clear, thermocouple leads had to pass through inside of the inner tube. Each thermocouple junction was soldered in a groove, 10 mm long with a depth of 0.46 mm that was machined in the wall of the tube, such that the outer surface of the tube remained smooth. The inlet and outlet ends for the inner tube provided exit ports for thermocouple leads out of the inner tube.

To measure the local temperatures along the annular passage two thermocouples were attached on the surface of outer tube wall at intervals exactly midway between the inner tube measuring stations.

Pressure measuring ports (1 mm inner diameter) were installed near the inlet and outlet of annular passage such that the pressure drop length was 5 m. A with 0.86 kPa pressure transducer (item PT), shown in Figure 1, was utilized.

The entire set-up was thermally well insulated from the laboratory.

EXPERIMENTAL PROCEDURE

Firstly, calibration of all thermocouples and a pressure transducer was done. The pressure transducer was calibrated using a water column and a manometer with accuracy of 0.25%. Thermocouples were calibrated *in situ* using PT100 RTDs (Resistance Temperature Detector) with accuracy of 0.1 °C. Calibration curves were created with which measured data were conditioned during the data-processing stages.

During experimental test-runs, three different tests types were conducted with reference to the fluid in the annulus, namely: adiabatic, heated and cooled. In all cases, the water in the inner tube and in the annular passage flowed in opposite directions. Since the annular passage was the focus of this investigation, its flow was independent while the inner tube flow was depended on the annular flow. The annular flow ranged from 0.0068 kg/s to 0.3 kg/s (260 to 14720 Reynolds number), in order to ensure that all flow regimes were covered. The flow in the inner tube was such that the difference in temperature between the inlet and outlet was always approximately 1°C or lower. This was done in order to approximate a uniform wall temperature boundary condition on the inner tube surface. By increasing the inner tube flow rate this temperature difference could be reduced, but practical limitations prevented this. The inlet temperatures for hot and cold water were approximately 50°C and 20°C, respectively for diabatic test, while adiabatic test were conducted at a temperature of 25°C. Data was logged upon reaching a steady state condition. Steady state condition were deemed to have been reached when the change in energy balance between the inner tube and annular passage fluids were less than 0.1% and inlet and outlet temperature fluctuation of 0.1 °C or less was achieved, over a period of 1 minute. Up to 120 data points were collected for each data log. The average energy balance error was 1.9 % while the maximum energy balance error was 4%

VALIDATION OF TEST PROCEDURE

Prinsloo, *et al.* [4] did a portion of their investigation using a similar test section; therefore, the present experiment was validated against their results. For this purpose, six of their test cases where reproduced, the data analysed, and the results compared with their published results. The results for Prinsloo *et al.* and present investigations are shown in Figure 3. It can be observed that the calculated Nusselt numbers in this study compared well with those of Prinsloo *et al.* The slight difference of a maximum of 0.068% between the results could be due to measurement uncertainties.

PROCESSING OF RESULTS

Convection heat transfer rate for an existing system at a specified temperature difference is determined by Newton's law of cooling:

$$\dot{Q} = hA_s \Delta T_{LMTD} \tag{1}$$

Where h is the convection heat transfer coefficient, A_s is the surface area from which convection heat transfer takes place, and ΔT_{LMTD} is the logarithmic mean temperature difference.

Our interest is to get the convection heat transfer coefficient; therefore, the rest of the parameters including the heat transfer rate in equation (1) should be known or analysed first. The convection heat transfer rate between the water in annular passage and the inner wall can be found by:

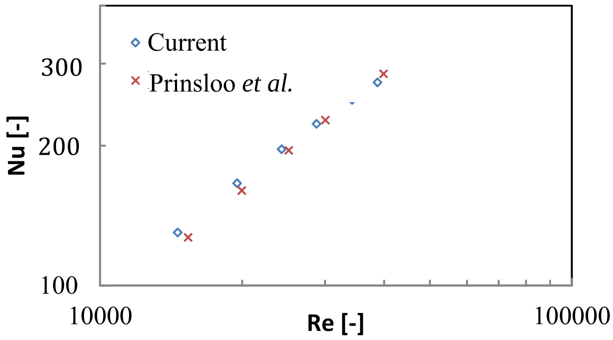

Figure 3 Comparison of heat transfer for two similar experiments.

$$\dot{Q}_o = \dot{m}_o c_p (T_{oi} - T_{oo}) \tag{2}$$

Here the mass flow rate was obtained from the reading of the relevant flow meter.

