HEFAT2008
6th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
30 June to 2 July 2008
Pretoria, South Africa
Paper number: NM1

A study of Heat Transfer Enhancement on a Tilted Rectangular Stainless Steel Plate

M.B.W. Nabhan
Department of Mechanical Engineering, College of Engineering
University of Bahrain
Isa Town, P.O.Box: 32038
Kingdom of Bahrain
E-mail: mnabhan@eng.uob.bh

Abstract

An experimental study is presented to investigate the heat transfer variations in a horizontal trapezoidal duct with the addition of heat plate baffle, namely; rectangular stainless steel flat plate. Fully developed turbulent air flow is maintained in the test section, which is connected to a subsonic wind tunnel.

Air forced over a hot plate will eliminate an amount of heat from the plate according to the rules of cooling law. The air flow characteristics are determined using one baffle with different angle positions and different air flow velocities.

The pressure drop Δp, Reynolds Number $R_{e,D,}$ Pressure Coefficient C_p and heat transfer Nusselt Number Nu, show a strong dependence on baffle plate altering angle positions. It is found that the larger change in test plate angle position from 60^0 to 90^0, the larger the pressure drop, pressure drop coefficient, and lower the heat transfer enhancement.

Introduction

Placing obstructions (baffle) in a duct under fluid flow and see the effect of baffle rotation on air flow characteristics as in a rotated valve disc in its body is important to estimate the amount of waste in energy for designers and professionals. Such obstruction provides a means to turn away the flow and cause impingement, recirculation, and separation. Due to placing of the baffle, direction of the fluid is sidetracked from the original path. This may lead to complex flow model and make the flow multi dimensional. A theoretical solution is not straightforward to analysis. An experimental study will be more suitable in this situation. The effect of rectangular baffle openings as in slider valves on air flow characteristics, energy losses and heat transfer enhancement has been evaluated by Nabhan et al. [1]. One way to obstruct a fluid flow is by placing an obstruction or a baffle in its path, which diverts the flow from its original path. This may lead to complex flow pattern and render the flow multi-directional.

Numerical prediction of fluid flow and heat transfer in a circular tube with longitudinal fins interrupted in the stream-wise direction has been studied by Kelkar et al. [2] and the results have indicated that in the periodic fully developed regime, for Prandtl number of 0.7, a tube with staggered arrangement of fins produces less heat transfer enhancement than a tube with continuous fins. Hsieh and Shou-Shing [3] have analyzed turbulent heat transfer and flow characteristics in a horizontal circular tube with strip-type inserts and found that Nusselt numbers were between four and two times of the bare tube values at low and high Reynold's numbers respectively. Moreover, a numerical study of flow field and heat transfer in a heated circular tube with an inner tube inserted has been investigated by Fu et al. [4]. The results had shown that, except at very low Peclet number, the heat transfer rate of the heated tube increases when an inner tube was inserted. Aoyama et al. [5] investigated turbulent heat transfer enhancement by a row of twisted plates at 90 degrees alternately in different directions and found that heat transfer coefficients increased. In addition, Zhang et al. [6] investigated turbulent heat transfer enhancement by a row of twisted-tape inserts and axial interrupted ribs and found that heat transfer coefficients also increased. Moreover, Bunker et al. [7] observed that the heat transfer enhancements for dimpled internal surfaces of circular tubes are larger than bare tube values. In addition, Mendes et al. [8] determined experimentally heat transfer coefficients and pressure drop data for turbulent flow through internally ribbed tubes and found that ribbed tubes have higher heat transfer performance for most of the cases investigated. A similar behavior for heat transfer augmentation was observed by Zhu et al. [9] using a flag type insert in a circular tube. Moreover, a theoretical study by Wang et al. [10] to develop a novel heat transfer enhancement technique in a circular tube with micro-fiber fin found that the heat transfer improves with an increase in the length of the fin to diameter of tube ratio and the thermal conductivities ratio of the fin, and the fluid.

The work investigates the effect of baffle geometry and position of baffle on fluid characteristic, such as pressure drops, friction factor and heat transfer. An experimental dimensionless form of the Pressure Coefficient C_p, Reynolds number $R_{e\text{-}D}$, and Nusselt number Nu is established.

