AN EXPERIMENTAL STUDY ON HEAT TRANSFER AND PRESSURE DROP OF AIR-SOLIDS TURBULENT FLOW INSIDE A HORIZONTAL TUBE WITH DIFFERENT ENTRANCE ANGLES

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ABSTRACT

The present work deals with the effect of turbulent air-sand suspension flow on heat transfer and pressure drop in a horizontal tube heated with a uniform heat flux at different entrance angles. Four different inlet bend-angles (θ=30, 60, 80 and 90 deg.) in addition to the case of plain tube (θ=0 deg.) were studied at different Reynolds numbers (from 10000 up to 65000), sand particles of two different average-sizes (120 and 200 microns) and different solids-loading ratios (up to 0.40). It was obtained for the plain tube flow that the presence of the solid particles in the turbulent flow enhances the heat transfer coefficient giving an enhancement ratio of the Nusselt number of 1.5 at M=0.4 and Re=35000 for suspended particles with 120 microns average-size. Moreover, at higher Reynolds numbers (over 64000) slightly enhancement in the heat transfer coefficient was observed. For the flow with different entrance angles, the heat transfer coefficient enhances in the upstream portion of the test tube. The results of clean-air flow show that about 1.18 fold increase in average Nusselt number was corresponding to about 1.5 fold increase in the pressure drop along the test-section. Also, for air-solids flow the presence of the solid particles in the flowing air enhances the heat transfer coefficient up to mass-loading ratio of M=0.15 and further addition of solids affects slightly the heat transfer coefficient. New correlations for the Nusselt number and its enhancement ratio were obtained for both cases of plain tube and for the tube with different entrance angles.

NOMENCLATURE

SI system of units was used for the whole following parameters:

Specific heat
Inner diameter of the test-section
Solid particles mean size
Connective heat transfer coefficient
Thermal conductivity
Total length of the test-section tube
Mass flow rate.
Heat flux
Temperature
Axial distance measured from tube inlet.
U Flow v

Greek l

Difference.
Dynamic molecular viscosity.
Entrance angle.
Density. ρ

Subs

Case of clean air flow, M = 0
Fluid (gas) phase
Particle phase
p.t. Plai
S Flow witl
Asymptotic value
θ Flow with entranc

Supers

Average value.

Dimensionless G

Solids-loading ratio $M = m_p/m_f$.
Solids to gas thermal capacity ratio, $M_c = m_p($
Nusselt number, $Nu = hD/k_f$.
Prandtl number.
P* Dimensionless pressure, $\Delta p/ 0.5\rho U^2$
Reynolds number, $Re = 4m/\pi D\mu_f$. Re

1. INTRODUCTION

Turbulent flow in T-junction branchment is an important types that exists in a wide variety of engineering situations ranging from internal combustion engine manifold and ventilation ducting to fluid devices.

In heat transfer and fluid flow applications, the existence of highly efficient thermal systems presents an important aim of most designers and researchers. This will bring about some savings in volumes and reduces overall weights [1]. In practice, several methods and techniques have been utilized to achieve this goal. Among these techniques is the inclusion of secondary and gas-solids flows. In these types of flows, abnormal turbulence are created. This in turn increases both the heat transfer rate and the pressure drop.

Davise and Al-Arabi [2] carried out an experimental work to study the effect of inlet abnormal conditions on heat transfer under uniform heat flux with water as a working fluid. The tested inlet conditions are: 90-deg-angels bend, sharp edge and different orifices. They concluded that for values of x/D (axial distance/diameter ratio) less than the critical value (thermal entrance length), the local heat transfer varies from one cross-section to another according to the amount of the abnormal turbulence. Also, for values of x/D greater than the critical value, the local heat transfer coefficient is independent of x/D is a constant. Moreover, the critical values of x/D for abnormal turbulence are higher than that for normal turbulence.

Khalil [3] carried out experimentally study for the effect of different orifices inserted at the tube inlet on heat transfer and pressure drop. Air as a working fluid, was flowing through a brass tube of 38 mm inner diameter and 4m long under uniform heat flux conditions. Four different orifices were used. It was concluded that the thermal entry increases with the rising of either Reynolds number or the turbulence intensities. However, for the same turbulence intensity the increase in the heat transfer coefficient is small compared with that in the pressure drop. Furthermore, after certain turbulence level no pronounced increase in heat transfer exists, whiles the pressure drop is still increasing. Moustafa [4] studied analytically the flow characteristics in a two-dimensional multidirection T-junction domain. The flow was treated as steady, incompressible and turbulent.

