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Influence of Transverse Flat Baffles on Heat Transfer Augmentation in a Confined Rectangular Channel

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Abstract

Heat transfer intensification finds its importance in many practices such as air conditioning, heat exchanger and refrigeration structures. On heat transfer augmentation many researchers conducted experimental and numerical investigations to develop various techniques and methods. Heat transfer enhancement techniques are used to nurture the heat transfer rate without disturbing the complete performance of the system. The energy transport is considered enhanced if the flow is agitated and mixed well. This is the fundamental principle in the development of enhancement techniques that generate swirl flows. Nowadays, baffle inserts have widely been applied for improving the convective heat transfer in various applications due to their effectiveness, low cost and easy setting up. In proposed work an array of straight flat baffles was mounted transversely to the direction of air flow in a rectangular cross-sectioned duct and experiment was conducted for different Reynolds numbers ranging from 4938 to 6478. Investigations were performed to evaluate the influence of the presence of transverse straight flat baffles on the flow path. It was observed experimentally that the straight flat baffles greatly influences rate of heat transfer than that of the channel without baffles. The flow visualization for various Reynolds numbers also reports the significance of the presence of straight flat baffles in transverse directional flow. The influence of transverse flat baffles significantly effects the flow and found that increase in heat transfer rate and considerable pressure drop. Increase in Nusselt number, pressure drop and convective heat transfer coefficients were reported for various Reynolds number.

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1. Introduction

Augmentation heat transfer or enrichment is a method of improving the convective heat transfer rate. Various heat transfer enhancement techniques are used in different types of industrial applications such as heat exchangers, electronic cooling devices and thermal regenerators. In the heat exchanger design, ribs, fins or baffles are often employed as turbulators in order to upturn the convective rate of heat transfer. The baffle geometry and arrangement in the duct affect the flow field and influence the convective heat transfer. Specifically, the baffles angle, cross-section of the baffle, the ratio between baffle-to-duct height and baffle pitch-to-height ratio are the parameters that impact the global thermal performance. Enrichment in the removal of heat tends to improve the efficiency and the life span of the devices.

Wang and Sunden [1] proposed the augmentation of heat transfer leads to decrease in the size of heat exchanger and reduction of the capital cost. Tube and fin type's heat exchangers with different fin patterns namely Rectangular, Polynomial and Sinusoidal are the normally used heat exchangers [2]. The learning of elevating in the performance of heat transfer is referred as enhancement, augmentation, or intensification of heat transfer [3]. Heat exchanger plays a vital role in domestic, power production plants and industries like chemical, food, electronics and other manufacturing of equipments, air- conditioning, Solar heater, refrigeration, petrochemical, automobiles and space applications [4-7].

Excellent characteristics of nanofluids exhibits their great potential use in heat transfer areas like microelectronics, plate heat exchangers and hybrid-powered engines [8]. Different types of inserts has been used in heat exchangers Coiled wire [9-11], ribs/fins/baffles [12-14], winglets [15,16] and helical/twisted tapes [17,18]. Rupesh J. Yadav et al. [19] Transfer of heat and friction factor was analyzed employing the twisted tape with four varying twist ratios. The duct used for the experimentation is square and hexagonal duct and the results were compared with the circular duct. The result says that decline in the twist ratio and elevation in the Reynolds number leads to raise in Nusselt number. Decline in the twist ratio generates good swirl motion. This swirl motion creates healthier mixing and higher pressure drop leads to better transfer of heat and higher friction factor. Circular duct using twisted tape with twist ratio of 3.5 shows maximum pressure drop. Circular duct shows maximum Nusselt number and friction factor followed by hexagonal and square duct.

J.P. Meyer and S.M. Abolarin [20] reported experimental data on transitional flow region making use of twisted tape along with three varying twist ratios and three different heat fluxes were investigated. At the entry of the test section a square edge inlet is introduced. Couple of methods was applied to probe the starting region and terminal region of the transition flow. It was found that for varying heat flux and constant twist ratio, the heat flux prolonged the transition from laminar to transitional flow. Smith Eiamsa-ard and PongjetPromvonge [21] Serrated twisted tape with varying width ratio and depth ratio were reviewed and the outputs were examined with the typical twisted tape and plain tube The rate of heat transfer rises with raise in depth ratio and declining in width ratio Handling of serrated twisted tape increase in rate of heat transfer by 72.2 and 27% compared to normal twisted tape and plain tube Further observation shows that serrated twisted tape generates high turbulence at the periphery edge pertaining to the twisted tape Serrated twisted tape shows decreasing in thermal performance factor with the rise in the Reynolds number.

