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Heat Transfer and Friction Factor Characteristics in Twisted Square Duct with Inserts

M. Udaya Kumar
Research Scholar,
Department of Mechanical Engineering
Indur Institute of Engineering & Technology
Siddipet (T.S) [INDIA]
Email: iiet.uday@gmail.com

Md. Yousuf Ali
Professor
Department of Mechanical Engineering
Avanthi College of Engineering and Technology
Hyderabad (T.S) [INDIA]
Email: Required

ABSTRACT

The present work experimental and numerical investigations have been carried out to study the thermal performance characteristics of twisted square duct with inserts. Experiments were conducted for air with constant heat flux condition for the twist ratio of 6.12 and Reynolds number from 8500 to 35000. Inorder to reduce pressure drop and increase the thermal performance twisted square ducts are best suited. The results of friction factor, Nusselt number and thermal performance factor are presented. Experimental Nusselt number for twisted square duct with circular rod is about 2.52 times higher than plain square duct, while friction factor is 6.45 times higher (f/fo= 6.45). Thermal performance (enhancement ratio) of the twisted square duct with circular rod insert is 1.29. In this work Numeric simulations were calculated by using the CFD software package ANSYS 18.2 FLUENT has been used. Heat transfer and numerical flow behaviors such as temperature, friction factor contours of the twisted square duct and velocity profile are also reported. The results obtained in the study will help to design and development of heat exchanger

***Keywords**:— Heat transfer, Twisted square duct, Reynolds number, friction factor, Twist ratio, inserts*

I. INTRODUCTION

Ducts with non circular cross sections are widely used in heat exchangers and other devices. In many instances, designer is faced with existing equipment where the space occupied by the cooling passage is minimal and, the heat and mass flow rates are limited by the size of existing or retrofit pump or fan. In these situations, where cool an passage must be designed so that the volume of the passage is restricted to some value and the heat and mass flow rate of the coolant are dictated by the available equipment. In such cases, non circular duct might be the only option.

The great importance laws prevailing the transfer of heat and types of apparatus have for their main objects to control the heat flow. Generally, design of any industrial process plants is influenced by need for optimum utilization of energy as effective usage of heat for economic design and operation. The need for effective utilization and recovery of heat has been prompted the development of various heat transfer enhancement techniques for heat exchangers. Augmentation techniques increase convective heat transfer by reducing thermal resistance in heat exchanger.

Ievlev et al. [1] presented the results of an experimental study on heat transfer and hydraulic resistance of air flow inside twisted elliptic tube. Twist ratios used are in the range of 6.5–35. Reynolds number range covered was 3000 to 10,000. It was also shown that twisted tubes permit an appreciable increase in the heat transfer and a substantial reduction of the heat exchanger dimensions for the range of parameters studied. Wang et al. [2] experimentally and numerically studied three twisted square ducts with uniform cross section, divergent cross section and convergent cross section along the duct length. Twisted ratio used is 42. Experiments were conducted using air as working fluid under constant wall heat flux boundary conditions with Re range of 10,000–100,000. Thermal performance is compared with respect to straight square duct under constraints of identical mass flow rate, pumping power and pressure drop. It was concluded that for range of parameter studied divergent duct always enhance heat transfer, convergent duct always deteriorate heat transfer and straight duct may enhance or deteriorate performance depending upon comparison conditions.

