AUGMENTED OF TURBULENT HEAT TRANSFER IN AN ANNULAR PIPE WITH ABRUPT EXPANSION

by

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This paper presents a study of heat transfer to turbulent air flow in the abrupt axisymmetric expansion of an annular pipe. The experimental investigations were performed in the Reynolds number range from 5000 to 30000, the heat flux varied from 1000 to 4000 W/m², and the expansion ratio was maintained at D/d=1, 1.25, 1.67, and 2. The sudden expansion was created by changing the inner diameter of the entrance pipe to an annular passage. The outer diameter of the inner pipe and the inner diameter of the outer pipe are 2.5 and 10 cm, respectively, where both of the pipes are subjected to uniform heat flux. The distribution of the surface temperature of the test pipe and the local Nusselt number are presented in this investigation. Due to sudden expansion in the cross-section of the annular pipe, a separation flow was created, which enhanced the heat transfer. The reduction of the surface temperature on the outer and inner pipes increased with the increase of the expansion ratio and the Reynolds number, and increased with the decrease of the heat flux to the annular pipe. The peak of the local Nusselt number was between 1.64 and 1.7 of the outer and inner pipes for Reynolds numbers varied from 5000 to 30000, and the increase of the local Nusselt number represented the augmentation of the heat transfer rate in the sudden expansion of the annular pipe. This research also showed a maximum heat transfer enhancement of 63-78% for the outer and inner pipes at an expansion ratio of D/d = 2 at a Re = 30000 and a heat flux of 4000 W/m².

Key words: *sudden expansion, separation flow, turbulent heat transfer, annular passage*

Introduction

Enhancements of the heat transfer in many practical applications are obtained by some modifications of design. The phenomenon of separation flow represents one of the most important applications to improve the heat transfer rate in thermal systems and occurs due to the change in the cross-section of the passage. The fluid flow in a sudden expansion or contraction and over backward or forward-facing steps are good examples of know separation and recirculation flow. Ota and Kon [1] adopted the effect of the separation flow on the heat

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transfer over a flat plate. The heat transfer to the air flow in the separation, re-attachment, and re-development regions over a flat plate was experimentally studied and Ota and Kon [1] observed the effect of separation flow on the enhancement of the heat transfer. Vogel and Eaton [2] conducted two-dimensional studies of combined heat transfer and fluid dynamic measurements down-stream of a backward-facing step. The results showed correlations between the heat transfer coefficient and the Reynolds number by using the local skin friction. During the last decades there are many experimental and numerical studies concerned thermal performance using different geometries and types of fluids. A good example for thermal performance could be found at fluid flow over single or double forward-facing step [3-6], over single-double backward-facing step [7-10], through the sudden expansion [11-13], and through the sudden contraction [14, 15]. The results obtained indicated that thermal performance increases of step height, Reynolds number, and volume fraction of nanofluids. With the development of high accuracy in devices, there have been experimental studies find the positions of the separation and re-attachment points for turbulent and laminar flows. Armaly *et al*. [16], Eaton *et al*.[17] and Adam and Johnston [18] used direction probes of a wall-flow for measuring the positions of the points while Lee and Mateescu [19] used multi element hot-film sensor arrays for measuring the separation and re-attachment lengths of flow over facing step. Smyth [20] measured local heat transfer rates of air flow in a sudden enlarged tube with a Reynolds number up to $5 \cdot 10^4$ and compared the experimental results with predicted results. The results showed that the Nusselt number decreased in the separation region and increased in the re-attachment region. Baughn *et al*. [21] have experimentally studied the air flow and heat transfer in a circular pipe with a sudden expansion at a constant heat flux. The experiments were conducted with Reynolds numbers between 5300 to 87000 and expansion ratios of 0.266 to 0.800. They noticed the increase of the Nusselt number with an increase of the expansion ratio and Reynolds number. The same trends were obtained previously by Ede *et al*. [22] for water. In contrast, Zemanick and Dougall [23] obtained different results for air flow at Re above 30000, especially at the peak of the Nusselt number. Said *et al*. [24] carried out a numerical study of the heat transfer to fluid flow in an abrupt expansion with Re at $4 \cdot 10^4$ to $5 \cdot 10^4$, an expansion ratio from 0.2 to 0.6 and a Prandtl number from 0.7 to 7. They observed that these boundary conditions have a significant effect on the mean time average Nusselt number with Prandtl number greater than unity. El-Shazly *et al*. [25] performed experimental studies on heat transfer to air flow in sudden expansion with Re from 7760 to 40084 and an expansion ratio of 0.32, 0.49, and 0.61. They found an improvement of the heat transfer with an increase of the expansion ratio and the Reynolds number. Dae *et al*. [26] conducted an experimental and numerical study of heat transfer to air flow in sudden expansion and contraction of a circular pipe for the Reynolds number range of 4300 to 44500 and an expansion ratio of 0.4. The maximum Nusselt number was obtained at a step height between 9 and 12 in the whole range of Re numbers, and the experimental results agreed with the numerical results. Zhang *et al*. [27] performed investigations of the heat transfer to supercritical CO_2 flow in a symmetric sudden expansion duct. They showed the increase of the Nusselt number with the increase of the heat flux and the Reynolds number. The authors also showed that the re-attachment length on the lower wall increased but decreased on the upper wall. Recently, to enhance the heat transfer rate, some researchers [28] used the techniques of sudden the expansion of channels with swirl generators, such as spiral springs of different pitches and propellers. They observed an enhancement of the heat transfer in sudden expansion, as reported by others.