Manvendra Tiwari and Sujoy Kumar Saha [22] circular pipe with transverse rib integrated with oblique teeth twisted tape were analyzed. The output shows that enhancement of heat transfer for the laminar flow was observed. Sagnik Pal and Sujoy Kumar Saha [23] Moderate amount of enhancement were observed for Circular tube with roughness integral spiral rib incorporated along with the twisted tape having oblique teeth for laminar flow. Table 5 presents the summary report on the effect of peripherallycut twisted tape using air or water as the working medium to analyze the heat transfer characteristics.PongjetPromvonge [24] V-finned counter twisted tapes fitted inside the square duct shows the enhancement in transfer of heat and drop in pressure V- finned twisted tape shows highest value of transfer of heat and pressure drop for the largest blockage ratio and also for the

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smallest pitch ratio. Maximum Nusselt number and the friction factor were obtained for blockage ratio= 0.42, pitch ratio = 4 and paramount thermal increment factor = 1.75 was attained for blockage ratio =0.21, pitch ratio = 4.

ChaitanyaVashistha et al. [25] Single, twin and four co and counter twisted tape were taken for the experimentation to analyze the flow pattern, heat transfer and Nusselt number characteristics. Observation for the twist ratio 2.5 shows that rise in Reynolds number, decrease in the friction factor and increase in Nusselt number were found for all the case of twisted tape . Four counter swirl twisted tape shows enrichment in Nusseslt number and the friction factor compared to other twisted tapes. Co swirl twisted tape generates eddy motion in the same direction. Counter swirl twisted tape generates eddy motion in opposite direction to each other this leads to generation of vortex in opposite direction. This also initiates the secondary flow and increases the vortex and unsteady motion occur results in good fluid mixing. The thermohydraulic performance rises with the reduction in the twist ratio is Further notification shows that the highest value of thermohydraulic performance of 1.26 were observed in case of four counter swirl twisted tape with twist ratio of 2.5.

M. Kh. Abdolbaqi et al. [26] twin twisted tape with counter swirl flow and Co swirl flow arrangements were analyzed. The Counter swirl flow preparations improves rate of heat transfer compared to the Co swirl flow arrangement. Heat transfer is 22.5% higher and 61% more than the Co swirl flow arrangement and plain tube. For the minimum twist ratio the thermal enhancement index for the counter swirl flow is 2.52 and Co swirl flow arrangements is 2.26. Yuxiang Hong et al. [27] Conducted experiments on the multiple twisted tapes to analyze the thermal and flow resistance. It is noted that all multiple twisted tapes perform good heat transfer than the plain tube. Amar Raj Singh Suri et al. [28] multiple square perforated twisted tapes with square wings were analyzed experimentally. The results reveal that multiple square perforated twisted tape with square wings gives Nusselt number and the friction factor of 6.96 and 8.34 times more than plain pipe. Multiple square perforated twisted tape along with the square wing shows good thermal hydraulic performance than the perforated and multiple twisted tape. Wing depth ratio plays a dominant role in the enhancement in the thermal hydraulic performance and it was achieved as 3.08.

K. Yongsiri et al. [29] Impact of pitch ratio (P/D) and alternate length ratio (l/P) on the friction and thermal performance factors and Nusselt number using alternate axis helical twisted tape [HTT-A] helical twisted tape [HTT] were studied experimentally. HTTA achieve good result for the enhancement in transfer of heat and the thermal performance factor of 14.1% and 1.9% than the HTT. HTT –A with small pitch ratio 1.0 and nearby twist length shows highest heat transfer and friction factor compared to other pitch ratios 1.5 and 2.0 and alternate twist length. Indri Yaningsih and Agung Tri Wijayanta [30] Study on the typical twisted tape [TT], Triangular wing [T-Tri], Rectangular Wing [T-Rec] and Trapezoidal wing [T-Tra] for varying Reynolds number, constant wing-chord ratio (d/W) 0.31, constant wing-span ratio (b/W) 0.23 and constant twist ratio (y/W) 3.8 were carried out experimentally. The major results are Nusselt number raises along with escalating in the Reynolds number. Nusseslt seems to be more for the twisted tape with the wings than the TT and the Plain tube. Twisted tape with the wings in the alternate axis generates physically powerful impact of the working fluid at the alternate point of the alternate axis and extra additional fluid disorder by the wing.

