Augmented Heat Transfer in Square Duct with 90° ARC of Circle and V Shape RIB

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Abstract— In the advanced gas turbines of today, the turbine inlet temperature can be as high as 1500°C; however, this temperature exceeds the melting temperature of the metal airfoils. Therefore, it is imperative that the blades and vanes are cooled, so they can withstand these extreme temperatures. Cooling air around 650°C is extracted from the compressor and passes through the airfoils. With the hot gases and cooling air, the temperature of the blades can be lowered to approximately 1000°C, which is permissible for reliable operation of the engine. In order to avoid premature failure, designers must accurately predict the local heat transfer coefficients and local airfoil metal temperatures. By preventing local hot spots, the life of the turbine blades and vanes will increase. In the diameter present work, the effect of the rib angle orientation on the local heat transfer distribution and pressure drop in a square channel with two opposite in- line ribbed walls was investigated for Reynolds numbers from 45000 to 75000. The objective of the present work is to study the effect of rib height to the hydraulic ratio (e/Dh) on the local heat transfer distributions and flow friction in ribbed square channel with 90° arc of circle rib and V shape of circle ribs. Rib geometries, comprising three rib height-to-channel hydraulic diameters (blockage ratio) of 4.482, 6.75, and 9.018 as well as rib spacing (pitch to height ratio) is 10. A square channel, roughened with different blockage ratio ribs on two opposite walls in in-line manner and perpendicular to flow direction was tested. An experimental result shows that 90 degree arc of circle rib has maximum HTC as compared to V shape of circle rib with maximum pressure drop penalty It is observed that the heat transfer augmentations in the channel with 90° arc of circle attached ribs increase with increase in the rib height to hydraulic diameter ratio (e/Dh) but only at the cost of the pressure drop across the test section.

Key words: Heat Transfer Enhancement, 90° Arc of Circle Rib, V Shape of Circle RIB, Augmentation, Square Channel, Nusselt Number

I. Introduction

The use of artificial roughness in the form of repeated ribs has been found to be an efficient method of enhancing the heat transfer to fluid flowing in the duct. Detailed information about the heat transfer and flow characteristics in ribbed ducts is very important in designing cooling systems of gas turbine engines. The application of artificial roughness in the form of fine wires and ribs of different shapes has been recommended to enhance the heat transfer coefficient by several investigators. Gas turbines play a vital role in the today's industrialized society, and as the demands for power increase, the power output and thermal efficiency of gas turbines must also increase. One method of increasing both the power output and thermal efficiency of the engine is to increase the temperature of the gas entering the turbine. This temperature exceeds the melting temperature of the metal airfoils. Therefore, it is imperative that the blades and vanes are cooled, so they can withstand these extreme temperatures. Various cooling methods have been developed over the years to ensure that the turbine blade metal temperatures are maintained at a level consistent with airfoil design life. One such method is to route coolant air through rib-roughened serpentine passages. The rib geometry and arrangement in the channel also alter the flow field resulting in different convective heat transfer distribution. In particular, the angled ribs, the rib cross-section, the rib-to-channel height ratio and the rib pitch-to-height ratio are all parameters that influence both the convective heat transfer coefficient and the overall thermal performance.