Nomenclature

A	[m^2]	Area of test duct
A_b	[m^2]	Opening area at baffle
V	[m/s]	Local fluid velocity
V_m	[m/s]	Mean fluid velocity
g	[m^2/s]	Gravity constant
h	[$W/m^2.{}^oC$]	Coefficient of heat transfer
Lc	[m]	Characteristic test plate length
L	[m]	Test duct length
P_o	[Pa]	Stagnation pressure
P_s	[Pa]	Static pressure
P_d	[Pa]	Dynamic pressure
ΔP	[Pa]	Static Pressure drop
k	[W/m.K]	Thermal conductivity coefficient
T_s	[^{o}C]	Temperature of plate surface
T_F	[^{o}C]	Temperature of surrounding
Δx	[m]	Anemometer probe position
S	[m^2/s]	Δx x V
N	[m]	Perimeter of duct
ρ	[kg/m^3]	Density
μ	[kg/m.s]	Absolute fluid viscosity
υ	[m^2/s]	Kinematics viscosity = μ/ρ
γ_a	[N/m^3]	Specific weight of air
C_p		Pressure coefficient
$R_{e,D}$		Reynolds number
Nu		Nusselt number

Theory

The pressure drop needed to sustain an internal flow across the baffle in the test duct shown in Fig.1, depends on duct mean velocity V_m, the density of fluid ρ_f, the fluid viscosity μ_f, the duct hydraulic diameter D_H, the clearance height y, and the plate area A_p. This could be expressed as;

$$\Delta P = f\,(V_m, \rho_f, \mu_f, D_H, y, Ap) \qquad (1)$$

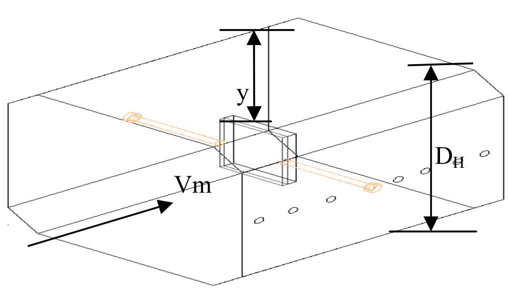

Fig.1 Test duct section with baffle flow geometry

The pressure drop in the investigations is determined by measurement, using the static pressure Multifunction Calibrator instrument arrangement attached to the wind tunnel where;

$$\Delta P = P_o - P_s = (1/2)\,\rho_a\, V_m{}^2\, C_p \qquad (2)$$

The local air velocity V is measured using Hot-Wire Anemometer device, where the mean air velocity in the duct before the test plate is determined using the following relationship,

$$V_m = S / D_H = [\Delta x_1\,(0+V_1)/2 + \Delta x_2\,(V_1+V_2)/2 + \Delta x_3\,(V_2+V_3)/2 + \Delta x_3\,(V_3+V_4)/2 + \Delta x_4\,(V_4+V_5)/2 + \Delta x_5\,(V_5+V_6)/2 + \Delta x_6\,(V_6+0)/2] / D_H \qquad (3)$$

The dimensionless pressure drop coefficient, C_p and Reynolds number, $Re_{,D}$ for the flow arrangement under investigation is given by the following relationship in order to present the flow characteristics;

$$Re_{,D} = \rho_f\, V_m\, D_H/\mu_f \qquad (4)$$

The total pressure drop, $\Delta P_t = \Delta P_f + \Delta P_p + \Delta P_e + \Delta P_{D/S} = ½\, V^2{}_m\, \rho_f\, [f\,(L/D_h) + K_p + K_e + C_D]$

The pressure drop coefficient, C_p is a function of the total static pressure drop ΔP_t comprising, duct friction ΔP_f, inertia losses ΔP_p, entrance/exit losses ΔP_e, and the drag/skin friction $\Delta P_{D/S}$ pressure drops and is given by equation 5 below.