Siva Rama Krishna et al. [31] Single direction and left right direction of full length twisted tape were analyzed. The author incurred that both the full length twisted tape performs increases in Nusselt number with increases in the

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Reynolds number and decreases in the space distance highest value of Nusselt number is reached for space distance of 2 in. Compare to the single direction full length twisted tape the Nusselt number is less for the left right direction full length twisted tape. Immediate change in the direction of the flow is not permitted and this leads to the less intensity of the turbulence is generated in the left right direction of full length twisted tape. Intensification in transfer of heat is more than left right full length twisted tape. Observation shows that performance ratio is enhanced when the Reynolds number rise and is upto laminar flow. Maximum performance ratio of 1.06 is achieved for the Reynolds number of 2550. K. Syed Jafar and B. Sivaraman [32] parabolic trough collector (PTC) with twisted tapes with varying twist ratios and nail twisted tape inserts were investigated. Output observed that nail twisted tape shows highest Nusselt number compared to the plain tube and plain twisted tape [P-TT]. Presence of the nail in twisted tape minimize the discharge diameter in tube and generates the swirl motion leads to improve the fluid mixing and destruction of thermal boundary coating. This makes in enhancement of heat transfer for a longer duration within the core and the absorber tube wall. Plain twisted tape with twist ratio 2 performs enhance in Nusselt number by 10% to 15% than plain tube and for all Reynolds number the nail twisted tape for twist ratio 2 performs enhance in Nusselt number by 20% to 30%. Increment in the thermal performance factor was observed for nail twisted tape with twist ratio of 2 around 27% than the plain tube absorber.

Ravi Datt et al. [33] circular tube engaged with square wing and with solid ring twisted tape were investigated experimentally. Ring pitch ratio (d_R/D_T) of 0.5 to 2.0, wing pitch ratio (P_W/W_T) of 2.0 to 3.5, Wing Depth ratio (W_d/W_T) of 0.083 to 0.333, Ring diameter ratio (d_{iR}/ d_{oR}) of 0.831 to 0.932 and number of twisted tape of 1 to 4 were the geometrical parameters included for the experimentation. The outcome shows that the multiple square wings with solid ring twisted tape perform that the increase in rate of heat transfer of 5.66 times than the plain tube.Bipin Kumar et al. [34] Comparative study on the solid twisted tape (STT), perforated twisted tape (PTT) and Peripherally V cut with perforated twisted (VCT) were taken into investigation to analyze the effect on heat transfer and the pressure loss characteristics. The output shows that highest Nusselt number of 2.99 was attained for pitch ratio 1 in VCT.

From the literatures it was observed that many researchers experimented the presence of twisted tapes on heat transfer flow path to enhance the heat transfer in passive method. In all the experimentations the twisted tapes were placed in coaxial direction of the flow and experimented. No literatures reported about the presence of inserts in a transverse direction of the flow and its influence on heat transfer enhancement. This observation cultivated the interest towards present investigation to study the influence of presence of straight flat baffles placed in a transverse direction of the flow. An array of straight flat baffles mounted on a flat plate introduced in a transverse direction of the flow to investigate influence of the baffles on heat transfer enhancement.

Experimental Setup

The experimental set up shown in figure 1 consists of a rectangular channel to which the test section is fixed. The test section is an aluminium plate which has a length of 0.20m, width of 0.10m and thickness of the plate is measured as 0.005m. An array of straight flat baffles is bolted to the test section such that the baffles are in transverse direction of the air flow. The cross sectional width and height of the rectangular channel are 0.10m and 0.06m respectively and the effective length of the channel (i.e. distance between the inlet temperature and outlet temperature) is 0.5m. The array of baffles contains 20 straight flat baffles, each with height of 0.037m and width of 0.01m in

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a matrix of 5×4. Each baffle is connected to an individual K type thermocouple wire with an accuracy of 1°C, to measure the temperature of the baffle. Two cartridge heaters are used to supply the heat input to the test section. The test section is heated for power ranging from 50watts to 150watts with a regular interval of 25watts. A blower is connected to create the flow of air through the channel through an orifice pipe which has a coefficient of discharge of 0.62.