Other investigations for annular channels with sudden expansion were conducted by Shuja and Habib [29]. They studied the heat transfer to separated flow in annular diffusers with expansion by changing the wall angle of an annular passage. An enhancement of local heat transfer coefficient was observed with the increase of the wall angle of annular diffusers. In addition, Togun *et al*. [12] experimentally studied the heat transfer to air flow in sudden expansion for a concentric annular pipe with a heated outer pipe heated, and they observed an increase of the local heat transfer coefficient by increase of the step height, the Reynolds number and the heat flux.

This paper experimentally studied the effect of turbulent separation air flow on the augmentation of heat transfer performance on inner and outer heated pipe of the test section. The boundary conditions are covered by Reynolds numbers from 5000 to 30000, expansion ratios from 1.25 to 2, and heat fluxes between 1000 and 4000 W/m^2. Experimentally generated new data will help in the design of high efficiency heat exchangers.

Experimental apparatus

General description of the apparatus

The experimental set-up consists of two parts, where the first part is an open air circuit and the other part is for measurement, as shown in figs. 1 and 2.

Open circuit

Figure 1 represents a diagram of the experimental set-up, consisting of a blower (B) for the delivery of variable velocity air by an inverter controlled blower, and an aluminum delivery pipe (P) that is 10 cm in diameter and 200 cm in length, which transfers air to a settling chamber that has honeycomb (H) for filtering air. A flow meter (F) is mounted on the delivery pipe at 15*D* from blower to measure mean velocity where that calm length considered achieving a fully developed flow. The entrance pipe (E) has a variable inner diameter (d = 10, 8, 6, 5 cm) and length (150 cm), and the outer pipe of the test section (O) is heated with uniform heat flux and has constant diameter (D = 10 cm) and length (100 cm).The inner pipe of test section has a 2.5 cm outer diameter (d_o) and a total length of 290 cm, which has two parts: the first is unheated (N1) (length 190 cm) and fixed at the center of settling chamber (S), and the second part is heated (N2) (length100 cm). The upstream pipe (E) is connected with the downstream pipe (O) by four flanges according to the diameters of the upstream pipe used in the experimental set-up. The inner pipe downstream region is subjected to a uniform heat flux. The two piping parts of the inner pipe are joined by a teflon flange to avoid

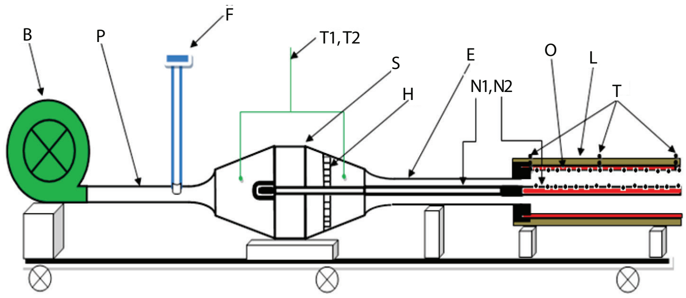

Figure 1. Schematic diagram of the experimental apparatus

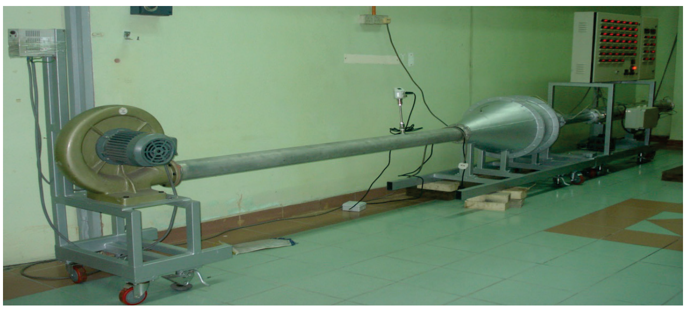

Figure 2. Photograph of test rig

conduction heat transfer to the upstream region as shown in fig. 3. Two layers of fiber glass (L; with a thickness of 20 mm) are applied to cover the outer pipe of the downstream section to reduce surrounding heat losses. There are two thermocouples (T1, T2) in the settling chamber, and three thermocouples(T3, T4, T5) are located at the end of the outer pipe of the test section to measure the inlet and outlet air bulk temperatures, respectively.

Method of heating

The heater rod (R) is approximately 1 meter long and its power (2 kW) supply is located at the center of the inner pipe of the test section. To avoid contact resistance and to ensure only the transfer of heat by conduction, the left over space of the inner pipe is filled with epoxy (M), which has high thermal conductivity, as shown in fig. 3. Twenty-three thermocouples of type (k) are used for measuring the surface temperatures of the inner pipe. The thermocouples are installed below the outer surface of inner pipe, as shown in fig. 3. The outer pipe of the test section is heated by a uniform coil heater made of nickel chrome. The 23 thermocouples located on the outer pipe were installed by making holes up to approximately 2 mm near the inner surface. Three thermocouples were located on the outer surface of the outer pipe, and three thermocouples were fixed on fiber glass (T) located at the inlet, the middle and the end of the outer pipe of the test section and the insulation fiber glass to measure the heat conduction losses to the surroundings, as shown in fig. 1. All thermocouples were connected by wires to the digital electronic display unit for data collection. The specifications of the digital temperature indicator are presented in tab. 1.