Several investigations have been carried out to study the effect of these parameters of ribs on heat transfer and friction factor for two opposite roughened surfaces. Han et al. [1] studied experimentally the heat transfer in a square channel with ribs on two walls for nine different rib configurations. Average heat transfer and friction factor were reported for (P / e) =10 and e/Dh=0.0625. They reported that the angled ribs and 'V' ribs yield higher heat transfer enhancement than the continuous ribs. The heat transfer augmentations and the friction factor were highest for the 60° orientation amongst the angled ribs. Han et al. [2] also investigated the influence of the surface heat flux ratio on the heat transfer in a square ribbed channel with e/H=0.063 and (P / e) =10, by heating either only one of the ribbed walls or both of them, or all four channel walls. They reported that the former two conditions resulted in an increase in the heat transfer with respect to the latter one and the average Nusselt number tends to decrease for increasing Reynolds numbers and the thermal boundary condition becomes less relevant at higher Reynolds number. Han and Zhang [3] reported the heat transfer augmentation in a square channel with seven different configurations of broken ribs and found that 60° broken 'V' ribs provide higher heat transfer at about 4.5 times the smooth channel and perform better than the continuous ribs. The experiments were conducted on two wall ribbed channels with e/D=0.0625 and (P / e) =10 for all. Taslim et al. [4] conducted measurements of the heat transfer in a straight square channel with three e/H ratios (e/H=0.083, 0.125 and 0.167) and a fixed (P / e) =10 using a liquid crystal technique. Various staggered rib configurations were studied, especially for the angle of 45°. Experimental data showed a significant increase in average Nusselt number for the increase of the e/H ratio. The best value of the e/H ratio found to lie between 0.083 and 0.125 was reported. Lau et al. [5] examined the turbulent heat transfer and frictional characteristics in a square channel with V-shaped ribs. Thermocouples were used to measure the wall temperature. Various angles of attack of the V-shaped rib arrays were studied and the 60° V shaped ribs with a pitch to rib height ratio (P / e) of 10 was found to have highest channel heat transfer per unit pumping power. Han [6] studied experimentally heat transfer and friction in channels with two opposite rib-roughened walls with the effect of pitch –to-height and rib-to-equivalent diameter. He reported that staton number of ribbed side wall is 1.5 to 2.2 times that of the smooth duct.

Thus, the main aim of the present work is to extend the experimental data available on various rib shapes with similar e/Dh ratio of 0.083, 0.125 and 0.167 mounted on a square channel. These results provide insight into improved designs of internal passages for gas turbine. Experimental results using air as the test fluid from two different ribs are presented in turbulent channel flows in a range of Reynolds number from 45000 to 75000.

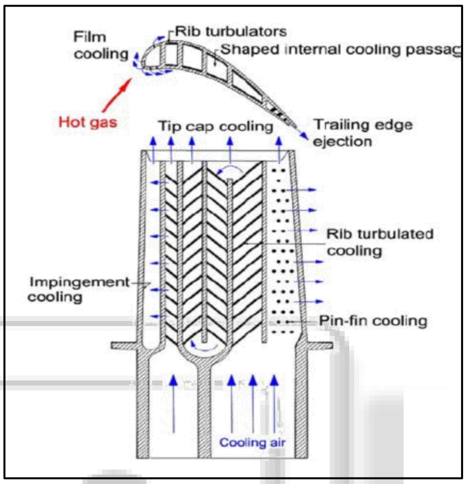

Fig. 1: Gas Turbine Cooling

II. EXPERIMENTAL SETUP

A schematic diagram of the experimental set up is presented in Fig. 1 while the details of two rib shapes mounted in in-line arrays on the square duct used in the heat transfer experiments are depicted in Fig. 2 respectively. In Fig. 1, a circular pipe was used for connecting a high-pressure blower to a settling tank, which an orifice flow meter was mounted in this pipeline. The overall length of the channel is 2000 mm. Each of the ribbed walls was fabricated from 6 mm thick aluminum plates, 54 mm wide and 650 mm long (L). Ribs with blockage ratio are 0.167, 0.125and 0.083 mm. The channel test section consisted of the two parallel walls, the principal walls. The AC power supply was supplied for the plate-type heater, used for heating the both plate of the test section, only to maintain uniform surface heat flux. Wood bars, which have a much lower thermal conductivity than the metallic wall, were used for square duct to form thermal barrier. Air as the tested fluid in both the heat transfer and pressure drop experiments, was directed into the systems by a 7.5HP blower. Control valve is used to provide desired air flow rates. The flow rate of air in the systems was measured by an orifice plate. The pressure across the orifice was measured using U tube manometer. In order to measure temperature distributions on the principal upper wall, nine thermocouples were fitted to the wall.

The thermocouples were installed in holes drilled from the side face and centered of the walls with the respective junctions positioned within 27 mm of the side wall and axial separation was 65 mm apart. To measure the inlet bulk temperature, one thermocouples were positioned upstream of duct inlet. All thermocouples were K type, 0.5 mm diameter wire. Two static pressure taps were located at the top of the principal channel to measure axial pressure drops across the test section, used to evaluate average friction factor. These were located at the center line of the channel. One of these taps is 770mm downstream from the leading edge of the channel and the other is 40 mm upstream from the trailing edge. The pressure drop was measured by inclined manometer. To quantify the uncertainties of measurements the reduced data obtained experimentally were determined.