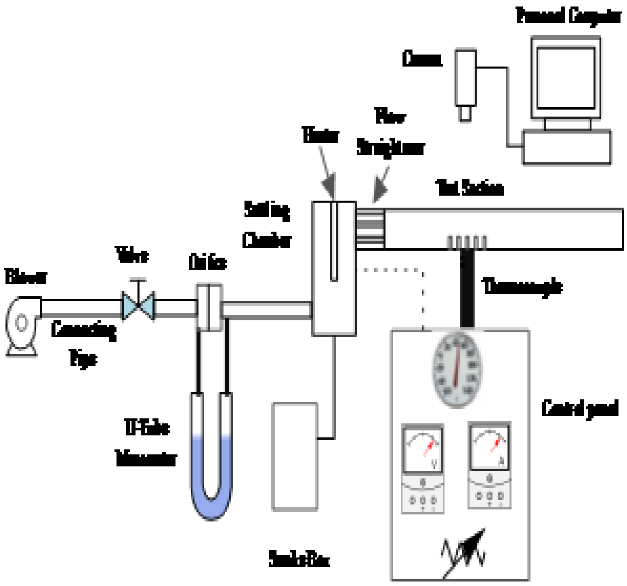

Figure 1. Schematic view of the Experimental set up

The control unit having Model Selec DTC 303 for temperature control and measuring with a full scale accuracy of ±0.25%, Model Selec MV 15 voltmeter ranges 0–480Vwith an full scale accuracy of ±0.5%, ammeter Model Selec MA 12 ranging from 0 - 20Awith an full scale accuracy of ±0.5% of and a autotransformer maximum 230V, 10A. K type thermocouples were used for measuring the temperature with a calibrated accuracy of ±01°C.Lutron AM-4201 Digital Anemometer with a resolution of 0.1m/s and an accuracy of ± 2 % was used to record the velocity of the flow through the rectangular channel.

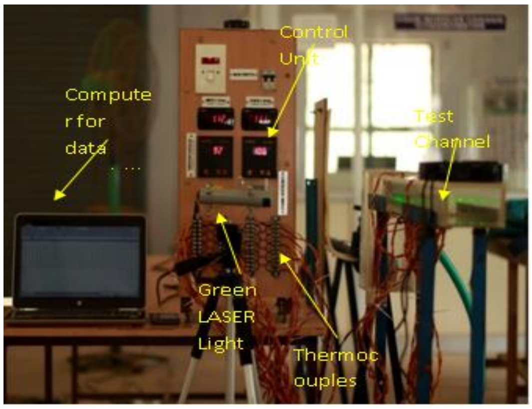

Figure 2. Photographic view of Experimental setup

Experimental Procedure

Heat transfer rate in a horizontal rectangular cross-sectional duct with straight flat baffles mounted on the test section was investigated experimentally under the internal flow turbulent situation. Experiment was carried out using air as the working fluid under the ambient condition. Five varying input power ranging from 50watts to 150 watts with a hiatus of 25watts were applied. Three varying flow rates of 1.92×10^{-3} m^3/sec, 2.674×10^{-3} m^3/sec and 3.269×10^{-3} m^3/sec were preserved till steady state is attained. Reynolds number of the flow and the velocity of the air is calculated using anemometer at four extreme corner of the rectangular channel and at the center position of the rectangular channel.

The temperature at the entry and the exit of the rectangular channel, temperature at every twisted tape were measured till it reaches the steady state condition for the 50Watt power input and manometer reading h = 20cm and 20.5cm. Temperature at each baffle was measured by fitting the temperature sensors at the bottom of the twisted tape. Identical method was implemented for the remaining heat inputs varying from 75watt to 150watt to measure the temperature at the entry and exit of the rectangular channel, temperature for all the baffles. For various Reynolds number and power inputs of 50W, 75W, 100W, 125W and

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150W experiment were conducted and readings were recorded.

Data reduction

Observed data were used to predict the required quantities using the equations (1) to (9). The uncertainties during the experimental; investigation were found by Kline and McClintock's method [35]. Slips for different measures were calculated by the recognised technique of estimation of investigational data. The uncertainties for selected measures from the experimentations were assessed as 2.02% for the Reynolds number, 0.71% for the input heat, 2.09% for the temperature, 2.61% for the coefficient of heat transfer and for the Nusselt number was found to be 4. 32%.

Orifice meter

$$a_1 = \frac{\pi}{4}(D)^2 \quad (1)$$

$$a_2 = \frac{\pi}{4}(d)^2 \quad (2)$$

Where, a_1= Area of the orifice at inlet (m^2), a_2= Area of the orifice (m^2), D= Diameter of the pipe (m), d= Diameter of the orifice (m)

Hydraulic diameter

$$D_h = \frac{4A}{P} \quad (3)$$

Where, D_h - Hydraulic diameter of the rectangular channel, A - Area of the cross section (m^2)

P - Perimeter (m)

Reynolds Number

$$Re = \frac{VD_h}{\upsilon} \quad (4)$$

Where, V = Velocity of the air in the channel (m/s), D_h = Hydraulic diameter (m), υ = kinematic viscosity (m^2/s)

Discharge rate

$$Q_{th} = \frac{a_1 a_2}{\sqrt{a_1^2 - a_2^2}}\sqrt{2gh_a}\ m^3/s \quad (5)$$

$$Q_a = C_d Q_{th}\ m^3/s \quad (6)$$

Where, Q_a = actual discharge (m^3/s), C_d = coefficient of discharge, for this experiments C_d = 0.62 was considered throughout the experiment, Q_{th} = theoretical discharge (m^3/s)