Abdel-Moneim [5] carried out an experimental study on the heat transfer and pressure drop characteristics of turbulent gas-solids flow in a horizontal tube of 40 mm inner diameter and 3m long under constant heat flux conditions. Two particles-sizes of 30 and 200 micros were suspended into the air flow. It was concluded that the presence of the solid particles in the air flow increases both the heat transfer coefficient and the pressure drop. These increases depend strongly on both Reynolds number and the solids-loading ratio. Also, at higher range of Reynolds number (over 60,000) no significant increase in the heat transfer coefficient was observed. Moreover, no significant increase was obtained in the pressure drop with the addition of 30 microns particles, while the addition of 200 microns solid particles increases the pressure drop.

The present work is divided into two parts. The first part deals with the case where fully developed turbulent flow already exists at the commencement of heating. This condition is maintained for the plain tube flow when a calming section of sufficient length is used. The second part deals with the case where abnormal turbulence exists at the commencement of heating due to the presence of a bend with different angles at the entrance to the test-section.

EXPERIMENTAL APPARATUS AND PROCEDURE. 2.

The experimental set-up shown in Fig.(1) was used to study the effect of solids loading-ratio and the entrance angle on both heat transfer and pressure drop characteristics. The test rig used in the present work comprises: an air passage, a solid particles feeding system, an entrance section with different entrance angles, a heated test-section with main and guard heaters, a solid particles separation system (two cyclones) and different measuring instruments. The test tube and the entrance section were made of brass tubes with 40 mm diameter. The length of the test section was 3m representing a length to diameter ratio of about 75. Details of the entrance section are shown in Fig.(2). The solids loading ratio based on the mass ratio was calculated from:

$$M = \frac{\dot{m}_p}{\dot{m}_f} \qquad (1)$$

where, m_p and m_f are the solids and fluid (air) mass flow rates, respectively. The wall temperatures were measured along the test section length at several test stations (40 different test positions) using thermocouples made of copper-constant wires of 0.5 mm diameter with an accuracy of $\pm$ 0.5%. Generally, two diametrically opposed, top-bottom, thermocouples were fixed on the test tube wall to measure the tube wall temperatures. While the pressure was measured at different locations along the tube by using a system of manometers.

The local heat transfer coefficient from the tube inner surface to the fluid flow for each segment of the test tube was calculated from:

$$h = \frac{q''}{\left(T_s - T_m\right)} \qquad (2)$$

where, q'' is the input heat flux and T_s and T_m are the mean temperatures of the wall surface and the flow at each segment, respectively. Also, the local Nusselt number and Reynolds number were obtained as;

$$Nu = \frac{hD}{k_f} \qquad (3)$$

and

$$Re = \frac{4\dot{m}_f}{\pi D \mu} \qquad (4)$$

where, the clean air flow properties (k_f and μ_f) were evaluated at the flow mean temperature.

3. RESULTS AND DISCUSSION

In the present experimental work, the effect of various parameter such as Reynolds number, solids loading ratio, solid particles size and the flow entrance angle on both heat transfer and pressure drop were investigated. The heat transfer coefficient was for both clean air and air-solids flow for normal and abnormal flow-entrance conditions.

3.1. Clean-Air Flow Results

The local Nusselt numbers for clean air flow were calculated along the test tube utilizing the present experimental data at different Reynolds numbers (Re=23400 and 52400) and at different inlet angles (θ=90, 30, 0 deg.). It was then plotted axially against the dimensionless axial location x/D as shown in Fig.(3). The local Nusselt number takes a relatively high value at the tube inlet and this value decreases with the axial distance until it reaches an asymptotic value at the end of the thermal entry length, then it begins to increase at the tube exit region. The phenomenon of increasing the local Nusselt number at the tube exit may be attributed to the effect of the axial conduction through the tube wall. Figure (3) shows that the value of the local Nusselt number at the tube inlet as well as its asymptotic value are functions of the associated Reynolds number.

Also, the local Nusselt number distributions for the different entrance angles, =0 deg.). Moreover, θshown in Fig.(3), take the same trend as that for the plain tube (the existence of entrance angle in the flow field enhances the local Nusselt number in the upstream portion of the test tube. The generation of the abnormal turbulence increases the value of the local Nusselt number above that for the plain tube at the same Reynolds number. This behavior may be attributed to the flow separation and reattachment that accrued immediately downstream of the test section inlet. However, in the absence of the tangential velocity component, the resulting turbulence appears to die down very rapidly after reattachment [6]. Consequently, the thermal entry length is increased above the value of the plain tube with increasing the entrance angle. A comparison between the present results for clean-air flow with these published by Davies and Al-Arabi [2] is shown in Fig.(4). This comparison was carried out for the conditions of a Reynolds number of 10700 and an entrance angle of 90 deg. Figure (4) shows that the present results are in a very good agreement with the measurements of [2].