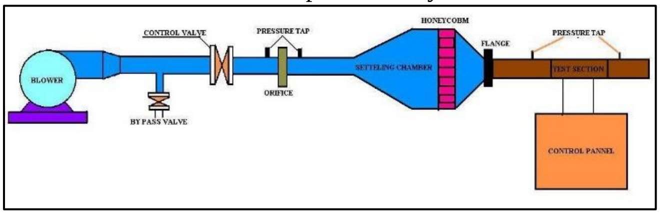

Fig -2: Experimental Setup

III. DATA REDUCTION

The average heat transfer coefficients are evaluated from the measured temperatures and heat inputs. With heat added uniformly to fluid (Qair) and the temperature difference of wall and fluid (Tw−Tb), average heat transfer coefficient will be evaluated from the experimental data via the following equations [1-8]

$$Qair = Q_{conv} = m \times Cp \times (T_o - T_i) = V \times I \quad (1)$$

$$h = Q_{conv} / (A \times (T_w - T_b)) \quad (2)$$

In which,

$$T_b = (T_i + T_o) / 2 \quad (3)$$

And,

$$T_w = (\sum T_s / 9) \quad (4)$$

Then, average Nusselt number is written as:

$$Nu = (hD_h / k) \quad (5)$$

The friction factor is evaluated by:

$$f = (2 \times \Delta P \times D_h) / (\rho \times l \times V^2) \quad (6)$$

Where, ΔP is a pressure drop across the test section and V is mean air velocity of the channel. The thermal enhancement factor, η, defined as the ratio of the, h of an augmented surface to that of a smooth surface, h0, at a constant pumping power:

$$\eta = (Nu/Nu_s) / (f/f_s)^{(1/3)} \quad (7)$$

IV. RESULTS & DISCUSSION

A. *Verification of smooth channel*

The present experimental results on heat transfer and friction characteristics in a smooth wall channel are first validated in terms of Nusselt number and friction factor. The Nusselt number and friction factor obtained from the present smooth channel are, respectively, compared with the correlations of Dittus–Boelter and Blasius found in the open literature [1-8] for turbulent flow in ducts.

Correlation of Dittus–Boelter,

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.4} \quad (8)$$

Correlation of Blasius,

$$f = 0.042 Re^{-0.2} \quad (9)$$

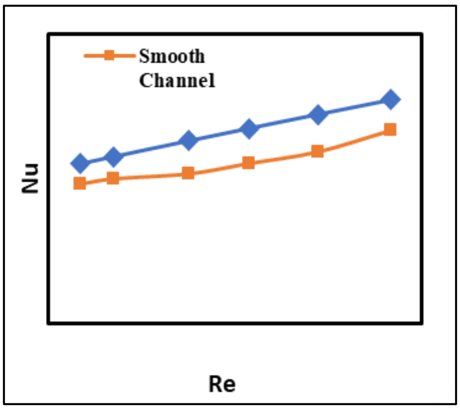

Fig.4 Variation of nusselt number v/s Reynolds number for smooth duct

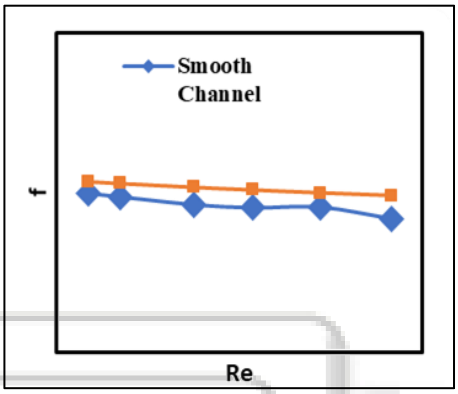

Fig.5 variation of friction factor v/s Reynolds number for smooth duct

Fig. 4 and 5 shows, respectively, a comparison of Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (8) and (9). In the figures, the present results reasonably agree well within ±20% for both friction factor correlation of Blasius and Nusselt number correlation of Dittus–Boelter.