Yang et al. [3] experimentally evaluated performance of five twisted elliptical tubes. Aspect ratio (major diameter/minor diameter) of elliptical tubes used was in the range of 1.49 to 2.15 and twist ratio range covered was 17.4–32.8. Water was used as the working fluid for Re range of 600–55,000 covering laminar, transition and turbulent regime. They concluded that for twisted duct flow remains laminar for Re 6 2300. In a twisted tube, the heat transfer enhancement is higher for laminar regime compared to transition and turbulent flow regimes. M. Khoshvaght et al [4] In this study spirally-coiled twisted-duct is introduced and analyzed both experimentally and numerically. The working fluids are water and water nano fluid. The maximum enhancement of Nusselt number and friction factor are recorded for the spirally-coiled twisted-ducts with the lowest twist-pitch is equal to 0.05 and coil-pitch is equal to 0.015. Maximum performance values are 1.88 and 1.33 are recorded of 1.88 and 1.39 for nanofluid and water. P. Samruaisin et al [5] Examined that pressure loss and heat transfer behaviors of tube integral with commonly spaced quadruple twisted tape elements were investigated under turbulent flow regime. Experiments were conducted at twist ratio of 2.5 under constant heat flux conditions. Over the range reported, the regularly spaced quadruple twisted tapes in cross-arrangement with $s/y = 0.5$ gives heat transfer rate up to 6.6% over than that of the quadruple twisted tapes and the maximum thermal enhancement factor of 1.27. S.Rashidi et al [6] reported that heat and fluid flow behaviors of square-cut twisted tape inserts under turbulent flow and constant heat flux conditions. The maximum thermal enhancement factor is 1.37 at the largest perforated width to tape width ratio is equal to nine. Yuxiang Hong et.al [7] the effect of twin overlapped twisted tapes in a spiral grooved tube under turbulent flow with Reynolds number varied from 8000 to 22000 using air as working fluid. The performance evaluation criterion is approximately 1.05–1.14 for spiral grooved tube. OrhanKeklikcioglu and VeySelOzceyhan [8] studied that an experimental investigation was carried out of a circular tube inserted with wire coil. These inserts had same length placed 1mm and 2mm distance from the inner wall tube with various pitch ratios equal to 1, 2, 3. Tests were conducted for Reynolds numbers from 3429 to 26663. The maximum thermal performance was achieved 1.82 for Reynolds number 3429. This is due to laminar boundary layer disturbance. Sompol

Skullong et al. [9] Effect of insertion of delta-wing tape used as vortex generator The experiment is conducted for turbulent airflow with the Reynolds number from 4200 to 25,500. The delta wings are in a forward-wing arrangement with three wing inclination angles (α=30°, 45° and 60°) and with five ratios of wing-pitch to tube-diameter =0.5, 1.0, 1.5, 2.0 and 2.5. The results indicate that the delta wing tape provides maximum thermal enhancement factor value is 1.49. The 60° delta wing tape with pitch ratio 0.5 yields the highest Nusselt number and friction factor but the 30° one with pitch ratio =1.0 gives the best thermal performance factor Amnart Boonloi [10] The numerical investigations are carried out in a square channel fitted with discrete V-baffle and V-orifice. The improvement of the heat transfer rate is around 2.8–6 times higher than the smooth channel depended on blockage ratio V-tip directions and Reynolds number. Pongjet Promvong et al. [11] studied that an experimental research on thermal performance improvement in a constant heat-fluxed square ducts built-in winglet vortex generators and twisted tapes. The inserted duct at blockage ratio= 0.2, pitch ratio = 2 and twist ratio= 4 provides the highest heat transfer rate and friction factor but the one at blockage ratio = 0.1, pitch ratio = 2 and twist ratio = 4 yields the highest thermal performance. The application of combined vortex-flow devices gives thermal performance around 17% higher than the twisted tape alone. Rambir Bhadouriya et al. [12] reported that friction factor and heat transfer characteristics of air flow inside twisted square duct under uniform wall temperature conditions with twist ratio of 11.5 and 16.5, Reynolds number varied from 600-70000. The results show significant improvement in pressure drop and heat transfer in laminar and turbulent flow regimes till Reynolds number of 9500. Twist ratio of 11.5 shows quite higher pressure drop and heat transfer compared to plain square duct. Nihal UĞURLUBİLEK et.al [13] according to this study, heat transfer and turbulent flow characteristics through twisted square duct has been numerically investigated. The working fluid is considered as water and the Reynolds number range between10000-1200000. The governing equations implied using the commercial code FLUENT. It has been observed that Twisted square duct provides significant increase in terms of Nusselt number to 138 % over the plain square duct and utmost gain of 1.3 on thermal performance factor is obtained for the case of Reynolds number 10000. This represents the secondary flow occurred through the twisted square duct can increase the heat transfer rate. The edge size of square cross-section, the twist angle and the length of the channel are taken 0.01m, 360^{o} and 0.2 m, respectively.