Heat transfer rate

$$Q_h = \dot{m}C_p\,(T_o - T_i) = \rho A V C_p\,(T_o - T_i) \quad (7)$$

Where, $\dot{m}$ = mass flow rate (kg/s), $\dot{m}$ =ρAV, C_p = specific heat (J/kgK), T_i = inlet temperature (°C), T_o = outlet temperature (°C)

The heat transfer coefficient is evaluated as

$$h = \frac{Q_h}{A * \Delta T} \quad (8)$$

Nusselt number was calculated using the following expression

$$Nu = \frac{hD_h}{k} \quad (9)$$

Where, Re = Reynolds number, Pr = Prandtl number

Where A, the total area is the sum of area of the rectangular channel and area of the baffles.

$$A = A_p + A_b(Area\ od\ the\ plate + n * Area\ of\ the\ Baffle) = 0.02037m^2$$

Temperature difference is given by $\Delta T = T_w - T_b$ and Bulk mean temperature (T_b) is calculated as follows $T_b = (T_o + T_i)/2$

Result and Discussion

Investigations were performed for various Reynolds number and three air flow situations to study the influence of the transverse flat baffles on the flow path in normal direction of the flow. Reynolds number considered for this study ranges from 4938 to 6478 and heat inputs ranges from 50watt to 150watt. Variation of influence of transverse flat baffles on Nusselt number, Pressure drop and friction factor with respect to the

Reynolds number and flow situations were reported.

Effect of heat input on Nusselt number and Convective heat transfer coefficient

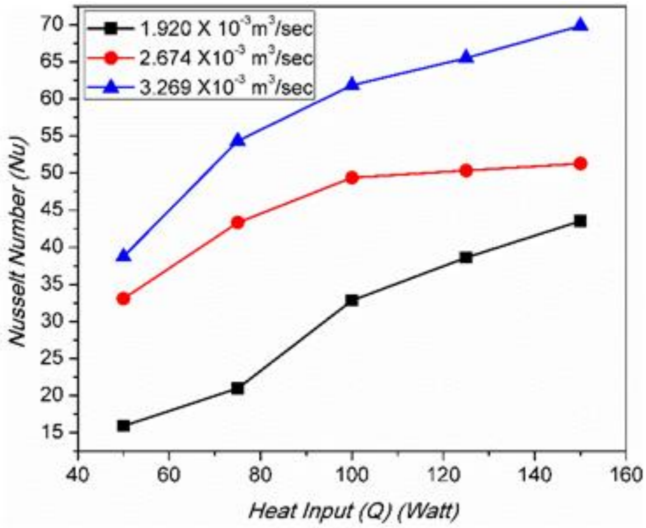

Figure 3. Relationship between Heat input and Nusselt number

Figure 3 depicts the relation between the heat input and Nusselt number for the three different flow situations. From the figure it was observed that the increase in Nusselt number was evident in increasing the heat input. It's a fact that the quantity of heat increases the potential difference of the temperature by virtue of which Nusselt number increases. At 50W heat input, while increasing the flow rate of air from 1.92×10^{-3} m^3/sec to 2.674×10^{-3} m^3/sec it was observed that 51.85% increase in Nusselt number and further increasing the flow rate to 3.269×10^{-3} m^3/sec, 14.63% raise in Nusselt was recorded. This increase in Nusselt number was due to the turbulence created by the transverse flat baffles provided on the flow path. This inserts suddenly creates a stagnation zone when the flowing air hits the baffles by virtue of which the agitation in flow takes place and enhances the heat transfer. Similarly for 150W heat input the raise in Nusselt number was predicted to be 37.68% while increasing the flow from 1.92×10^{-3} m^3/sec to 3.269×10^{-3} m^3/sec. From this increasing in Nusselt number it was disclosed that the presence of transverse flat baffles disturbs the flow in a significant way and responsible for the growth of the heat transfer.