B. *Effect of rib geometry*

The present experimental results on heat and flow friction characteristics in a uniform heat flux channel equipped with three blokage ratio are presented in the form of Nusselt number and friction factor. The Nusselt numbers obtained under turbulent flow conditions for all Blockage ratio turbulators with only one rib pitch (P=10 mm) are presented in Fig. 6. Nusselt number increases with the rise of Reynolds number.

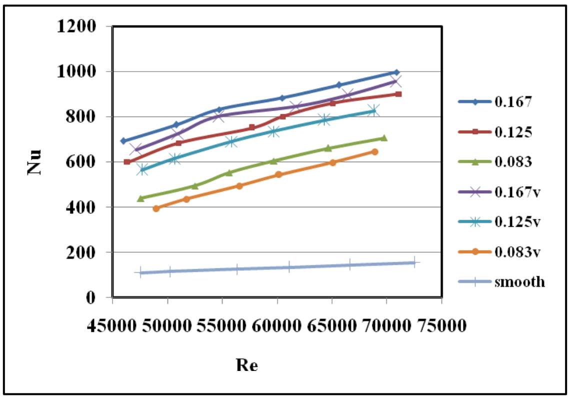

Fig. 6: Variation of Nusselt number v/s Reynolds number for blockage ratio

It is worth noting that the 90 degree arc of circle rib with blockage ratio 0.167 provides the highest value of Nusselt number, the increase in Nusselt number value is about 655% over the smooth channel.

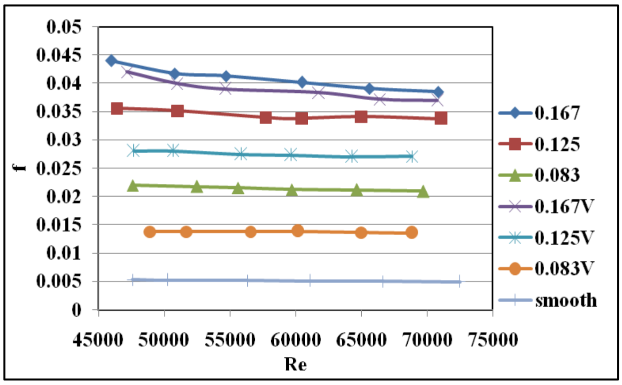

Fig. 7: Variation of Friction factor v/s Reynolds number for blockage ratio\

In the figure 7, the variation of the pressure drop is shown in terms of friction factor with Reynolds number. It is apparent that the use of rib turbulators leads to a substantial increase in friction factor over the smooth channel. The increase in friction factor for rib turbulators is considerably higher than that for the smooth channel. This can be attributed to flow blockage, higher surface area and the act caused by the reverse flow. As expected, the friction factor obtained from the 90 degree arc of circle rib is substantially higher than that from the V-shape of circle rib. The friction factor value of the 90 degree arc of circle rib with e/Dh=0.167 is found to be higher than that with the V-shape of circle rib with e/Dh=0.167 around 58–69%. The losses mainly come from the dissipation of the dynamical pressure of the air due to high viscous losses near the wall, to the extra forces exerted by reverse flow and to higher friction of increasing surface area and the blockage because of the presence of the ribs.

C. *Performance evaluation*

The Nusselt number ratio, Nu/Nu_0, defined as a ratio of augmented Nusselt number to Nusselt number of smooth channel. In the figure8, the Nusselt number ratio tends to decrease with the rise of Reynolds number for all turbulators. It is interesting to note that at higher Reynolds number, the Nu/Nu_0 value of the 90 degree arc of circle rib with 0.083 is nearly the same as V-shape of circle rib with 0.083one.