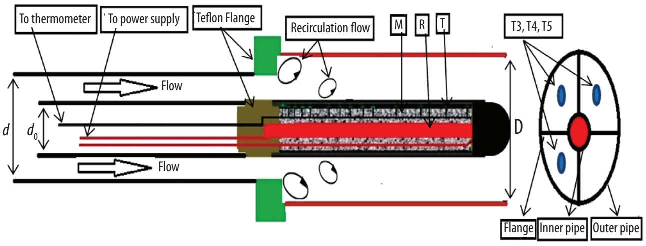

Figure 3. Schematic diagram of the test section

Table 1. Specifications of measuring system and apparatus

Digital temperature controller model: FY400/Power supply 85-265 V/AC/DC, dimension 29H×48W×95L mm,
Accuracy 0.2% Fs ± 1digital, frequency 50/60 HZ, input 0-20 mA
Heater of rod length 1 meter, 10 mm Dia, power 1 kW
Blower, model MB-15 HP1, 3 phase, 100 mm Dia, air volume (cmm/cfm) 21/745, static pressure (mm w.g.) 220, RPM 2950
Flow sensor model VA300-080 standard version for CS 2390 , 9 mm Dia, 100 mm length, accuracy 0.3%
Multi panel meter – Model: MT4W, dimension 96W×48H×100L mm, measured input DC voltage, DC current, AC voltage, AC current, power supply 100-240VAC
Thermocouple, type k (–200 °C to +1200 °C)

Electrical heating circuit

The electrical heating circuit consists of two variac to regulate the input current to the rod and coil heater and then provide a uniform and constant heat flux. A digital ammeter was used to measure the heater current and a digital voltmeter to measure the voltage of heater power supply. More details about the specifications of the measuring devices are shown in tab. 1.

Experimental procedure

The experimental procedure was as follows:
– a suitable entrance pipe diameter was selected to obtain the required expansion ratio,
– all the instruments were calibrated to being used for data tuning,
– the blower-discharged air at different velocities was obtained by turning a regulator connected to the blower,

– the variac was regulated to supply the required heating current to the heater,
– the steady-state condition was achieved after 2 hours in each experimental observation. However, at every half an hour increment from the beginning, the readings were taken to observe the rate of increase of temperature. The subsequent runs for other Reynolds numbers and height ranges were conducted by following the same procedure,
– during experiment two runs at the same condition were performed to provide data reproducibility. It was observed that the acquired data were correct within ± 2% accuracy and 95% confidence level, and
– for each test run, the following readings were recorded: the heater current in amperes, the heater voltage in volts, the readings of all thermocouples in °C, and the step height in mm.

Data reduction

The heat transfer from the test section can be calculated from the input power to the heater by Ohm's law:

$$\text{Power} = V \times I \tag{1}$$

The total heat generated (Q_t) = power,
Convection heat transfer (Q_c) to flowing air is represented by eq. (2):

$$Q_t = Q_C - Q_{loss}$$

where Q_{loss} is the heat losses from the outer pipe and the fiber glass to the surroundings.

The total losses due to conduction from the outer pipe and the fiber glass can be calculated by eqs. (3):

$$Q_{loss} = Q_{loss}\left(\text{Fiber glass}\right. + Q_{loss} \quad \text{outer pipe}$$

$$Q_{loss} = \left[\frac{\Delta T}{\frac{1}{2\pi KL}\ln\left(\frac{r_0}{r_i}\right)}\right]_{\text{fiber}} + \left[\frac{\Delta T}{\frac{1}{2\pi KL}\ln\left(\frac{r_0}{r_i}\right)}\right]_{\text{pipe}} \tag{3}$$

where $(r_o)_{\text{pipe}}$ is the outer radius of the outer pipe and $(r_o)_{\text{fiber}}$ is the outer radius of the fiber glass, $(r_i)_{\text{pipe}}$ is the inner radius of the outer pipe, and outer (r_i) is fiber inner radius of the fiber glass.

The convection heat flux is given by Holman [30]:

$$q_c = \frac{Q_c}{A_S} \tag{4}$$

where A_s is the surface area.

The local heat transfer coefficient is given by Holman [30]:

$$h_x = \frac{q_c}{T_{sx} - T_b} \tag{5}$$

where T_{sx} measures the local surface temperature. T_b is the bulk air temperature and is calculated by $T_b = (T_i + T_0)/2$ [30].

The Reynolds number based on the hydraulic diameter is given by Holman [30]:

$$\text{Re} = \frac{UD_h}{\nu} \tag{6}$$

The local Nusselt number based on the length of the test pipe is given by Holman [30]:

$$\mathrm{Nu_x} = \frac{h_x L}{K_f} \tag{7}$$

The ratio of the augmentation of heat transfer could be calculated by eq. (8)

$$\eta = \frac{\overline{\mathrm{Nu}}_\mathrm{S} - \overline{\mathrm{Nu}}_\mathrm{S=0}}{\overline{\mathrm{Nu}}_\mathrm{S}} \cdot 100\% \tag{8}$$

where $\overline{\mathrm{Nu}}_\mathrm{s}$ and $\overline{\mathrm{Nu}}_\mathrm{s=0}$ are defined as the average Nusselt number with a step and without a step, respectively.

Results and discussion

The experimental measurements in the present paper include both the surface temperatures for the inner and outer pipes with a heat flux from 1000 to 4000 W/m^2, a Reynolds number from 5000 to 30000, and an aspect ratio from 1.25 to 2.