II. Experimental Setup

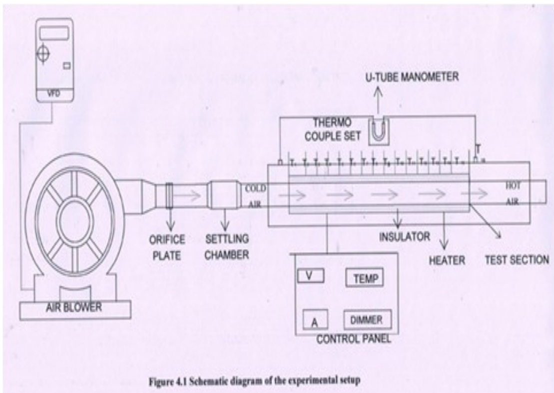

Figure 1: Schamatic diagram of the experimental setup

An experimental setup has been designed and fabricated to study the heat transfer and friction factor characteristics of air under laminar and turbulent flow in the twisted square duct. The twisted square duct is made of Aluminum material of length 1.5m, has a cross section 65 × 65mm, twist ratio 6.12, Suction system a 1kW blower was used to receive the air from the room to the test section. The complete tested segment was heated by nichrome heater wire giving constant heat flux conditions. The duct is controlled by a variance transformer to supply controllable electrical heating to the test duct. The duct is controlled by a variance transformer to supply controllable electrical heating to the test duct. n order to maintain isothermal heating condition and with high accuracy, wall temperature distributions were measured over the heated part with 14 thermocouples with 0.1°C resolution. Wood bars with low thermal conductivity, were fitted around the square duct to function as thermal barriers at the inlet and exit of the test section. The outer surface of the test duct was well insulated to lessen All thermocouples are type K and diameter of the wire is 1.5 mm. The thermocouple voltage outputs were fed into a data acquisition system (dtalogger) and then recorded via a personal computer.

After ensuring steady state condition, record inlet, outlet and surface temperature readings velocity flow of air, mass flow of air, pressure difference at constant heat flux conditions. Based on these readings calculate Reynolds number, Nusselt number, Heat transfer coefficient, results of the twisted square duct have been investigated.

III. NOMENCLATURE

A=Convection heat transfer area of duct, m^2

AR=Aspect ratio of duct, (W/H)

Cp=Specific heat capacity of air, J/kgK

D_h =Hydraulic diameter of duct, (H), m

Fo=Friction factor for plain duct

f=Friction factor for twisted square duct

H= Duct height, m

h =Heat transfer coefficient, W/m^2K

I =Current, A

K=Thermal conductivity of air, W/mK

L= Length of test duct, m

M=Mass flow rate of air, kg/s

Nuo =Nusselt number of plain duct,

Nu =Nusselt number of Twisted square duct

Pr =Prandtl number

Re =Reynolds number, (UD/ν) (dimensionless)

Q =Heat transfer, W

q =heat flux, W/m^2

T =Temperature, K

T_i =Air inlet temperature, 0^C or K

T_o =Air outlet temperature, 0^C or K

T_b =Bulk temperature, $(T_i+T_o)/2$ 0^C or K

T_s=Surface temperature, 0^C or K

U =Mean velocity, m/s

V= Voltage, V

W =Width of the duct, m

Y=Twist ratio, dimensionless (s/D), m

S=Pitch of the twisted tube, m

Greek letters

E=Turbulent kinetic energy dissipation rate

ϑ =Kinematic Viscosity, m^2/sec

θ=Twist angle

ρa=Density of air, kg/m^3

η=Thermal performance factor (enhancement ratio)

τ =Shear stress

μ =Dynamic viscosity, kg s^{-1} m^{-1}

t_w= twisted square duct

IV. Data Reduction

The present experiment was conducted to investigate the heat transfer augmentation in a twisted square Duct with twist ratio 6.12. The present experimental results on the heat transfer and friction characteristics in a square duct are first analyzed in terms of Nusselt number and friction factor.

The results obtained are displayed in dimensionless terms of Nusselt number and friction factor. The heat transfer coefficients are calculated by using the experimental data through the following equations:

$h=Q/A(T_s-T_b)$

$Q=m.c_p(T_o-T_i)$

Mass flow rate $m = \rho AV$

In which $T_b=(T_0+T_i)/2$

Where, A is the heat transfer surface area of duct, Ts is the average surface temperature.

Thus, Nusselt number is written as

$Nu=hD_h/k$

The Reynolds number based on the duct hydraulic diameter (D_h) is given by

$Re =UD_h/\nu$

Experimental friction factor f_e=

$$\frac{\Delta p * De}{2 * L * \rho_a * V^2}$$

Where, Δp is the pressure drop across the test duct and U is the mean air velocity in the duct. All properties of air are evaluated at the overall bulk air temperature.

V. Results and Discussion

Validation of smooth square duct:

The experimental results of Nusselt number and friction factor of the present plain Square duct are compared with those from correlations of Dittus–Boelter, Blasius and Petukhov found for turbulent flow in ducts.