$$C_p = 2\,\Delta P_t/\,\rho_f\, V^2{}_m \qquad (5)$$

For circular pipes, the diameter (D) is the characteristics length parameter used in the various equations. For non-circular ducts as the case in hand the hydraulic diameter (D_H) is used where,

$$D_H = 4\,A/N \qquad (6)$$

The heat transfer rate (Q) from the heated plate is given by the Nusselt number (Nu), where;

$$Q = h\,A\,\Delta T \text{ and } Nu = h\,Lc/K \qquad (7)$$

Experimental work

Experiments are conducted in the wind tunnel available in the University Fluid Mechanics laboratory. The wind tunnel is modified to encompass a horizontal trapezoidal test conduit section of hydraulic diameter 280 mm and 458 mm length designed and manufactured from plastic material. The test duct is positioned such that the airflow in this section is fully developed thus minimizing friction losses and ensuring fully developed boundary layer. The Honeycomb at the inlet of the wind tunnel is designed to provide uniform fluid flow in the region immediately after the Honeycomb and Bell-Mouth where the test duct section is positioned. The air flow is provided using a fan which has a rotational speed control unit. The test baffle plate is placed inside the test duct section and is secured on a tilt mechanism shown in Fig.2. Measurements of the free-stream air velocity and static pressure in the test section is performed using Hot-Wire Anemometer and Multifunction Calibrator instruments respectively.

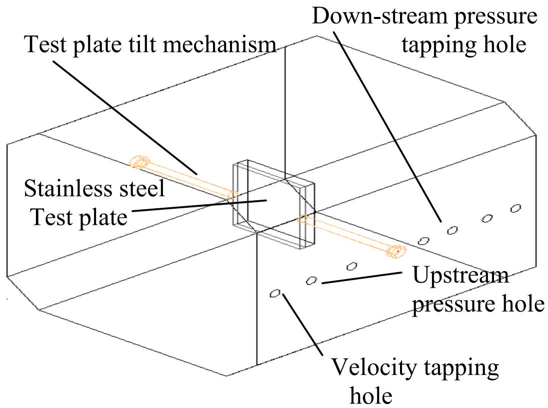

Fig.2 Schematic view of test conduit, test plate and tilt mechanism arrangement

The baffle test heat plate is made to suitable thickness and dimensions. A rectangular heated stainless steel plate of 0.66 cm thickness of which embedded on its back are 10 thermocouples uniformly distributed and connected to a digital-sense thermometer to read the temperatures at different locations. The plate is heated by a Ni-Cr bean and cable placed in a groove at the back of the plate, insulated by a plastic sheet and connected to an electrical circuit consisting of a voltmeter, ammeter and variable transformer to supply a steady heat flux. The temperature of the plate is changed by altering the voltage power supply by the variable transformer device. The plate is positioned in the fully developed flow section at the center of the test duct and is allowed to tilt by a rotating mechanism to produce different flow clearances.

The uncertainty in the measured parameters taken by the main instruments, namely the Multifunction Calibrator, Anemometer, and the K-type thermocouples are ± 0.05 %; ± 0.03 m/s, and ± 0.4 % respectively. This gives propagation of uncertainty in the measured results of 4.4% for $V_{m,avg}$ at 90° plate position, 3.2% $V_{m,avg}$ at 60° plate position, 8.2% $R_{e,Davg}$ at 90° plate position, 6.7% $R_{e\text{-}D,avg}$ at 60° plate position, 6% heat loss from the plate, 0.8% temperature differential, 9% heat transfer coefficient, 9.6% Nusselt number, and 0.05% Pressure drop.

Results

In this investigation, a heated baffle plate is used in order to test the effect of the clearance size and shape of baffle on the pressure drop, friction, and heat transfer rate through a trapezoidal duct. The difference between the static pressure before and after the duct is observed along the axial distance of the air duct test section using the Multifunction Calibrator instrument. The local air velocity in the duct is measured using Anemometer instrument. Equation (3) is used to find the mean air velocity for a particular motor controller setting. The Reynolds number $Re_{,D}$ and pressure coefficient C_p are also determined in order to present the flow characteristics in the test duct section using equations (4) and (5) respectively. Newton's cooling law is applied for convective heat transfer across the test plate to determine the Nusselt number Nu using equation (7).