Validation of the data

For validation the data, the results were compared at Reynolds number of 30000 with expansion ratio 2 and different heat fluxes with the experimental results of Togun *et al*. [12] at expansion ratio 1.8 and Reynolds number of 17050 where have similar behavior of the profile of surface Nusselt number as shown in figs. 4 and 5.

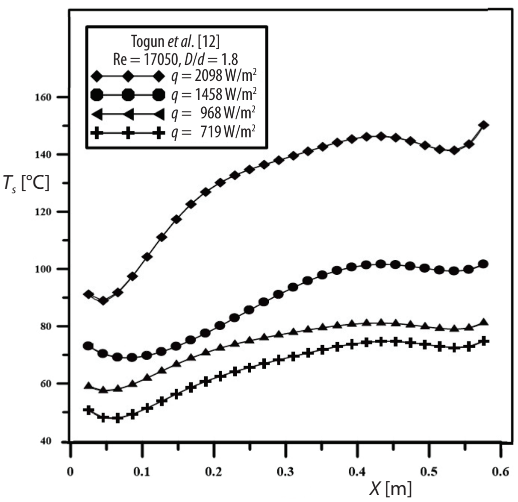

Figure 4. Variation of the surface temperature with axial distance, *D*/*d* = 1.8, Re = 17050

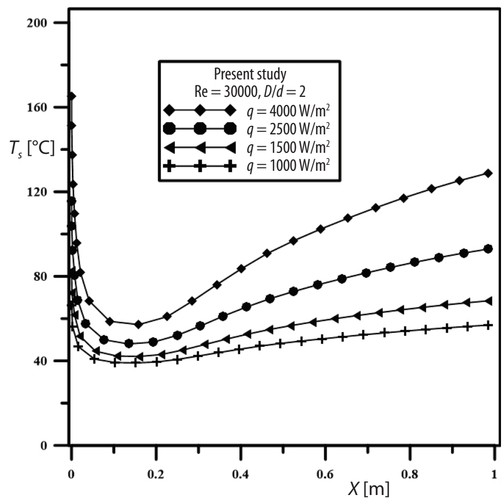

Figure 5. Variation of the surface temperature with axial distance, *D*/*d* = 2, Re = 30000

Uncertainty analysis of the test results

The uncertainty analysis of the measured data along with related parameters as found from the data reduction procedure is presented in tab. 2 and is assessed based on the error propagation method [31, 32].

Distribution of surface temperatures

The distribution of the surface temperature on the outer and inner pipes at a heat flux of q = 4000 W/m^2 and Re = 30000 for different expansion ratios are shown in fig. 6 and 7, respectively. The trends of the surface temperature are same on both the inner and outer pipes at an expansion ratio D/d = 1 (without step), but this behavior became different with the increase of the expansion ratio, where the surface temperature on the outer pipe decreases at the inlet region and then increases towards the exit. With an increase of the expansion ratio, the surface temperature on the inner pipe increases at the inlet region, then decreases suddenly and increases again at the end. The reduction of the surface temperature on the outer and inner pipes represents the re-circulation region, and the location of the re-attachment point at the inlet of the outer pipe contrasts with the location after inlet region for inner pipe. Generally, the effect of the expansion ratio on the distribution of the surface temperature appears clearly at the inlet region of the outer pipe and after the inlet region of the inner pipe, where the minimum of the surface temperature represents the re-attachment point and, before that, the re-circulation region, which causes an enhancement of the heat transfer due to turbulence enhancement. The increase of the re-circulation region results from the increase of the expansion ratio, which leads to an increase of the augmentation of the heat transfer.

Figures 8 and 9 show the distribution of the surface temperature for the outer and inner pipes at an expansion ratio of D/d = 2 and a Reynolds number of Re = 30000 for different heat fluxes. The results showed an increase in the surface temperature of the inner and outer pipes with an increase of the heat flux at the same Reynolds number and expansion ratio. In addition, figs. 10 and 11 show the effect of the Reynolds number on the surface temperature of the test section, where the decrease of the Reynolds number leads to an increase of the surface temperature at the heat flux q = 4000 W/m^2 and the expansion ratio D/d = 2.

Table 2. Uncertainty of variables

Variable name	Uncertainty error
$\overline{\mathrm{Nu}}$	±5%
$\mathrm{Nu_x}$	±6%
Re	±6%
q	±8%

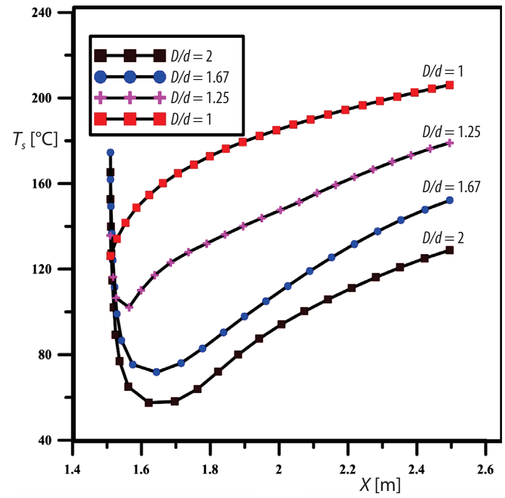

Figure 6. Variation of the surface temperature with axial distance for outer pipe, q = 4000 W/m^2, Re = 30000

