Journal of Advances in Science and Technology
Vol. 12, Issue No. 25, (Special Issue) December-2016, ISSN 2230-9659

Experimental Investigation of Forced Convection Heat Transfer through a Square Duct Provided with Different Configurations of V-Shaped Rib Turbulators

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Abstract – This research focuses on the results of an experimental investigation of heat transfer to the air flowing through a square duct. The enhancement in the heat transfer is achieved by roughening the vertical walls of the square duct using different configurations of V-shaped rib turbulators made of aluminium material. The walls of this duct are uniformly heated by using band heater. The effect of different configurations of V-shaped rib turbulators on the heat transfer coefficient and friction factor is investigated at different flow rates of air and at different heat inputs. The system parameters and operating parameters are varied and the investigation is done within the limits, as power input 150-600 W, velocity of air flow from 2.5-5 m/s, against the variation of Reynolds number: 18500-36000. It is observed that the duct provided with 60° parallel continuous V-shaped rib turbulator shows optimum performance. Also the experimental values of heat transfer coefficient and friction factor are compared with the ones obtained from correlations.

Keywords: Square Duct; Heat Transfer Enhancement; V-Shaped Rib Turbulators; Nusselt Number; Turbulence; Overall Enhancement Ratio

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1. INTRODUCTION

Heat transfer enhancement by introducing artificial roughness on the surfaces of the duct is one of the most conventional methods used to enhance the efficiency of a gas turbine. Generally power output and efficiency of turbine is improved by increasing the temperature of gas at the inlet. This can be achieved by roughening the internal cooling passages of the turbine blades, which leads to enhance the heat transfer performance with optimum pressure drop. In most of the modern gas turbines the application of inserts to enhance heat transfer is common thing. The roughening of the surfaces by the use of inserts improves the turbulence in the fluid flow near the wall surfaces of the duct and leads to creation of obstruction to the laminar boundary layer to improve the heat transfer rate. Also provision of inserts increases the heat transfer area.

The thermo hydraulic performance of the ribs is governed by many parameters such as the geometry which includes shape, size, angle and spacing. A number of investigations have been carried out on heat transfer enhancement in a rectangular or square duct provided with different types of inserts. Han [1] experimentally studied effect of V-shaped broken rib on local heat transfer and pressure drop in a square duct with two opposite walls provided with ribs. Ray [2] numerically investigated heat transfer performance through square duct with twisted tape insert. Tanda [3] carried out an experimental investigation of Heat transfer in rectangular channels with transverse and V-shaped broken ribs. Maximum heat transfer is attained for 60 degree V-shaped ribs. Sahu [4] investigated the effect of ribs on heat transfer coefficient and thermal efficiency of solar air heater. It is found that, heat transfer coefficient is enhanced by 1.30 times and maximum thermal efficiency obtained as 83.5% than that of smooth duct. Karmare [5] experimentally investigate the effect of metal grit ribs on heat transfer in the rectangular duct. SK Saha [6] carried out an experimental study of heat transfer and pressure drop of laminar flow of viscous oil flowing through non circular ducts roughened with transverse ribs on two opposite surfaces of the ducts. Wei [7] carried out experimental and numerical study of convection heat transfer in a channel provided with 90° ribs and V-shaped ribs. Giovanni Tanda [8] carried out the experimental study to determine the effect of angled rib turbulator, inclined at 45° on forced convection heat transfer in a rectangular channel. M. Udaya [9] performed an experimental study to investigate the effect of twisted tape insert on heat transfer enhancement in square duct. Yongsiri [10] numerically investigated the effect of the inclined detached ribs with different attack angles on thermal performance behaviors. Priyank [11] performed a two-dimensional CFD investigation to study forced convection heat transfer in a rectangular duct provided with inserts on the underside of the top wall. The results of numerical investigation were found in accordance with the theoretical correlations.