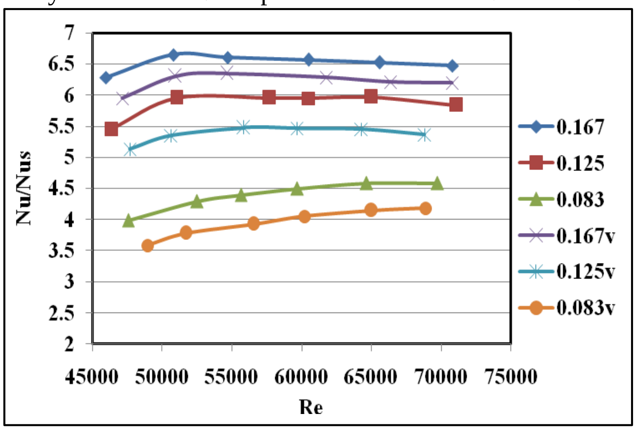

Fig. 8: Variation of Nusselt number ratio v/s Reynolds number for blockage ratio

The average Nu/Nu_0 values for the 90 degree arc of circle rib and V-shape of circle rib, respectively, around 6.5, 5.9, 4.3 and 5.5, 4.6, 3.4 for blockage ratio 0.167,0.125and 0.083.

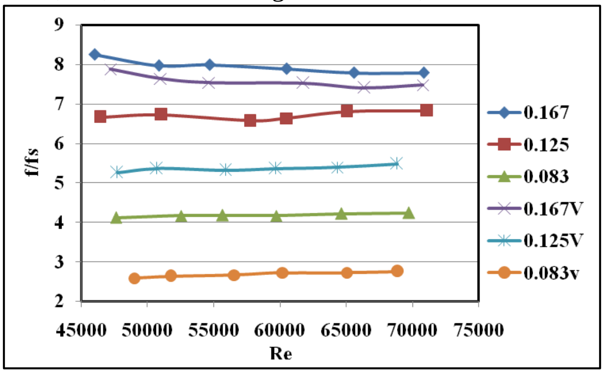

Fig. 9: Variation of Nusselt number ratio v/s Reynolds number for blockage ratio

Fig. 9 presents the variation of the friction factor ratio, f/f_0, with the Reynolds number value. It is observed that the friction factor ratio tends to increase with raising the Reynolds number. The mean f/f_0 values for the 90 degree arc of circle rib and V-shape of circle rib, respectively, about 7.9, 6.7, 4.1 and 5.5, 4.6, 3.4 for blockage ratio 0.167, 0.125and 0.083.

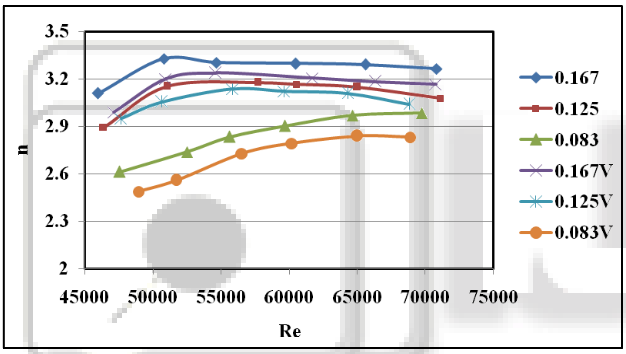

Fig. 10: Thermal Performance v/s Reynolds number

Fig. 10 shows the variation of the thermal enhancement factor (η) with Reynolds number for all rib turbulators. For all, the data obtained by measured Nusselt number and friction factor values are compared at a similar pumping power It is seen in the figure that the enhancement factors (η) generally in between 2 to 4 for all ribs at very high Reynolds number values only. The enhancement factor tends to decrease with the rise of Reynolds number values for all turbulators. It is worth noting that the enhancement factors (η) of the 90 degree arc of circle rib are higher than those with the V-shape of circle rib all Reynolds number values. The enhancement factor (η) of the 90 degree arc of circle ribs is found to be the best among all turbulators and is about 3.1 at the lowest value of Reynolds number while that of V-shape of circle rib is slightly lower.

V. Conclusion

Experimental study has been carried out to investigate air flow friction and heat transfer characteristics in a square channel fitted with different rib turbulators for the turbulent regime, Reynolds number of 45000 to 75,000. The use of the rib turbulators with e/Dh=0.167 causes a very high-pressure drop increase, and also provides considerable heat transfer augmentations, Nu/Nu_0=3.4-6.5, and f/fs=3.4-7.9, depending on rib geometry. Nusselt number augmentation tends to increase with the rise of Reynolds number. The 90 degree arc of circle rib should be applied instead of the V-shape of circle rib to obtain higher heat transfer and thermal performance.

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