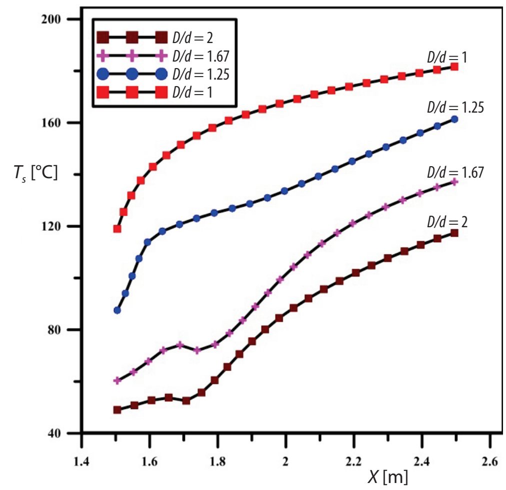

Figure 7. Variation of the surface temperature with axial distance for inner pipe, q = 4000 W/m^2, Re = 30000

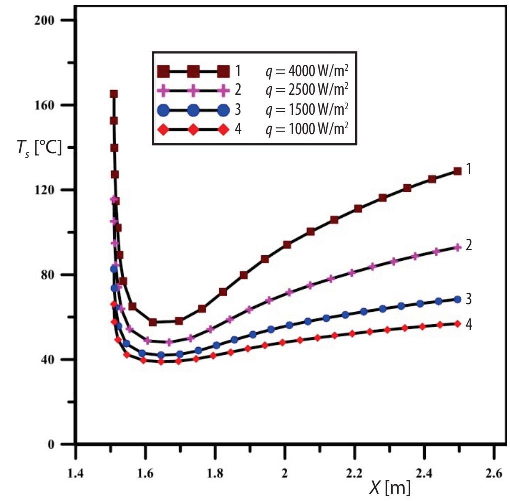

Figure 8. Variation of the surface temperature with axial distance for outer pipe, Re = 30000 and D/d = 2

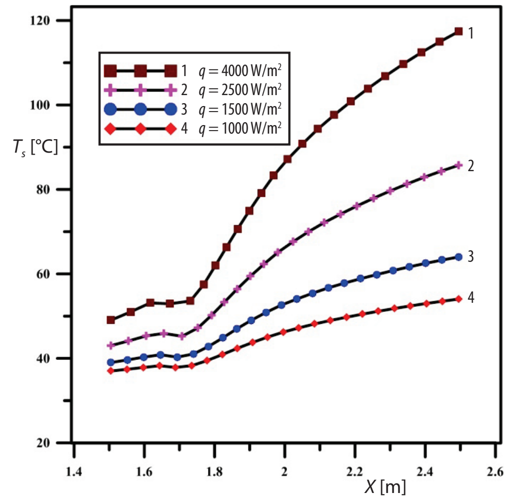

Figure 9. Variation of the surface temperature with axial distance for inner pipe, Re = 30000 and D/d = 2

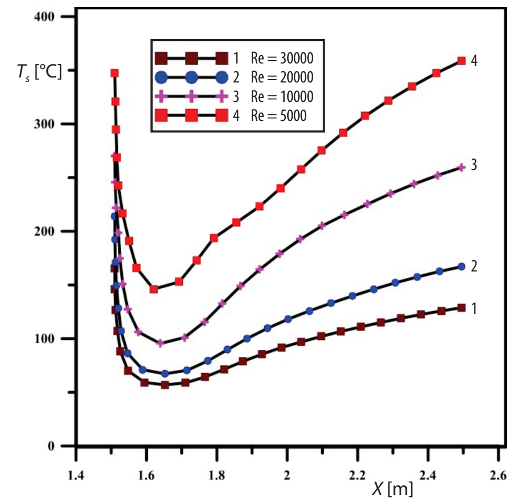

Figure 10. Variation of the surface temperature with axial distance for outer pipe, q = 4000 W/m^2 and D/d = 2

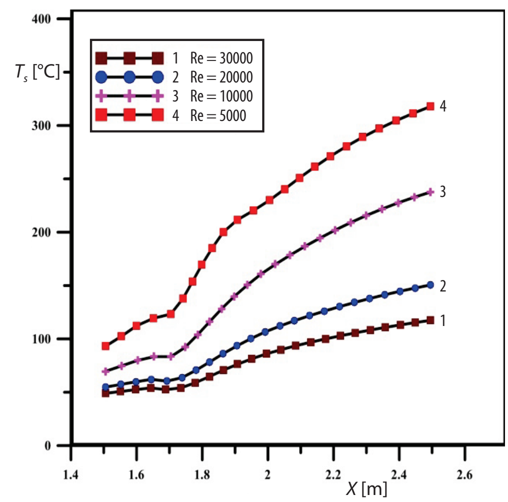

Figure 11. Variation of the surface temperature with axial distance for inner pipe, q = 4000 W/m^2 and D/d = 2

Local Nusselt number in abrupt expansion

The experimental data were reduced by equations (1)-(8) for expansion ratios D/d = 1, 1.25, 1.67, 2, heat fluxes q = 1000, 1500. 2500, and 4000 W/m^2, and Reynolds numbers of Re = 5000, 10000, 20000, and 30000. The local heat transfer coefficient was calculated by using eq. 5 to determine the local Nusselt number from eq. 7.

The variation of local Nusselt number (Nu_x) on the outer and inner pipes at an axial distance (X) for heat flux q = 4000 W/m^2 and an expansion ratio D/d = 2 are plotted in figs. 12 and 13, respectively. The local Nusselt number for the outer pipe increases with an increase of the Reynolds number at the inlet region to reach the maximum value at axial distance 1.64 m and then decreases towards the end. In contrast, the local Nusselt number for inner pipe decrease at the inlet region and then increases up to 1.7 m in axial distance then again decreases up to the exit. The obtained increment of the Nusselt number for the outer and inner pipes represents the enhancement of the heat transfer rate.