The surface area of the outside wall of the inner tube in (1) was calculated as:

$$A_s = \pi L_{hx} D_1 \tag{3}$$

By using equation (4), the average temperatures for the inner wall (x = "iw") , the water inlets (x = "ii" or "oi") and the water outlets (x = "io" or "oo") for both inner tube and annulus were calculated from the measurements from the relevant thermocouples.

$$\overline{T}_x = \left(\sum T_{x,j}\right)/n \tag{4}$$

Here n is the number of thermocouples per measuring station or thermocouple set. The logarithmic mean temperature difference for the annular passage can be calculated as:

$$T_{LMTD} = \frac{(\overline{T}_{iw} - T_{oi}) - (\overline{T}_{iw} - T_{oo})}{\ln[(\overline{T}_{iw} - T_{oi})/(\overline{T}_{iw} - T_{oo})]} \tag{5}$$

The mean dimensionless Nusselt number for annular passage was based on the hydraulic diameter and calculated as:

$$Nu = hD_h/k \tag{6}$$

Where the hydraulic diameter D_h of the annulus was calculated as:

$$D_h = D_0 - D_1 \tag{7}$$

Here D_0 and D_1 represent the outer and inner annular wall diameters respectively. The Reynolds number for flow in annular passage was calculated as:

$$\mathrm{Re}_o = \frac{\dot{m}_o D_h}{\mu_o A_o} \tag{8}$$

Based on the measured pressure drop the friction factor for annular flow was calculated by:

$$f = \frac{2 D_h \Delta p}{\rho L_{pd} V_o^2} \tag{9}$$

where the average velocity of water in the annulus was calculated as:

$$V_o = \dot{m} / (\rho_o A_o) \tag{10}$$

All water properties were calculated with the method of Popiel and Wojtkowiak [13] at the average bulk fluid temperature preliminary taken as the average between the measured inlet and outlet fluid temperatures of the relevant flow passage.

RESULTS AND DISCUSSIONS

As mentioned, an approximate uniform wall temperature on the inner annular surface was considered for varying annular mass flow rates that covered all flow regimes from laminar to turbulent. The average wall temperatures for the heated and cooled cases were 49°C ± 1°C and 20.2°C± 1°C, respectively. Friction factor characteristics were examined for adiabatic, heated and cooled cases, while heat transfer characteristics were, by definition, only considered for diabatic cases.

The friction factors for adiabatic, heated and cooled cases of the water in the annular passage are plotted with respect to the Reynolds number in Figure 4. For all three case types, similar behaviour was observed, namely that: friction factors rapidly decreased linearly in laminar flow regime and had a lower rate of decrease in transitional and turbulent flow regimes. In the laminar and transitional regimes the friction factors for the cooled case was higher than both the heated and adiabatic cases. The heating case followed the cooling case and the adiabatic case had the lowest friction factors. However, as the Reynolds number increased, the differences of friction factors for the three cases in turbulent flow regime got smaller. This phenomenon could be a result of secondary flow in the laminar and transitional regimes. In the Reynolds number range of 270 – 2700, the friction factors for the cooled case were approximately 2.02 times higher, than those of the adiabatic case and 1.37 times higher than for the heated case.

Also shown in Figure 4 are the perceived ranges of the transitional flow regime as identified by a change in the data point gradients (indicated visually for the cooled case). It was observed that transitional flow regime for the adiabatic case

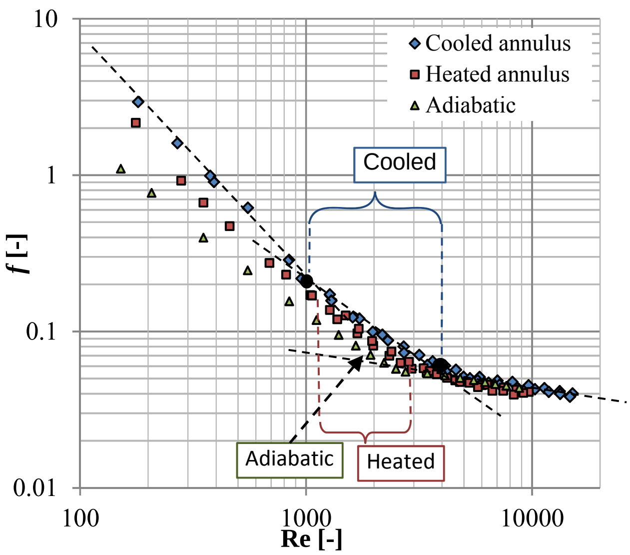

Figure 4 Friction factor characteristics for adiabatic, heated and cooled cases.

was relatively short and occurred at a Reynolds number of approximately 2000. For the heated case the transitional regime appeared to stretch from a Reynolds number of about 1200 to about 3000. For the cooled case, the transitional flow regime's Reynolds number range was found to be even wider at approximately 1000 to 4000.