2. EXPERIMENTAL SYSTEM AND DATA REDUCTION

2.1 Experimental Setup

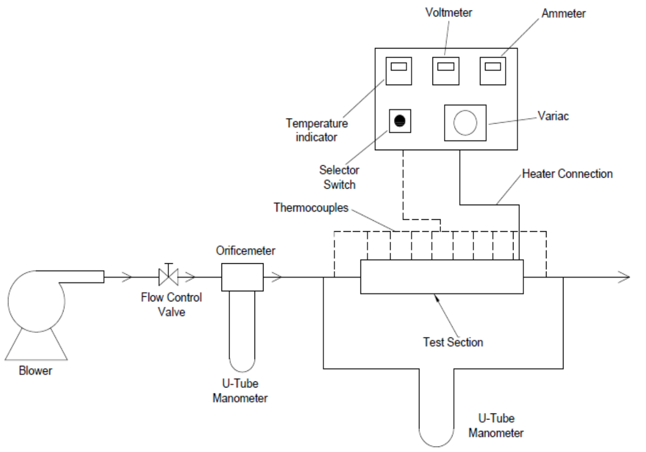

Fig. 1. Schematic of Experimental System

Schematic diagram of experimental set-up for investigation of forced convection heat transfer in a square duct provided with different configurations of V-shaped rib turbulators is as shown in fig. 1. The test set-up is an open loop air flow system which comprises of centrifugal blower, flow control valve, manometers, test section, control panel, etc. The vertical walls of the square duct are provided with the different configurations of continuous and broken V-shaped rib turbulators as per requirement. The square duct is surrounded by band heater for heating purpose. Eight thermocouples are embedded on the test section and two thermocouples are placed in the air stream at the entrance and exit of the test section to measure air inlet and outlet temperature. The temperatures at different locations can be read directly from the temperature indicator by using selector switch. The outer surface of the test section was well insulated to minimize heat loss to surrounding. Air flow is controlled by a flow control valve and is measured with the help of Pitot tube and water manometer. Heat input can be set with the help of variac provided on control panel and same can be read out digitally with the help of voltmeter and ammeter. One more manometer is connected across the test section to measure the pressure drop occurred during the flow through test section.

The actual experimental setup is as shown in Fig. 2.

Fig. 2. Actual Image of Experimental Setup

The different configurations of V-shaped rib turbulators that are to be provided on the two opposite vertical walls of the square duct are as shown in fig. 3.

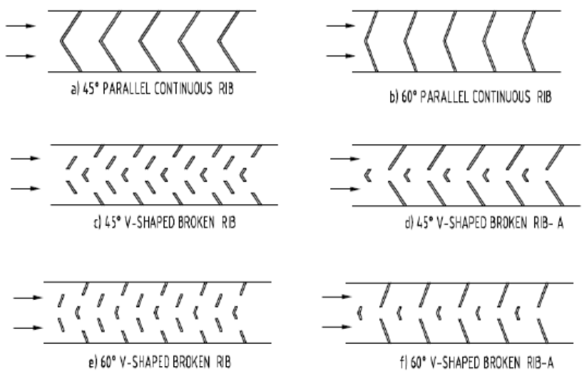

Fig. 3. Different V-shaped rib turbulator configurations

2.2 Experimental Procedure:

The test section is prepared for given configurations of V-shaped rib turbulators. The blower was switched on after keeping the valve open at desired rate to allow the airflow through the duct. Initially the experiment was carried out for plain duct. Then the experiment was carried out for different configurations of V-shaped rib turbulators having inclination angle of 45° and 60° with the direction of flow of air. A constant heat flux is applied to the test section by adjusting the voltage to desired value. The temperatures in the direction of flow of air are found out with the help of temperature selector switch to which thermocouples placed on the test section are attached. Note down the voltmeter and ammeter readings. Four values of flow rates were used for each set at fixed uniform heat flux. Similarly for each set four values of uniform heat flux were used for given flow rate. At each value of flow rate and the corresponding heat flux, system was allowed to attain a steady state before the temperature values; voltmeter & ammeter readings were recorded. The pressure drop across the orifice meter and test section was also measured when steady state was reached. Repeat the procedure for different heat inputs and flow rates.

During experimentation manometer reading to calculate flow rate of air, temperatures at different locations of the heated surface of square duct and temperatures of air at inlet and outlet of the test section and pressure drop across the given test section are measured.

2.3 Data Reduction:

All the parameters displayed on control panel which includes all the temperatures, voltage and current values are noted down. Manometer readings giving pressure drop across the test section and flow rate are noted down. Average temperature of duct wall and bulk temperature of air are calculated. Properties of air like thermal conductivity, kinematic viscosity, Prandtl number from the air table corresponding to above bulk temperature of air are determined. Then mass flow rate and velocity of air flowing through the test section are determined. Then convective heat transferred to air, convective heat transfer coefficient is calculated. Experimental Nusselt number and friction factor, Reynolds number are calculated. Overall enhancement ratio is determined by calculating the values of Nusselt number and friction factor for square provided with inserts and for plain square duct. Theoretical Nusselt number and friction factor values are obtained by using following correlations:

Dittus-Boelter equation (N_{uthe}) to determine Nusselt number:

$N_{uthe} = 0.023 \times Re^{0.8} \times Pr^{0.4}$

Gnielinski equation to determine friction factor:

$f_{the} = [0.79 \times \ln (R_e)-1.64]^{-2}$

3. RESULTS AND DISCUSSION

3.1 Effect of Inserts on Heat Transfer:

Fig. 4a & 4b shows the variation of experimental Nusselt number with Reynolds number for square duct with all types of inserts in comparison with plain square duct at 150W & 300W input respectively. From the above figures it is concluded that as Reynolds number increases the Nusselt number also increases. The highest value of Nusselt number is 460.4960 for square duct provided with 45° Type-C rib inserts at Reynolds number 36570.1. It is evident from fig. 4a and 4b that when square duct is provided with different types of inserts there is significant improvement in Nusselt number because of turbulence induced due to inserts and secondary flow. Blockages induces secondary flow which leads to swirl generation. This causes mixing of fluid which leads to increase in temperature gradient, which ultimately leads to enhancement in the heat transfer coefficient. From figures, the percentage increase in Nusselt number for square duct with inserts compared to plain square duct is in the range of 3% to about 32%. For square duct with inserts the heat transfer coefficient varies from 2.3% to 29% times the plain duct but the corresponding friction factor increases by 1.33 to 2.5 times the plain square duct values.

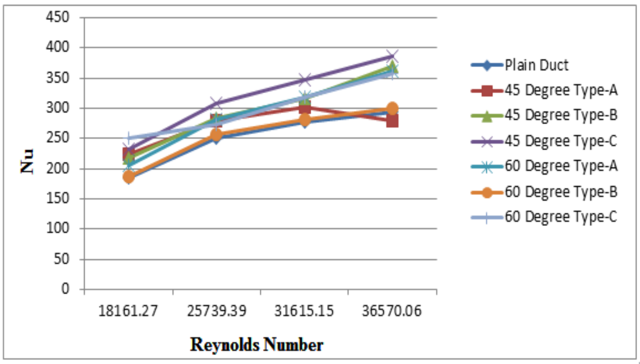

Fig. 4a. Experimental Nusselt Number Vs Reynolds Number at 150 W Input

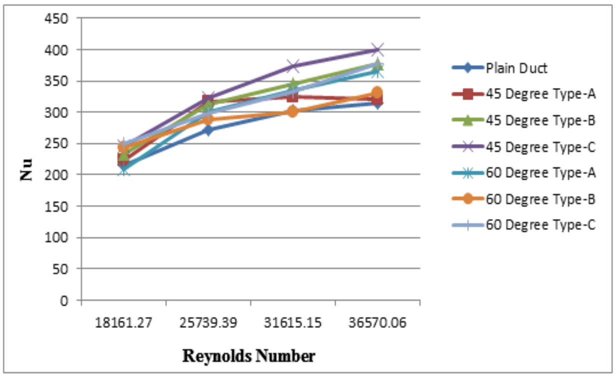

Fig. 4b. Experimental Nusselt Number Vs Reynolds Number at 300 W Input

3.2 Effect of Inserts on Friction factor:

Friction factor is the measure of pressure loss in the system to the kinetic energy of the fluid. Fig. 5a and 5b shows the variation of friction factor with Reynolds number for plain square duct and square duct fitted with different types of inserts at 150W and 300W respectively. From the above figures it is concluded that as the Reynolds number increases the friction factor decreases. The value of friction factor varies from 0.03048 to 0.12192 for Reynolds number in the range of 18161.27 to 36570.06 respectively.