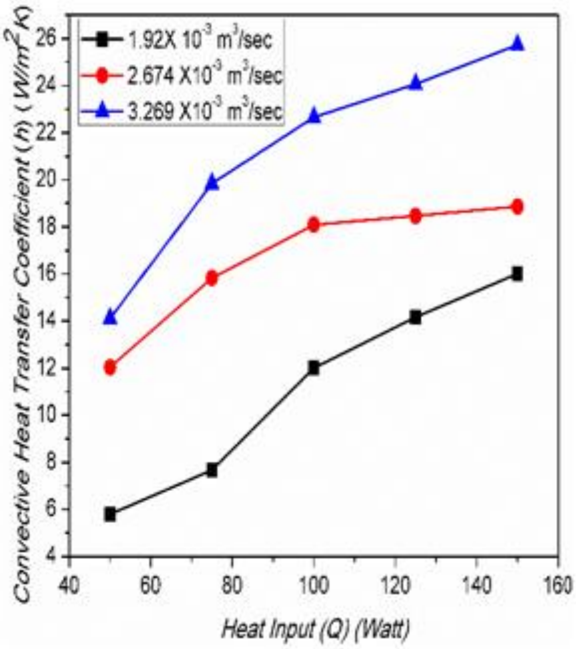

Figure 4. Relationship between Heat input and Convective heat transfer

The relationship between the range of heat input and convective heat transfer coefficient was disclosed in figure 4. It was clear that while increasing the heat input from 50W to 150W the convective heat transfer coefficient also increases. When the flow was maintained as 1.92×10^{-3} m^3/sec, the increase in convective coefficient was observed as 63.84% while changing the heat input from 50W to 150W. Similar trend was observed for other two flow situations also. The percentage increase in convective heat transfer coefficient was ranging from 36.15% to 45.17% during the increasing the heat input for 2.674×10^{-3} m^3/sec and 3.269×10^{-3} m^3/sec flow rates. This effect is recorded due to increasing trend in the temperature of the air at outlet of the channel. It is known fact that the heat input was directional proportional to the convective heat transfer. The presence of transverse flat baffles holds the heat and increases its temperature. This potential difference in temperature was responsible for the change and increase in convective heat transfer coefficient.