An experiment with the plate in a vertical position is conducted with different values of V and afterwards the flow clearances are changed by tilting the test plate at other tilt angles, and the different flow characteristics across the test plate are determined. The mean velocity, V_m is calculated from the measured various local velocities using equation (3) in the theory section, for plate positions 90° and 60°. Reynolds number is the ratio of inertial forces to viscous forces and consequently it quantifies the relative importance of these two types of forces for the given flow conditions. Thus, it is used to identify different flow regimes, such as laminar or turbulent flow. Laminar flow in ducts occurs at low Reynolds numbers ($R_{e\text{-}D} < 2300$), where the viscous forces are dominant, and is characterized by smooth, constant fluid motion, where turbulent flow, occurs at high Reynolds numbers ($R_{e\text{-}D} > 4000$) and is dominated by inertial forces, producing random eddies, vortices and other flow fluctuations. From equation (4) in the theory section $R_{e\text{-}D}$ is calculated for 90° and 60° tilt test plate angles.

Heat loss from the test plate is initially determined at no fluid flow conditions, i.e. zero fan speed to calibrate the plate for heat losses as shown in Fig.3. The temperature of the test plate was allowed to stabilize before recording the temperature points on the plate. This occurred at time equals 20 minutes after starting the power supply. This time was taken as a reference time for the tests which followed in the investigations with air flow afterwards.

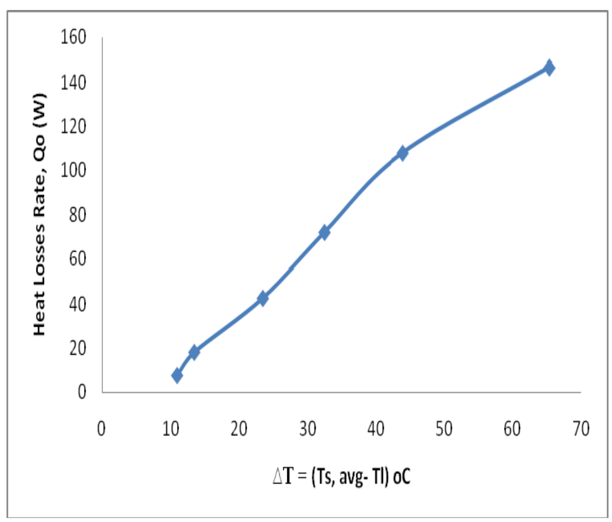

Fig.3 Heat losses calibration curve

The heat transfer coefficient, h can be determined at the in-flow for each mean velocity, for the different test plate tilt angles namely 90 and 60 degrees using equation (7) of the theory section. The graphs in Fig.4 and 5 show the relationship between Nusselt and Reynolds numbers. Moreover, using the Multifunction Calibrator instrument, the static pressure drop before and after the test plate is measured at tilt angles of 90 and 60 degrees and different flow velocities. Fig.6, 7 and 8 depict the pressure measurements and pressure coefficient variations with $R_{e\text{-}D}$ at the different test angles under investigation. Moreover, the thermal images of the heated plate are taken at the different tilt angles and air flow velocities to study the distribution of temperatures over the test plate. A sample of these photographs is shown in Figs.9 and 10.