Correlation of Dittus-Boelter,

$Nu = 0:023Re^{0:8}Pr^{0:4}$ for heating (1)

Correlation of Petukhov,

$f= 0.79\ (\ln Re-1:64)^{-2}$(2)

The comparison of Nusselt number and friction factor obtained from the present plain square duct with those from correlations of Equations. (1), (2) are represented.

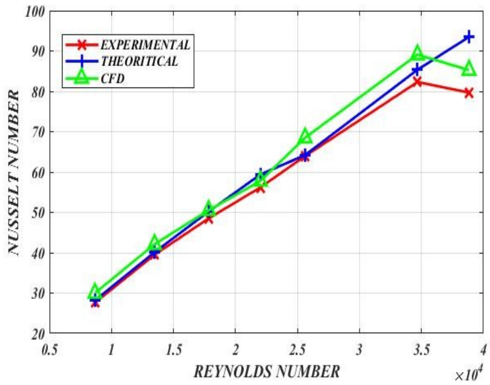

Figure 2: Data verification of Nusselt number versus Reynolds number for plain square duct.

The graph represents variation of Nusselt number with Reynolds number for plain square duct using air is test fluid The experimental values were compare with standard equation. It was observed that Percentage deviation between experimental and theoretical values is ±5.1. Therefore the experimental setup is deemed to be validated. Increasing nusselt number which in turns to increased heat transfer coefficient.

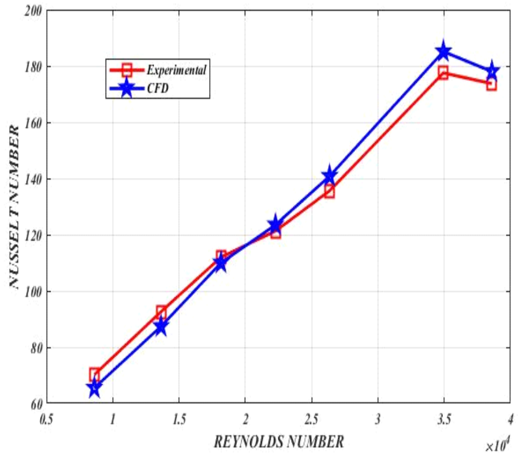

Figure 3: Comparison of Experimental and numerical results for Twisted square duct. with insert

It has been observed that deviation between experimental and CFD Values within the permissible limit. Twisted square duct with insert creates more turbulence. Therefore Nusselt number increases, which leads to increased heat transfer coefficient.

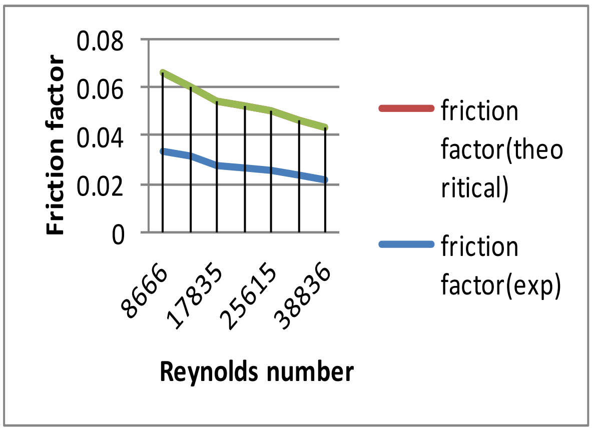

Figure 4: Comparision of Experimental and Theoritical values for plain square duct

The above diagram represents friction factor decreases while Reynolds number increases. This is due to the velocity of air flow is increased gradually. It leads to increased pressure drop. Hence friction factor is increased. The difference between experimental and theoretical values is less than 6 percentages.

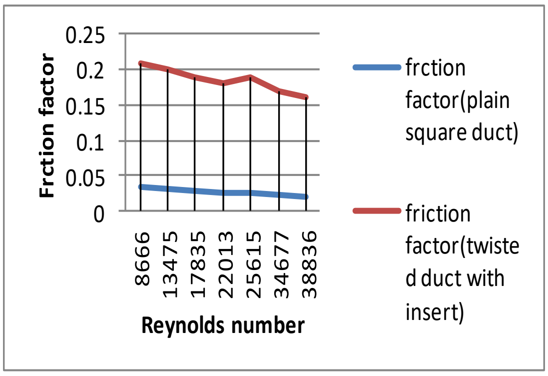

Figure 5: Comparision of Friction Factor Values for Plain Ductand Twisted with Inserts

The above diagram represents comparison between plain and twisted duct with insert. Friction factor for twisted square duct with insert is increased this is due to insert creates high turbulence. It leads to high pressure drop.