The average Nusselt number shown in Fig.(5) is enhanced with increasing the entrance angle at different Reynolds numbers. It was found that the maximum value of the average Nusselt number enhancement ratio (average Nusselt number with entrance angle/average Nusselt number for purely axial flow) is about 118% at an entrance angle =90°. This maximum value is corresponding to nearly 1.5-times increase in the θ pressure drop. Furthermore the present experimental data for clean-air flow for both purely axial (plain tube) flow and flow with different entrance angles were used to correlate the average Nusselt number enhancement ratio with the entrance angle as;

$$\overline{Nu} = \overline{Nu}_0\left(1 + 1.289x10^{-3}\ \theta^{1.12}\right) \quad (5)$$

where,

$$\overline{Nu_0} = 0.01916\ Re^{0.8} \quad (6)$$

, from 0 to 90 deg. θThis correlation is valid in the range of the entrance angles, and Reynolds number, Re, from 10,000 to 58,000. The results show that the correlation, Eq.(5), gives a good agreement with the present experimental results within $\pm$ 6.4% deviation.

The effect of the entrance angles on the dimensionless pressure distributions, P^*, along the tube if shown in Fig.(6). The results indicate a higher value of pressure drop in the upstream portion of the tube. This drop increases by increasing the entrance angle at the same Reynolds number as shown in Fig.(7). Figure (7) shows that the pressure drop ratio $\left(\Delta P / \Delta P_0\right)$ is highly affected by the entrance angles due to the increase in the flow separation [4].

Air-Solids Flow Results 3.2.

For purely axial (plain tube) flow, the presence of the solid particles in the air flow enhances the heat transfer coefficient for both two particle sizes within the experimental range of Reynolds number. This may be attributed to the increasing of turbulence intensity with particles addition. The enhancement ratio which is defined by the ratio $\left(\frac{\overline{Nu}_{\infty}}{\overline{Nu}_{\infty,0}}\right)$ of the asymptotic Nusselt number for air-solids flow to that for clean air flow is shown in Fig.(8-a) for 200 microns particle size and Fig.(8-b) for 120 microns size. Figure (8-a, b) shows that the enhancement ratio approaches a maximum value about 1.5 at Re=35,000 and M=0.40 for the flow with fine particles (120 microns size). According to the dimensional analysis which have been performed by [7], the present experimental data was correlated relating the asymptotic Nusselt number for air-solids flow with Reynolds number, solids loading ratio and particles to tube diameter ratio as:

(7) $$= 0.02105\ Re^{0.8}\ Pr^{0.4}\ (1+M)^{0.48}\ (1+d/D)^{-0.2}\ {}_{\infty}Nu$$

This correlation is valid within the range of Reynolds number from 35000 up to 80000, solids-loading ratio $0 \leq M \leq 0.4$ and particles to tube diameter ratio from 3.0×10^{-3} up to 5.0×10^{-3}. The comparison between the present correlation and the previous correlations is given in Table (1) and is shown in Fig.(9) and good agreement is found.

Table (1): Comparison between The Present and Previous Correlations

Workers	Correlation	Deviation
Present	$= 0.02105\ Re^{0.8}\ Pr^{0.4}\ (1+M)^{0.48}\ (1+d$	- - - - -
[5]	$= 0.0212\ Re^{0.8}\ Pr^{0.4}\ (1+M)^{0.45}\ (1+d/$	-1.8%
[7]	$= 0.0202\ Re^{0.8}\ Pr^{0.4}\ (1+M)^{0.48}\ (1+d/$	-5.5%
[8]	$= 0.14\ Re^{0.6}\ (1+$	-12.5%

For Air-solids flow with different entrance angles, the results of the local Nusselt number for the two particle-sizes are shown in Fig.(10). The present results show that for the two particle-sizes, the local Nusselt number increases with the increase in both the entrance angle and solids loading ratio up to a value of solids loading ratio of about 0.15. Over this value of the loading ratio no significant enhancement in the local Nusselt number was observed. Also, it is observed that a longer thermal inlet length was obtained with a higher entrance angle and a higher loading ratio due to the longer axial distance required for decaying the abnormal turbulence.

Furthermore, the effect of the particle size on the local Nusselt number is shown in Figs.(11& 12). The results show that a dominant effect of smaller size particles than the coarser ones on the Nusselt number. This can be attributed to the higher number density of the smaller size for the same loading ratio [9].