Figures 14 and 15 show the effect of the expansion ratio on the local Nusselt number for the outer and inner pipes at a heat flux q = 4000 W/m^2 and a Re = 30000. The augmentation of the heat transfer is observed with the increase of the expansion ratio for both the outer and inner pipe; thus, the local Nusselt number is enhanced with the increase of the expansion ratio due to recirculation flow.

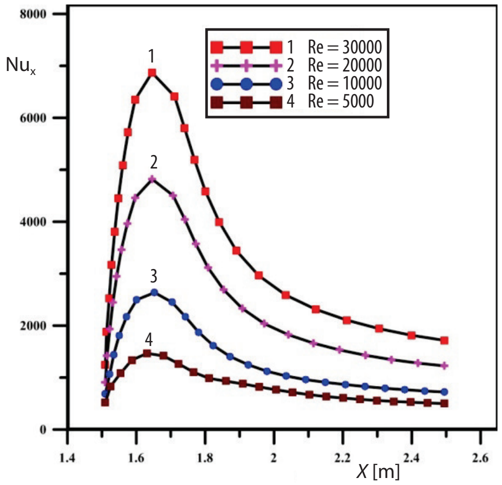

Figure 12. Variation of the local Nusselt number

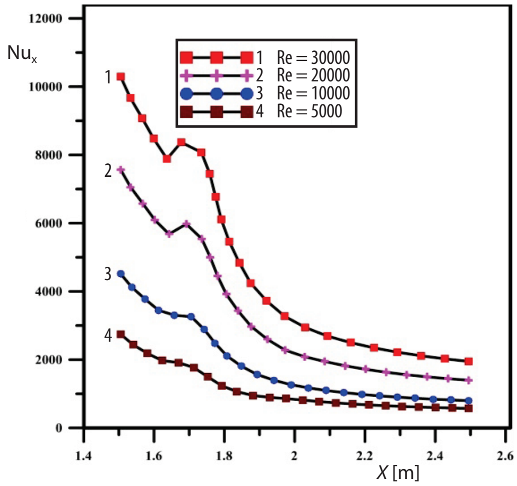

Figure 13. Variation of the local Nusselt number for outer pipe at q = 4000 W/m^2, D/d = 2 for inner pipe at q = 4000 W/m^2, D/d = 2

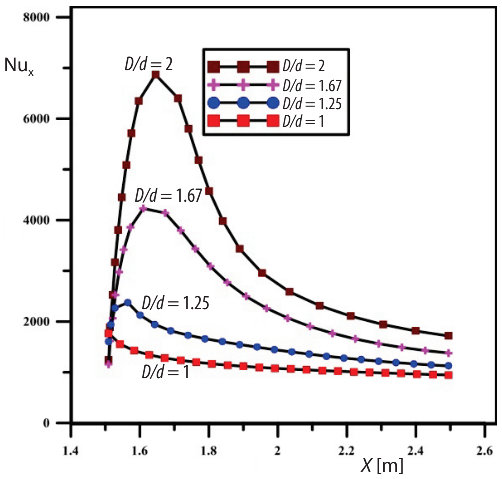

Figure 14. Variation of the local Nusselt number

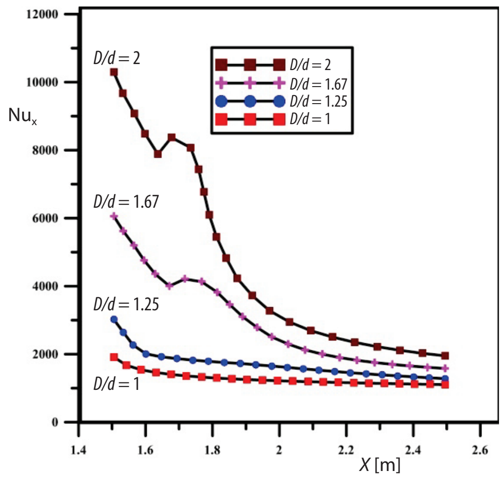

Figure 15. Variation of the local Nusselt number for outer pipe at q = 4000 W/m^2, Re = 30000 for inner pipe at q = 4000 W/m^2, Re = 30000

Performance evaluation

The average Nusselt number with Reynolds numbers at different expansion ratios and a heat flux of 4000 W/m^2 are presented in figure 16 and 17 for the inner and outer pipes, respectively. Increase of average Nusselt number observed for both inner and outer pipe with increase of Reynolds number and expansion ratio. Moreover, when both Reynolds number and expansion ratio increases the recirculation region increased hence increment in Nusselt number. The highest average Nusselt number was obtained at expansion ratio $D/d = 2$ and a Reynolds number of 30000 for the inner and outer pipes. The obtained augmentation of heat transfer was defined by a ratio of ($\overline{\mathrm{Nu}}_\mathrm{S} - \overline{\mathrm{Nu}}_{\mathrm{S}=0}$) to ($\overline{\mathrm{Nu}}_\mathrm{S}$) varied from 4-63% , and 11-78% for outer and inner pipes, respectively.