CFD Analysis:

Twisted square duct with circular rod baffle

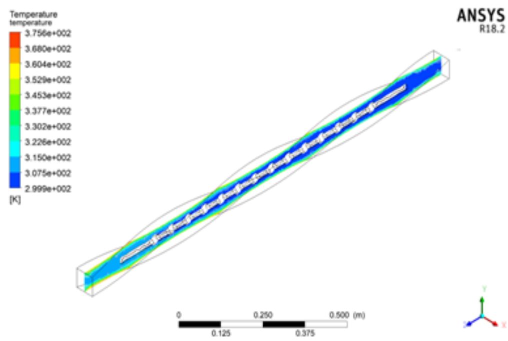

Figure 6: Temperature Contour of Twisted sqaure duct with circular rod at velocity of 2.2 m/sec

The above diagram represents temperature distribution of twisted square duct with circular rod baffle. It has been observed that the effect of twisted with baffle creates turbulence which leads to increases heat transfer rate. Simultaneously increase heat transfer coefficient also.

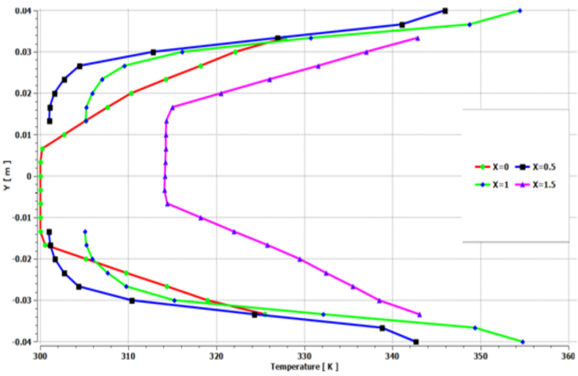

Figure 7: Temperature profile of Twisted sqaure duct with circular rod at velocity of 2.2 m/sec

In the above diagram at X=0.5 m and 1 m and along the Y direction 12.5 mm from above and below of the axis line there is no heating takes place. At X=0 and 1.5m there will be heating takes place through the duct. It is clearly observed that at X=1m the inlet and outlet temperature difference ix maximum. This is due to proper mixing of fluid in the duct.

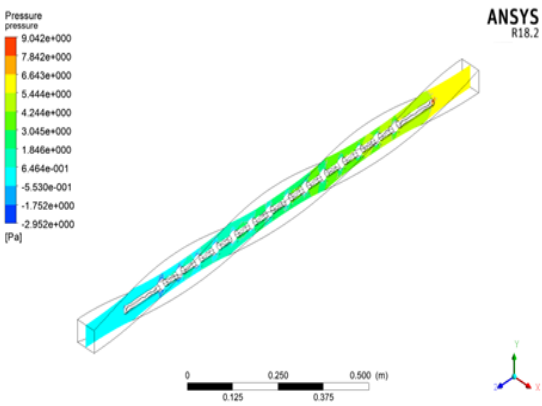

Figure 8: Pressure Contour of Twisted Square Duct with Circular rod at Velocity of 2.2 m/sec

The above diagram represents pressure drop for twisted square dcut with circular rod various pitch ratios have been presented and results are obtained. These are indicated that pressure drop is higher as compared to the plain duct. This is due to large amount of obstruction of flow passage is takes place by inserting circular rods in the twisted square duct.

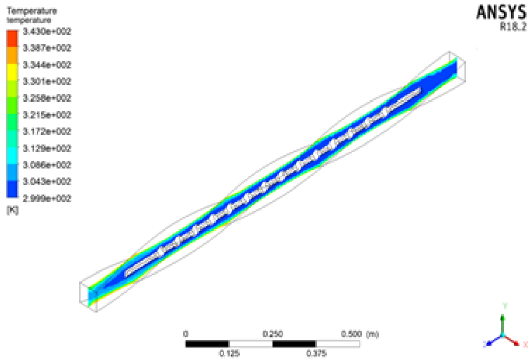

Figure 9: Temperature contour of Twisted square duct with circular rod at velocity of 4.5 m/sec

The above diagram represents the effect of temperature profile of twisted square duct with circular rod inserts has been presented. It is observed that heating of the duct is slightly decreased compard to plain square duct. This is due to aluminum circular rod inset absorbs certain amount of heat and distribution of heat to the surrounding areas is minimum. Therefore the temperature difference between inlet to outlet is decreased as compared plain square duct.