Effect of Inertia force on Nusselt number

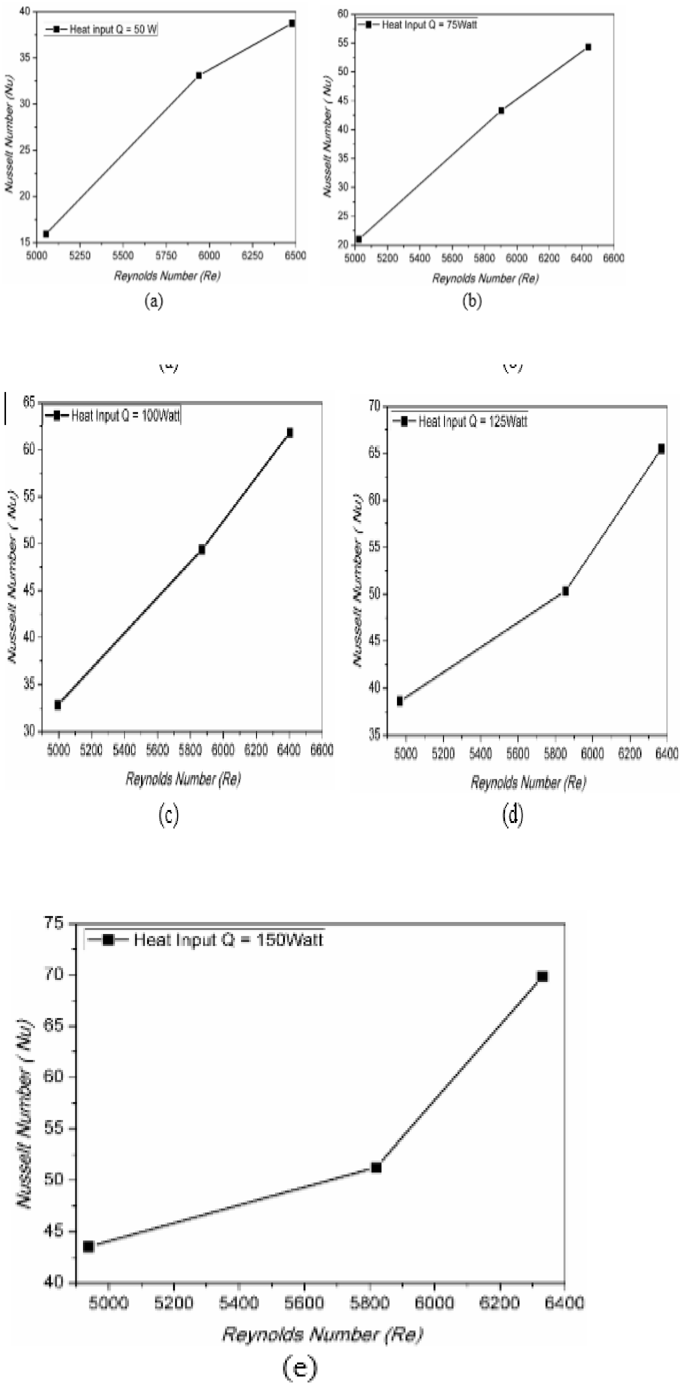

Figure 5. Reynolds number VsNusselt number for heat input 50W to 150W

The effect of inertia force on Nusselt number was shown in figure 5 and it portraits the relationship between the Reynolds number and Nusselt number for various heat inputs ranging from 50W to 150W. From figure 5(a) it was clear that there exist a linear relation relationship between the Reynolds number and Nusselt number. Nusselt number increases on increasing the Reynolds number for 50W heat input. Alike trend was witnessed on the figures 5(b) to 5(e) for heat inputs varying from 75W to 150W. The percentage increase in Nusselt number was predicted as 37.68% to 58.89% while changing the heat input from 50W to 150W. This growth in Nusselt number was expected due to the proper mixing of air inside the confined channel and the mixing was ensured by the availability of the transverse flat baffles.

Effect of mass flow rate on heat transfer

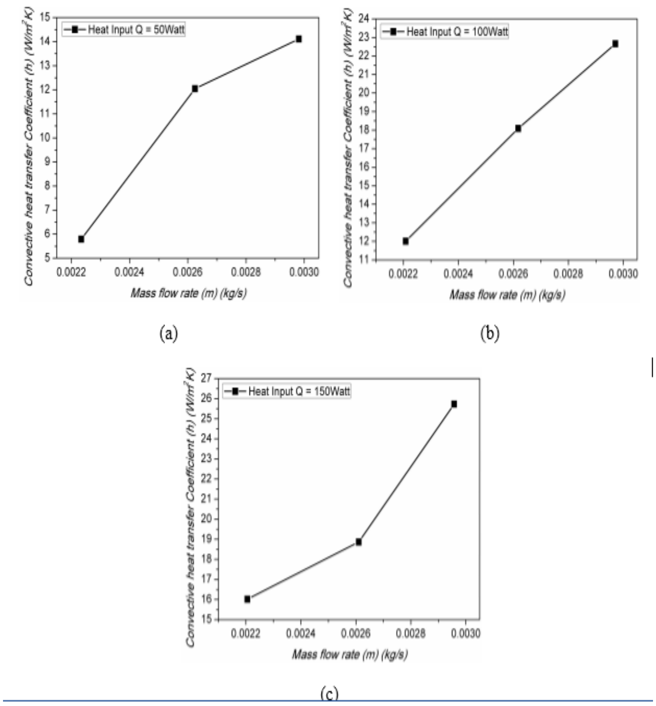

Figure 6. Mass flow rate Vs heat transfer coefficient for heat input 50W, 100W and 150W

The relationship between the mass flow rate and the convective heat transfer coefficient

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was depicted from figure 6. It was found that there exists a linear relationship between the mass flow rate and the convective heat transfer coefficient. On increasing the mass flow rate, heat transfer coefficient also increases by 58.95% at 50W heat input from figure 6(a). Similar trend was observed from figure 6(b) and 6(c) and 47%, 37.76% of increasing in heat transfer rate was predicted respectively while increasing the heat transfer from 100W to 150W. Decreasing in percentage of heat transfer coefficient was observed while changing the heat input from 50W to 150W was predicted due to no significant increase in temperature difference while increasing the mass flow rate. While increasing the mass flow rate, more amount of fluid was interacting with the flat baffles provided over the flow path in transverse direction of the flow.