Similarly, the average computed Nusselt numbers for heated and cooled cases are plotted with respect to the Reynolds number in Figure 5. For the two case types similar behaviour was observed, namely that: the Nusselt number increased rapidly in the laminar flow regime, then increased at a significantly lower rate in the transitional flow regime and increased again more rapidly in the turbulent flow regime. The Nusselt numbers for the heated case were on average 1.35 times higher than those of cooling case.

The transition from laminar to turbulent flow, based on the Nusselt number, started earlier than the transition based on the friction factor , similar to the observation of Lu and Wang [7], but lasted longer. For the heated case the transitional regime appeared to stretch from a Reynolds number of about 700 to about 3600. For the cooled case, the transitional flow regime Reynolds number range was found to be approximately 500 to 4500.

It is observed that the fully turbulent convective heat transfer for both the heated and cooled cases are achieved at higher Reynolds number than the 800 – 1200 range that Lu and Wang [7] found for their narrow annulus. The difference could be due to size of hydraulic diameter, annular diameter ratio, annulus inlet geometry, temperature difference, and the relative length of the test section. Table 1 summarizes some of the geometrical differences.

Further to this, it is unclear whether Lu and Wang controlled the hydrodynamic flow condition at the inlet of their test section, or precisely what their thermal condition was

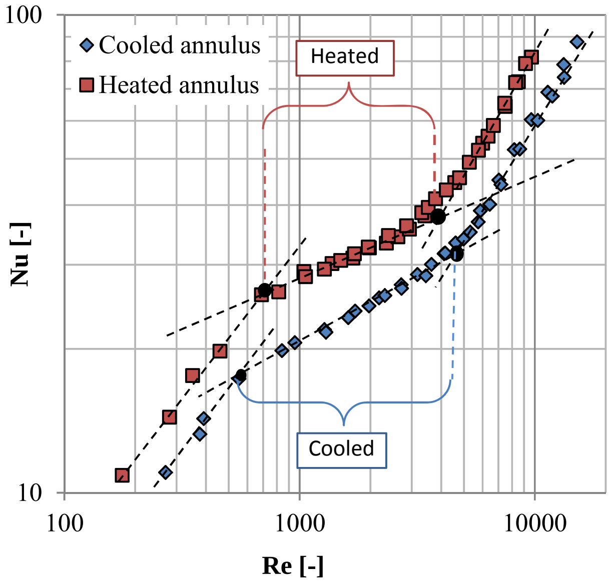

Figure 5 Heat transfer characteristics for heated and cooled cases.

Table 1 Differences between the current heat exchangers used by Lu and Wang and that of the current study.

	Lu and Wang	**Current study**
Hydraulic diameter	6.16 mm	20.2 mm
Annular diameter ratio	0.721	0.386
Pressure drop length	1.41 m	5 m

at the heat transfer wall. Since the annular space under consideration in this study was significantly larger than that of Lu and Wang, secondary flow pattern development may have been enhanced in our test cases.

The aspects mentioned above might indicate that results from transitional flow regime investigations may be geometry specific. Further investigations and data analyses are needed to be able to comment on this to a deeper extent. It would be interesting to consider the behaviour of the local heat transfer coefficients along the length of the test section within the transitional flow regime.

CONCULUSIONS

A preliminary experimental investigation was conducted on heat transfer and pressure drop characteristics in a horizontal annular passage. The heat transfer coefficients were higher for the heated case than for the cooled case, while the opposite was true for the friction factor. The friction factor for the adiabatic case was the lowest. The experimental results agreed in broad terms with data available in literature in the sense that the trends of both Nusselt number and friction factor graphs were similar. Also, the transition from laminar to turbulent based on the friction factor and Nusselt number occurred at different Reynolds numbers. The Nusselt number transition occurred earlier than friction factor transition. Further observations indicated that the Nusselt number transition was much longer when compared to that of a narrow annulus investigation by Lu and Wang [7, 8]. Further investigation is required.

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