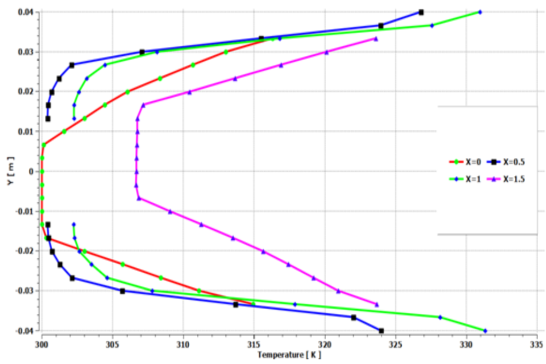

Figure 10: Temperature profile of Twisted square duct with cirular rod at velocity of 4.5 m/sec

The above diagram represents temperature profile at various distances through the duct. At X=0.0.5 meters distance from axis line 12.5 mm distance above and bellow there is no heating takes place during the region. At X=1.5 m there will be heating takes place, and the temperature difference is slightly decreased as compared to at velocity of 2.2 m/sec.

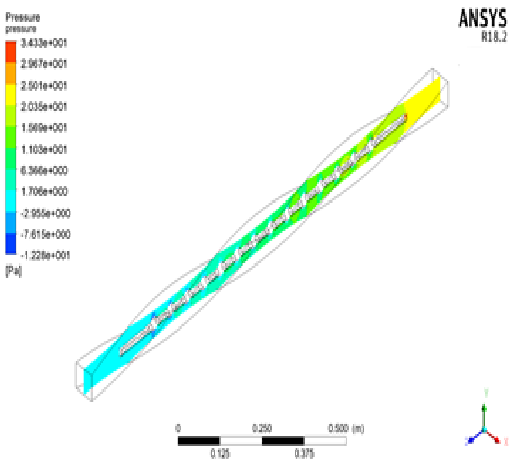

Figure 11: pressure contour of Twisted square duct with circular rod at velocity of 4.5 m/sec

The above diagram represents pressure drop for twisted with circular rod insert. It is visible that pressure drop is higher due to circular rod creates air flow in zig-zag manner. It leads to increases frictional losses. Therefore the performance of the duct is decreased.

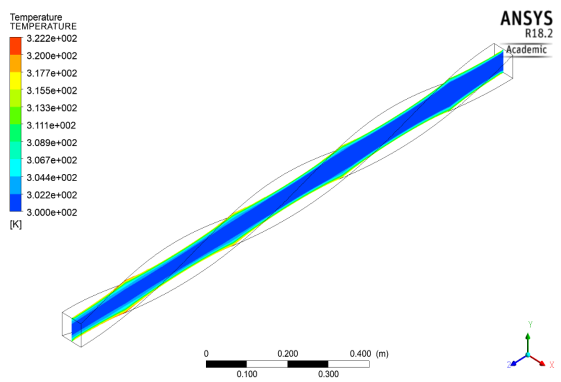

Figure 12: Temperature contour of Twisted square duct with circular rod at velocity of 8.6 m/sec.

The above diagram represents temperature distribution of twisted with circular rod insert as shown in Figure 12. It is visible that the difference between inlet and outlet temperature is gradually reduced This is because if the velocity is increased there will be chances to formation and stagnation of air flow molecules are less in the duct. This leads to decreases the temperature difference between entry and exit.

VI. CONCLUSIONS

Following main conclusions are drawn from these investigations

The heat transfer coefficient increases with Reynolds number in the twisted square duct insert and it was observed maximum at a velocity of 8.6 m/sec This is due to higher swirling flow or turbulence long residence time in the duct.

The experimental Nusselt number for twisted square duct is maximum at velocity 8.6 m/sec i.e. 114.2 under constant heat flux condition with air as test fluid.

The experimental results shows that twisted duct with insert (Nu/Nuo= 2.52) i.e. Nusselt number for the twisted square duct is about 2.52 times above that for the plain square duct while friction factor is 6.45 times higher (f/fo= 6.45).

Thermal performance (enhancement ratio) of the twisted square duct with insert is 1.29 Finally it is concluded that Twisted square duct with insert perform better heat transfer enhancement than plain square duct.

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