Effect of flat baffle on Pressure drop

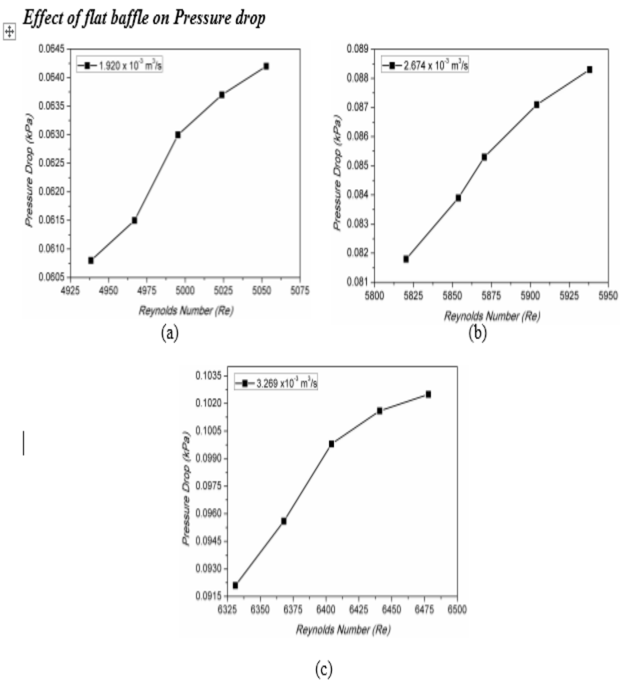
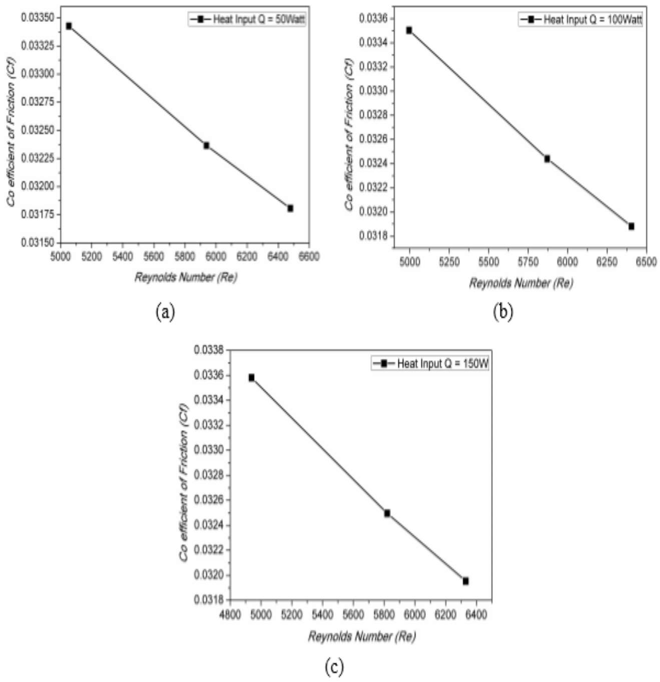


Figure 7. Reynolds Vs Pressure drop for three different flow rates

Figure 7 portraits the relationship between the Reynolds number and the pressure drop for three different flow rates. Figure 7(a) discloses that on increasing the Reynolds number the pressure drop also increases for 1.92×10^{-3} m^3/sec and the percentage increase in pressure drop was calculated as 0.789%. This drop in pressure was observed due to the sudden impact of flowing fluid on the transverse flat baffles availability on the flow path. Similar trend was found on the other flow rates also from figures 7(b) and 7(c). The percentage increase in pressure drop for flow rates 2.674×10^{-3} m^3/sec and 3.269×10^{-3} m^3/sec were predicted as 0.823% and 0.906% respectively. This insignificant drop in pressure in was recorded for test section length of 0.2m only. Significant pressure drop may be observed, if the pressure drop was measured at inlet and outlet of the confined channel. The length of the test section was not having a significant influence on the pressured drop and the obtained drop in pressure was recorded by virtue of the presence of the transverse flat baffles on the flow path. The array of flat baffles provided on the flow path in transverse direction of the flow make the flow disturbed significantly and produces turbulence on the flow. This local instability created in the flow inside the channel was responsible for the drop in pressure.

Effect of flat baffle on friction coefficient



The effect of transverse flat baffle placed on the flow path which influences the heat transfer was disclosed by figure 8 in which the relation between the Reynolds number and friction coefficient C_f was displayed. From figure 8(a) it was evident that on increase in Reynolds number the friction coefficient reduced significantly for 50W heat input. This was predicted due to the effect of inertia force which overcomes the viscous force while increasing the Reynolds number. Similar effect was found from the figure 8(b) and 8(c) for the heat input 100W and 150W. This significant change in friction coefficient was predicted due to the increase in surface area by the presence of flat baffles.

Conclusions

The experimental investigation to study the influence of flat baffle insert in a transverse direction of flow on heat transfer enhancement was investigated for three flow situation 1.92 x 10^{-3} m^3/s, 2.67 x 10^{-3} m^3/s and 3.269 x $10^{-3} m^3/s$. Reynolds number considered for the study was ranging from 4938 to 6478 and heat input ranges from 50W to 550W and revealed the following conclusions. Transverse flat baffles on the flow path normal to the direction of flow was significantly influences the augmentation of heat transfer when the flow varied from 1.92 x 10^{-3} m^3/s to 3.269 x $10^{-3} m^3/s$. The percentage increase in heat transfer was observed and which ranges from 38% to 52% while changing the heat input. Pressure drop recorded as less than 1% for various flow rates and it was evident that the presence of flat baffles on the flow path in transverse direction of the flow disturbs flow. No significant influence on the pressure drop was predicted due to the small length of the test section. The presence of flat baffles on the flow path in transverse direction of the flow significantly disturbs the flow and influences the friction coefficient and pressure drop in a trivial way.

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