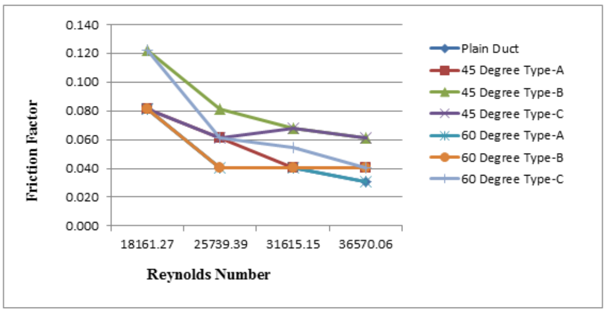

Fig. 5a. Experimental Friction Factor Vs Reynolds Number at 150 W Input

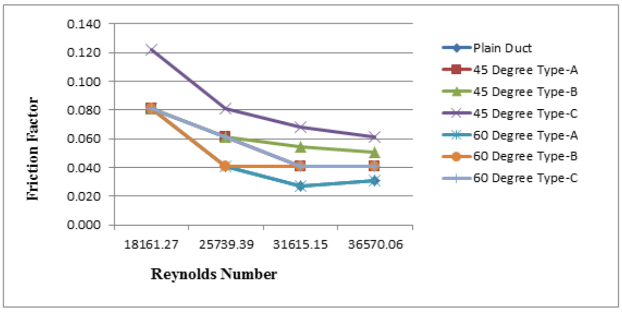

Fig. 5b. Experimental Friction Factor Vs Reynolds Number at 300 W Input

3.3 Effect of Inserts on Nusselt number ratio:

Fig. 6a and 6b shows the variation of Nusselt number ratio for different Reynolds numbers at different heat inputs for square duct provided with different type of inserts. It is observed that in most of the cases the Nusselt number ratio increases with increase in Reynolds number but however in some cases it decreases.

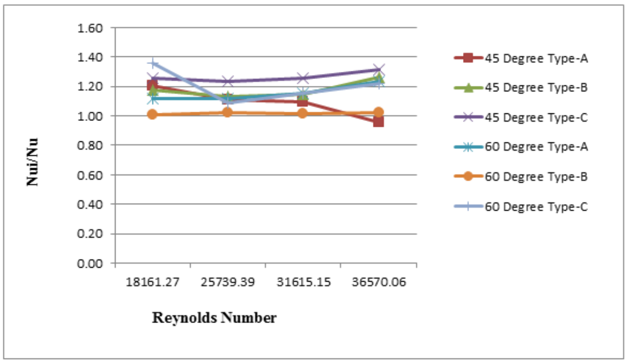

Fig. 6a. Nusselt Number Ratio Vs Reynolds Number at 150 W Input

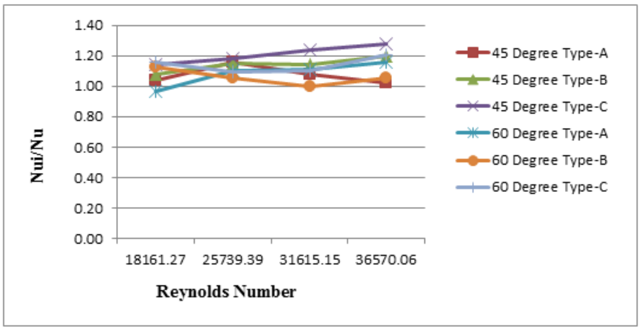

Fig. 6b. Nusselt Number Ratio Vs Reynolds Number at 300 W Input

3.4 Effect of Inserts on Nusselt number ratio:

Fig. 7a and 7b shows the variation of Friction factor ratio for different Reynolds numbers at different heat inputs for square duct provided with different type of inserts. It is observed that in most of the cases the friction factor ratio shows zigzag variation with increase in Reynolds number but however in some cases there is not much significant variation in friction factor ratio.