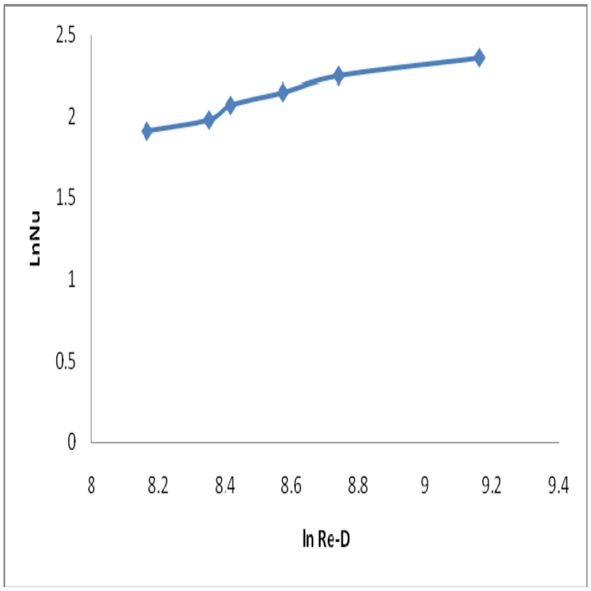

Fig.4 ln Nu versus ln $R_{e\text{-}D}$ for 90° position

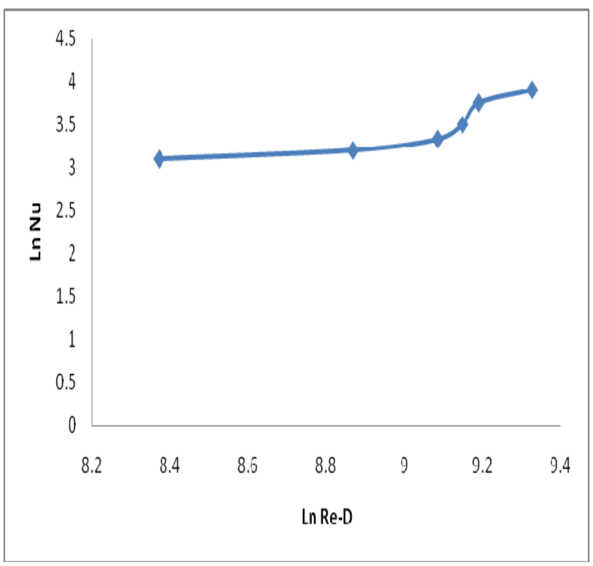

Fig.5 ln Nu versus ln $R_{e\text{-}D}$ for 60° position

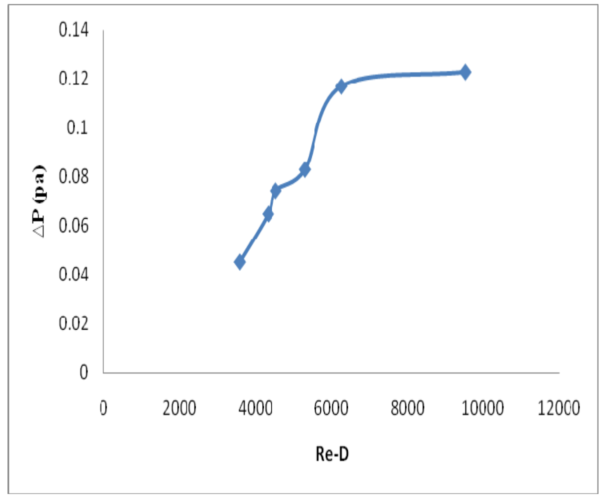

Fig.6 Static differential pressure at 90° tilt

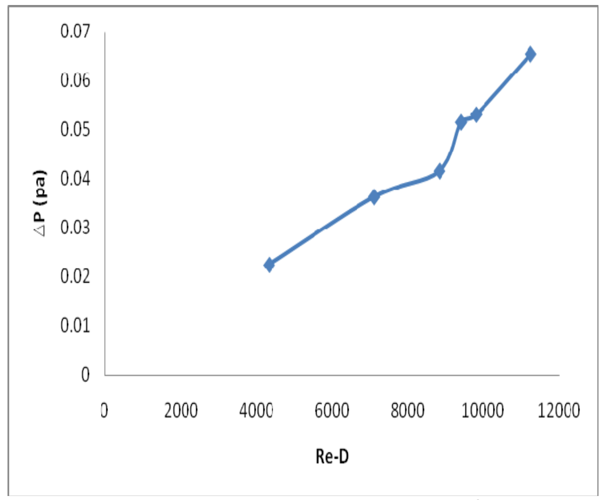

Fig.7 Static differential pressure at 60° tilt

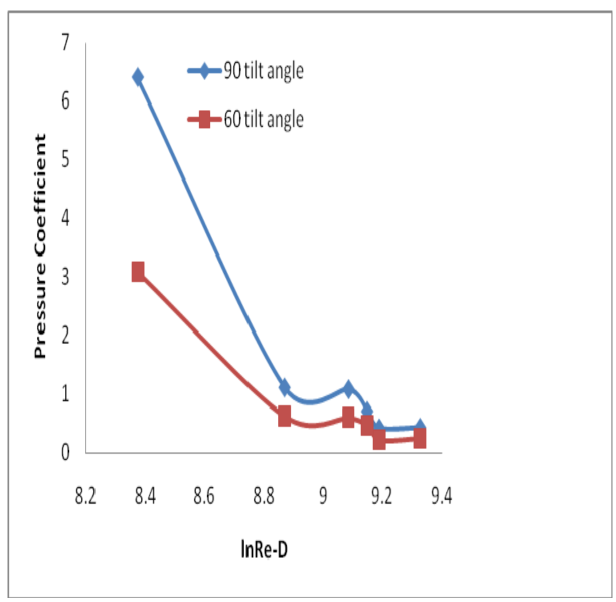

Fig.8 Pressure coefficient at 90°and 60° tilt angles

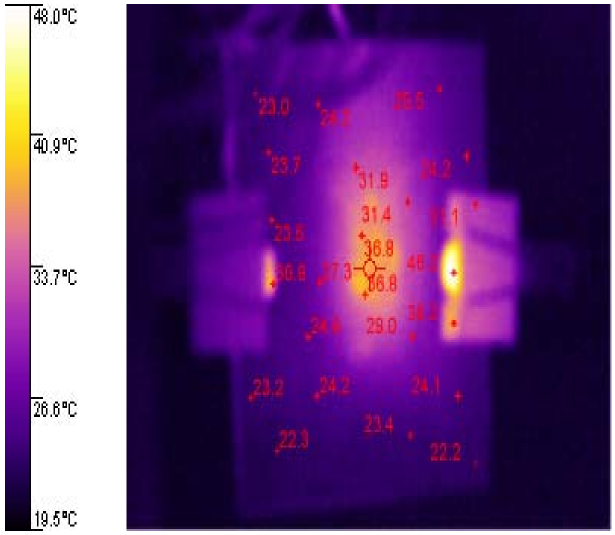

Fig.9 Thermal image of test plate at 90^0 tilt

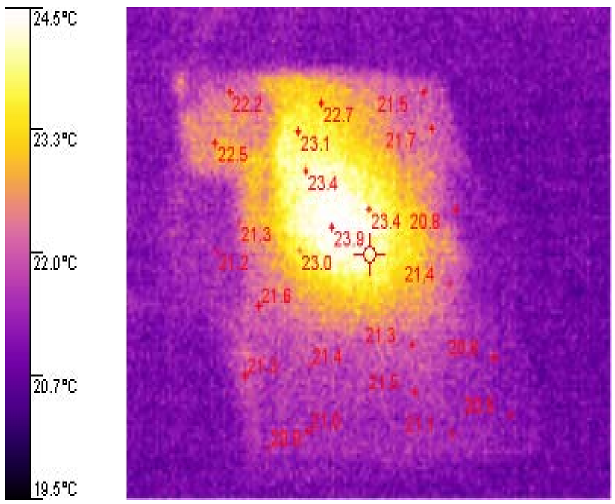

Fig.10 Thermal image of test plate at 60^0

Discussion of Results

An experimental study is presented to measure heat transfer, the pressure drop and pressure coefficient variations in a horizontal duct with the insertion of a rectangle stainless steel baffle heated plate. Fully developed turbulent air flow is maintained in the test duct section, which is connected to a subsonic wind tunnel.

Fig.3 depicts the calibration curve to calculate the heat loss from the plate at various heated plate temperatures. This information is used to determine the convective heat transfer rates afterwards.

Fig.4 and 5 show the mean Nusselt number, lnNu variations with Reynolds number, $R_{e\text{-}D}$. The heat transfer from the plate increases when the plate tilt angle decreases from 90^o to 60^o. This can be perhaps explained as the air stream before the test plate and after the plate has two kinds of flow structure. In front of the plate and behind of the plate there is a recirculation flow domain, which is separate from the main flow stream. It can be seen that when the angle position decreases, the clearance opening increases, thus the re-circulating flow domain decreases, which causes higher air mass flow rates and heat transfer rates. Also it is noticed that Nu number increases at higher $R_{e,D}$ values as expected. It is also noticed that the heat transfer is almost exponential with the 60^o tilt angle compared with a smooth and slow increase in heat transfer for the 90^o tilt angle at increasing flow velocities exemplified by the $\ln R_{e\text{-}D}$ values.

Fig.6 and 7 show the Reynolds number variation with the pressure drop across the heat plate. The pressure drop increases with increasing velocity and decreases with decreasing the angle of plate. It is noticed that as the angle position decreases the pressure coefficient is reduced as shown in Fig.8 due to lower drag on the 60^o plate position compared to the 90^o tilt position.