A new correlation for the ratio of the average Nusselt number for air-solid flow with different entrance angle to the average Nusselt number for air-solids flow in plain-tube $\left((\overline{Nu_S})_\theta / (\overline{Nu_S})_{p.t.}\right)$ was obtained by using the present experimental data. The present data was correlated by using the least square method and the following correlation was obtained:

$$\left((\overline{Nu_S})_\theta / (\overline{Nu_S})_{p.t.}\right) = 1 + 0.994 \ \theta^{1.14} \qquad (8)$$

where,

$$(\overline{Nu_S})_{p.t.} = 0.022 \ Re^{0.8} \ Pr^{0.4} \ (1+M)^{0.48} \ (1+d/D)^{-0.2} \qquad (9)$$

The present correlation Eq.(8) is valid within the range of the entrance angle of ($0 \leq \theta \leq 90$ deg.), Reynolds number of ($25000 \leq Re \leq 52000$), solids loading ratio up to 0.25 and d/D from $3x10^{-3}$ to $5x10^{-3}$. Equation (9) was obtained for the case of plain tube flow and it is valid with the range of Reynolds number up to 80000 and solids loading ratio up to 0.40.

4. CONCLUSIONS

The present measurements lead to the following conclusions:

1- For the two particles sizes, the presence of the solid particles in the air flow enhance the heat transfer coefficient. Also, the presence of solids, increases the pressure drop along the test-section.

2- For the flow with different entrance angles, the local Nusselt number enhances at the upstream portion of the tube only with increasing the entrance angle.

3- For clean-air flow with different entrance angles, the maximum enhancement ratio of average Nusselt number $\left(\overline{Nu} / \overline{Nu_0}\right)$ is about 1.18 corresponding to nearly 2 times of the pressure drop ratio.

4- For clean-air flow with different entrance angles, the following correlation for the average Nusselt number is valid within a range Reynolds number from 10000 to 58000 and entrance angle up to 90 deg.: $\overline{Nu} = 0.0191 Re^{0.8}\left(1 + 1.289x10^{-3} \ \theta^{1.12}\right)$.

5- For air-solids flow at relatively low loading ratio up to M=0.15, the local Nusselt number depends on both the entrance angle and solids loading ratio. At higher values of M no significant increase in the local Nusselt number was observed.

6. The asymptotic Nusselt number for purely axial (plain tube) flow was correlated within the range of Reynolds number from 32,000 to 80,000 and solids loading ratio up to 0.4 as; $Nu_\infty = 0.02105 \ Re^{0.8} \ Pr^{0.4} (1+M)^{0.48} (1+d/D)^{-0.2}$.

7- New correlation for the ratio $\left((\overline{Nu_S})_\theta / (\overline{Nu_S})_{p.t.}\right)$ was formulated as;

$$\left((\overline{Nu_S})_\theta / (\overline{Nu_S})_{p.t.}\right) = 1 + 0.0994 \ \theta^{1.14}$$

6. REFERENCES

1- Mostafa G.M., "Effect of Swirling Angle and Swirls Number on Heat Transfer and Pressure Drop in Turbulent Tube Flow", M.Sc. Thesis, Faculty of Eng., Cairo University, 1987.

Davies, V.C. and Al-Arabi, M., "Heat Transfer Between Tubs and A Fluid Flowing 2-
Through Them With Varying Degrees of Turbulence due to Entrance Conditions", Institution of Mech. Eng., West Minster, London, Vol. 169, No. 48, pp. 993-1006, 1995.

Khalil, E.E., "Heat Transfer by Forced Convection in The Entry Sections of Tubes"., 3-
M. Sc. Thesis, Faculty of Eng. Cairo University, 1973.

Mostafa A.A.A., "A Numerical Analysis of Turbulent Flow in Multidirectional T- 4-
Junction.", The First ICEMP., Cairo Egypt, Feb. 1991.

Abdel-Moneim S. A., "Heat Transfer to Air-Solids Turbulent Flow in Pipes", Ph.D. 5-
Thesis, Cairo University, 1992.

Guo, Z and Dhir, V.K., "Single and Two-Phase Heat Transfer in Tangential 6-
Injection Induced Swirl Flow", Int. J. Heat and Fluid Flow, Vol. 10, No. 3, pp. 203-210, September 1989.

Abdel-Moneim, S.A., "Heat Transfer to Air in Turbulent Pipe Flow With 7-
Suspended Solid Powder", M. Sc. Thesis, Faculty of Eng., Cairo University, 1989.

Farber, L. and Morley, M.Y., "Heat Transfer in Flowing Gas-Solids Mixtures in a 8-
Circular Tube", Ind. and Eng. Chem., Vol. 49, pp. 1143-1150, 1957.

Abou-Arab, T.W. and Abou-Ellail, M.M.M., "Modulation of Heat Transfer Dusty- 9-
Gas Pipe Flows", Proceedings of the 6th Int. Conf., For Mech. Power Engineering Hosted by Monoufia University, Cairo, 1986.