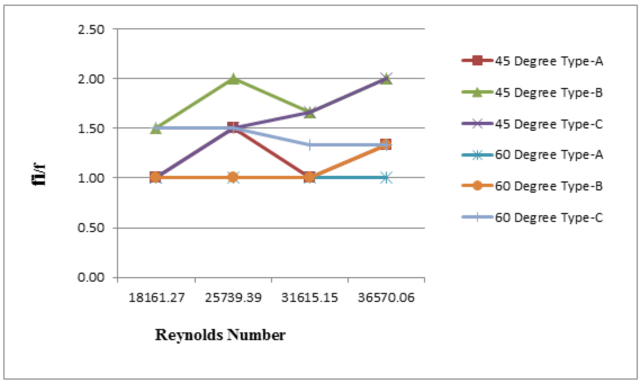

Fig. 7a. Friction Factor Ratio Vs Reynolds Number at 150 W Input

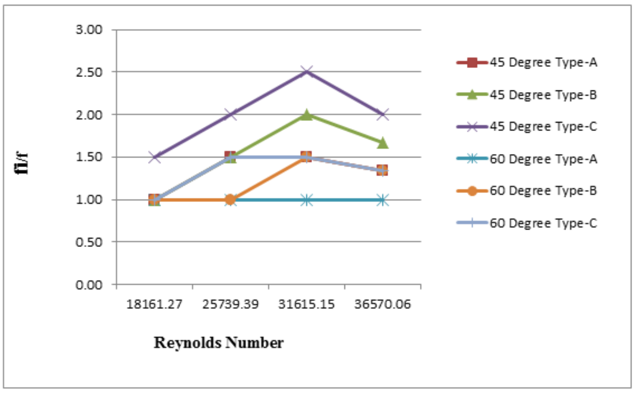

Fig. 7b. Friction Factor Ratio Vs Reynolds Number at 300 W Input

3.5 Effect of Inserts on Overall Enhancement ratio:

Fig. 8a and 8b shows the variation of overall enhancement ratio for different Reynolds numbers at different heat inputs for square duct provided with different type of inserts. The overall enhancement ratio is defined as the ratio of the heat transfer enhancement ratio to the friction factor ratio. This parameter is also used to compare different passive techniques and enables a comparison of two different methods for the same pressure drop. The friction factor is a measure of head loss or pumping power. For a particular Reynolds number, the thermo hydraulic performance of an insert is said to be good if the heat transfer coefficient increases significantly with a minimum increase in friction factor. Thermo hydraulic performance estimation is generally used to compare the performance of different inserts under a particular fluid flow condition. From fig. 8a and 8b, it is observed that in most of cases of square duct provided with different types of inserts, the overall enhancement ratio is in the range of 1.05 to 1.25. In some cases the value of overall enhancement ratio is less than 1. This is due to the fact that pressure drop occurred due to turbulence in those cases is more than the heat transfer enhancement. Among the different types of inserts the optimum results are obtained for square duct provided with 60° parallel continuous rib turbulator inserts.

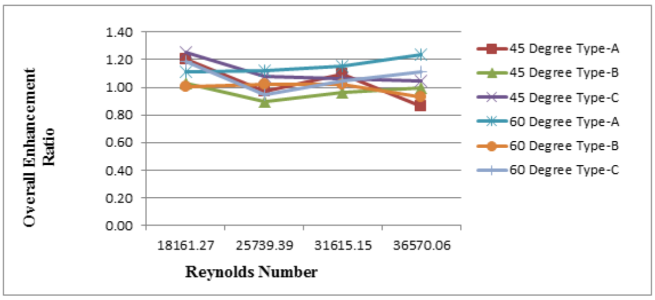

Fig. 8a. Overall Enhancement Ratio Vs Reynolds Number at 150 W Input

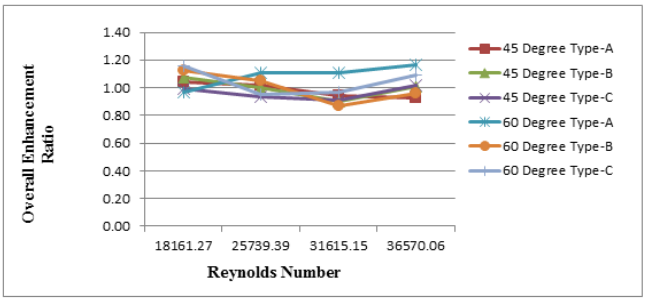

Fig. 8b. Overall Enhancement Ratio Vs Reynolds Number at 300 W Input

4. CONCLUSIONS:

The experimental investigation of forced convection heat transfer in a square duct provided with different configurations of V-shaped rib turbulators has been carried out.

The effect of different types of V-shaped rib turbulators at various Reynolds numbers on the heat transfer coefficient and friction factor is studied. The Reynolds number during the experimentation was varied from 18000 to 36500.

Following conclusions have been drawn from the experimentation:

1. The Nusselt number increases with increase in Reynolds number at different heat inputs for square duct provided with different types of inserts.

2. In most of cases of square duct provided with different types of inserts, the overall enhancement ratio is in the range of 1.05 to 1.25.

3. The square duct provided with 45° Type-C shows highest Nusselt number among all the types of V-shaped rib turbulators for different Reynolds numbers and at different heat inputs. However the pressure drop is also more for this insert as compared to remaining V-shaped rib turbulators because of more turbulence created during the flow.

4. The square duct provided with 60° Type-A inserts shows higher overall enhancement ratio for all the heat inputs because of higher degree of turbulence created.

5. The percentage increase in Nusselt number for square duct with inserts compared to plain square duct is in the range of 3% to 32%.

6. For square duct with inserts the heat transfer coefficient varies from 2.3% to 29% times the plain duct but the corresponding friction factor increases by 1.33 to 2.5 times the plain square duct values.

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