Fig.9 and 10 show the thermal images taken for the test plates. The temperature distributions on the periphery of the plates is the lowest compared to central parts. The maximum temperature is at the center of the test plate. This is attributed perhaps to the stagnation air velocity in the center of the plate. The air flow re-circulates after impact on the center of the test plate causing the temperature on the sides of the plate to reduce by convection heat transfer. Further reduction in temperature is attributed to larger air mass flow rate in this region. The 60^o tilted angle plate has lower temperature distribution on its surface due to larger air mass flow rates on the center and periphery. This indicates higher heat transfer enhancement compared to the 90^o tilt angle plate position.

Conclusions

The work in the test duct wind tunnel has comprised different investigations of the flow characteristics across the heated rectangle baffle plate. It is interesting to present the turbulent subsonic flow characteristics of these results together.

Pressure drop characteristics:

1- The pressure drop increases with increasing angle position.
2- The pressure drop increases with increasing mean velocity and mass flow rate.
3- The pressure drop coefficient decreases with increasing $R_{e\text{-}D}$ and it reduces at lower tilt angles.

Heat transfer characteristics:

1- Heat transfer from the baffle plate is highest at 60 degrees angle position.
2- The thermal images show that heat transfer is highest at the edges of the test plate and lowest at the centre due to stagnation air flow conditions.
3- The 60^o tilt angle has shown larger heat transfer rates on the center of the plate and periphery compared with the 90 degrees tilt angle, because of larger mass of air flow rate on the plate surface.

Recommendation

The author recommends the use of different test materials, test shapes, plate tilt angles and various duct cross sections for comparison with the present investigations to obtain optimum heat transfer enhancement and pressure drop values. In addition the tests should be subjected to higher velocities than the ones used in the present work to have a better understanding of the graph trends. Moreover, the value of this work could be enhanced further by using a mathematical model for comparison with the experimental findings.

Acknowledgment

The author would like to thank the Deanship of Research at the University of Bahrain for financially supporting this work.

References

[1] Nabhan et al., Fluid flow characteristics in a subsonic wind tunnel with baffle plates, senior project, department of mechanical engineering, university of Bahrain, June 2003.

[2] Kelkar K.M., Patankar S.V, Numerical prediction of fluid flow and heat transfer in a tube with longitudinal fins interrupted in the stream-wise direction, Journal of Heat Transfer, Transactions ASME, v 112, n 2, May, 1990, p 342-348.

[3] Hsieh, Shou-Shing, et al., Turbulent heat transfer and flow characteristics in a horizontal tube with strip-type inserts, International Journal of Heat and Mass Transfer, v 46, n 5, Feb, 2003, p 823-835.

[4] Fu, Wu-shung, et al., Enhancement of heat transfer for a tube with an inner tube insertion, International Journal of Heat and Mass Transfer, v 37, n 3, Feb, 1994, p 499-509.

[5] Aoyama, Yoshiyuki, et al., Turbulent heat transfer enhancement by a row of twisted plates, Nippon Kikai Gakkai Ronbunshu, B Hen, v 53, n 487, Mar, 1987, p991-996.

[6] Zhang, Y.M., et al., Heat transfer and friction characteristics of turbulent flow in circular tubes with twisted-tape inserts and axial interrupted ribs, J of Enhanced heat transfer, v 4, n 4, 1997, p 297-308.

[7] Bunker, Ronald S., et al., Heat transfer and friction factors for flows inside circular tubes with concavity surfaces, Journal of Turbo-machinery, v 125, n 4, Oct, 2003, p 665-672.

[8] Mendes, P.R.S, Mauricio, M. H. P., Heat transfer, pressure drop, and enhancement characteristics of the turbulent flow through internally ripped tubes, ASME, Heat Transfer Division, (Publication) HTD, v 82, 1987, p 15-22.

[9] Zhu, Yongquan, et al., The augmentation of heat transfer for gas using flag-type insert, Journal of Chemical Engineering of Chinese Universities, v 7, n 2, Jun, 1993, p 131-135.

[10] Wang, Song, et al., Convective heat transfer enhancement in a circular tube with micro-fiber fin, Qinghua Daxue Xuebao/Journal of Tsinghua University, v 40, n 4, Apr, 2000, p 55-57.