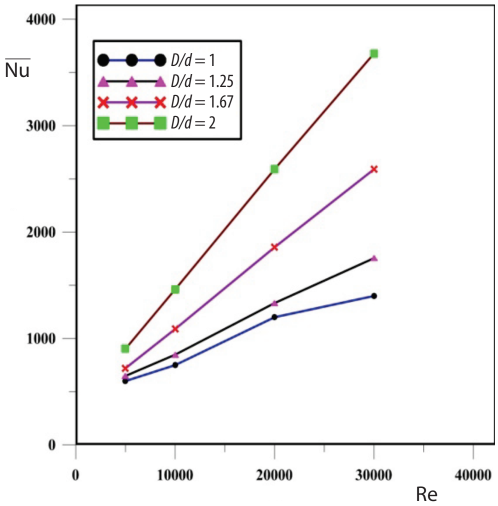

Figure 16. Variation of the average Nusselt number with Re at different expansion ratios and q = 4000 W/m^2 (outer pipe)

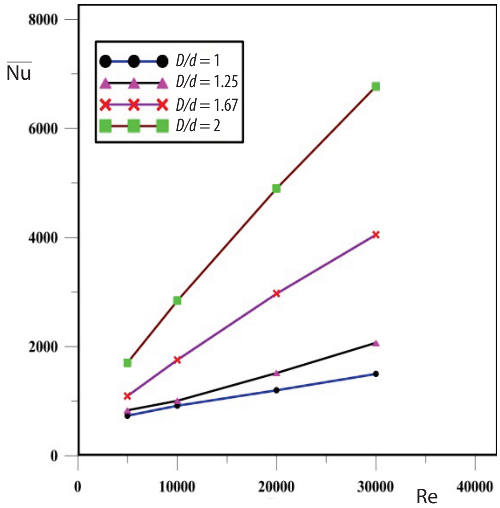

Figure 17. Variation of the average Nusselt number with Re at different expansion ratios and q = 4000 W/m^2 (inner pipe)

Conclusions

Heat transfer to turbulent air flow in the abrupt axisymmetric expansion of an annular pipe was experimentally studied. From the experimental results, the following conclusions can be written.

- The surface temperature for outer pipe is decreases at the entrance region due to recirculation flow generated by sudden expansion.
- The surface temperature for inner pipe is increases at the entrance region then decreases suddenly for the secondary recirculation flow and increases again at the end.
- The surface temperatures on the outer and inner pipes have same trend after Recirculation region.
- Increase of heat flux and decease of Reynolds number lead to increases of surface temperature on the outer and inner pipes.
- Nusselt number for both the inner and outer pipes increases with increased the expansion ratio for the inner and outer pipes of the test section.
- The increase of the local Nusselt number represented the enhancement of the heat transfer rate in the sudden expansion of the annular pipe.

Acknowledgement

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Nomenclature

- D – inner diameter of outer pipe of test section, [cm]
- d – inner diameter of entrance pipe, [cm]
- do – outer diameter of inner pipe for the whole annular passage, [cm]
- E – expansion ratio (= D/d), [–]
- H – heat transfer coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
- I – heater current, [A]
- K – thermal conductivity, [$\mathrm{Wm^{-1}K^{-1}}$]
- L – length of test section, [cm]
- Nu – Nusselt number, [–]
- $\overline{\mathrm{Nu}}$ – average Nusselt number, [–]
- Nu_x – local Nusselt number, [–]
- Re – Reynolds number, [–]
- T_b – average bulk air temperature, [K]
- T_{sx} – local surface temperature, [K]
- V – heater voltage, [V]

References

[1] Ota, T., Kon, N., Heat Transfer in the Separated and Reattached Flow over Blunt Flat Plates – Effects of Nose Shape, *International Journal of Heat and Mass Transfer*, *22* (1979), 2, pp. 197-206
[2] Vogel, J. C., Eaton, J. K., Combined Heat-Transfer and Fluid Dynamic Measurements Downstream of a Backward-Facing Step, *Journal of Heat Transfer Transactions of the ASMA*, *107* (1985), 4, pp. 922-929
[3] Safaei, M. R., *et al.*, Investigation of Heat Transfer Enhancement in a Forward-Facing Contracting Channel Using FMWCNT Nanofluids, *Numerical Heat Transfer*, *Part A, 66* (2014), 12, pp. 1-20
[4] Abdulrazzaq, T., *et al*., Heat Transfer and Turbulent Fluid Flow over Vertical Double Forward-Facing Step, *International Journal of Mechanical, Aerospace, Industrial and Mechatronics Engineering*, *8* (2014), 2, pp. 382-386
[5] Togun, H., *et al*., CFD Simulation of Heat Transfer and Turbulent Fluid Flow over a Double Forward-Facing Step, *Mathematical Problems in Engineering, 2013* (2013), ID895374
[6] Togun, H., *et al*., A Review of Experimental Study of Turbulent Heat Transfer in Separated Flow, *Australian Journal of Basic and Applied Sciences*, *5* (2011), 10, pp. 489-505
[7] Togun, H., *et al*., Heat Transfer to Laminar Flow over a Double Backward-Facing Step, *International Journal of Mechanical, Aerospace, Industrial and Mechatronics Engineering, 7* (2013), 8, pp. 117-122
[8] Togun, H., *et al*., Numerical Simulation of Laminar to Turbulent Nanofluid Flow and Heat Transfer over a Backward-Facing Step, *Applied Mathematics and Computation*, *239* (2014), July, pp. 153-170
[9] Togun, H., *et al*., Numerical Study of Heat Transfer and Laminar Flow over a Backward Facing Step with and without Obstacle, *International Journal of Mechanical, Aerospace, Industrial and Mechatronics Engineering, 8* (2014), 2, pp. 128-132
[10] Togun, H., *et al*., Numerical Study of Turbulent Heat Transfer in Separated Flow: Review, *International Review of Mechanical Engineering*, *7* (2013), 2, pp. 337-349
[11] Oon, C. S., *et al*., Computational Simulation of Heat Transfer to Separation Fluid Flow in an Annular Passage, *International Communications in Heat and Mass Transfer*, *46* (2013), Aug., pp. 92-96
[12] Togun, H., *et al*., An Experimental Study of Heat Transfer to Turbulent Separation Fluid Flow in an Annular Passage, *International Journal of Heat and Mass Transfer*, *54* (2011), 4, pp. 766-773
[13] Oon, C. S., *et al*., Numerical Simulation of Heat Transfer to Separation Air Flow in an Annular Pipe, *International Communications in Heat and Mass Transfer*, *39* (2012), 8, pp. 1176-1180
[14] Togun, H., *et al*., Numerical Study of Turbulent Heat Transfer in Annular Pipe with Sudden Contraction, *Applied Mechanics and Materials*, *465-466* (2014), Jan., pp. 461-466
[15] Orselli, R. M., De Lemos, M. J. S., Sudden Contraction in a Turbulent with a Porous Insert, *Latin American Journal of Solids and Structures*, *2* (2005), 3, pp. 269-290
[16] Armaly, B. F., *et al*., Experimental and Theoretical Investigation of Backward-Facing Step Flow, *Journal of Fluid Mechanics*, *127* (1983), Feb., pp. 473-496
[17] Eaton, J. K. *et al*., A Wall-Flow-Direction Probe for Use in Separating and Reattaching Flows, *ASME Journal of Fluids Engineering, 101* (1979), 3, pp. 364-366

[18] Adams, E. W., Johnston, J. P., Effects of the Separating Shear Layer on the Reattachment Flow Structure Part 2: Reattachment Length and Wall Shear Stress, *Experiments in Fluids*, *6* (1988), 7, pp. 493-499
[19] Lee, T., Mateescu, D., Experimental and Numerical Investigation of 2-D Backward-Facing Step Flow, *Journal of Fluids and Structures*, *12* (1998), 6, pp. 703-716
[20] Smyth, R., Heat Transfer in Turbulent Separated Flow, *Journal of Nuclear Science and Technology*, *11* (1974), 12, pp. 545-553
[21] Baughn, J. W. *et al.*, Local Heat Transfer Downstream of an Abrupt Expansion in a Circular Channel with Constant Wall Heat Flux, *Journal of Heat Transfer*, *106* (1984), 4, pp. 789-796
[22] Ede, A. J. Morris, C. I. H. R., Effect of the Local Heat Transfer Coefficient in a Pipe of an Abrupt Disturbance of the Fluid Flow: Abrupt Convergence and Divergence of Diameter Ratio 2/1, *Proc. Inst. Mech. Engrs.*, *170* (1956), 1, pp. 1113-1130
[23] Zemanick, P. P., Local Heat Transfer Downstream of Abrupt Circular Channel Expansion, *Journal of Heat Transfer*, *92* (1970), 1, pp. 53-60
[24] Said, S. A. M., *et al.*, Heat Transfer to Pulsating Turbulent Flow in an Abrupt Pipe Expansion, *International Journal of Numerical Methods for Heat & Fluid Flow*, *13* (2002), 3, pp. 341-364
[25] El-Shazly, K. M., *et al.*, Heat Transfer Characteristics of Pulsated Flow Downstream of Abrupt Expansion through Pipes, *Proceedings, 2nd International Conference on Advanced in Engineering Science and Technologies,* Phuket, Thailand, 2005, pp. 12-14
[26] Lee, D., *et al.*, Experimental and Numerical Study of Heat Transfer Downstream of an Axisymmetric Abrupt Expansion and in a Cavity of a Circular Tube, *J Mech Sci Technol*, *25* (2011), 2, pp. 395-401
[27] Zhang, X.-R., *et al.*, Bifurcation Phenomenon for Forced Convection of Supercritical CO_2 Sudden Expansion Flow and Heat Transfer in Symmetric Regime, *International Journal of Heat and Mass Transfer*, *53* (2010), 21-22, pp. 4467-4473
[28] Zohir, A. E., *et al.*, Heat Transfer Characteristics in a Sudden Expansion Pipe Equipped with Swirl Generators, *International Journal of Heat and Fluid Flow*, *32* (2011), 1, pp. 352-361
[29] Shuja, S. Z., Habib, M. A., Fluid Flow and Heat Transfer Characteristics in Axisymmetric Annular Diffusers, *Computers & Fluids*, *25* (1996), 2, pp. 133-150
[30] Holman, J. P., *Heat Transfer,* 9th ed., McGraw Hill, New York, USA, 2002
[31] Taylor, J. R., *An Introduction to Error Analysis: The Study of Uncertainties in Physical Measurements,* 2ed ed., University Science Books, Hill Valley, Cal, USA, 1997
[32] Kumaresan, V., *et al.*, Convective Heat Transfer Characteristics of CNT Nanofluids in a Tubular Heat Exchanger of Various Lengths for Energy Efficient Cooling/Heating System, *International Journal of Heat and Mass Transfer*, *60* (2013), May, pp. 413-421

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