

Experimental analysis of heat transfer performance on different ribs shapes in forced convection

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ABSTRACT

The heat transfer and friction loss performance of rib roughened rectangular cooling channel having rib of shapes a)Square ribs ,b)House shaped ribs and c)Boot shaped ribs is investigated experimentally by varying the pitch of ribs to 20,40 and 50mm for different rib geometries and varying the flow regime between Reynolds number 12000-32000. Ribs are a popular heat transfer augmentation device used in various heat-exchanging channels such as the internal cooling channels in gas turbine blades. The flow disturbance caused by the rib arrays greatly increases the production of turbulent kinetic energy, which enhances turbulent heat transfer in the channel. Different ribs shapes have been experimentally studied to find the optimum shape for higher heat transfer rate and corresponding friction losses. Compared to square and house shaped ribs ,Boot shaped ribs has been found to have highest heat transfer rate and moderate friction losses similar to square ribs whereas the House shaped ribs showed the worst heat transfer rate compared to other two. Comparisons in terms of Nusselt number, friction factor and Reynolds number is presented in this paper.

Keywords—Square Ribs, House Shaped Ribs, Boot Shaped Ribs.

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I. INTRODUCTION

The efficiency and power output of a gas turbine increases with higher turbine inlet gas temperature. Modern gas turbine vanes and blades are exposed to gas with temperatures which far exceeds the melting point of the component material. Thus, the blades and vanes have to be cooled in order to lower the temperature. When cooling the component it is important to know the correct boundary conditions, to avoid creating too large temperature gradients. Large temperature gradients cause thermal stresses and significantly decrease the component life. Blades and vanes are cooled by internal channels, through which the cooling air flows in different schemes and configurations. The heat transfer coefficient is increased by enhancement of the flow

turbulence and by breaking the flow boundary layer. The penalty paid for the increased heat transfer is higher pressure loss. Here insertion of ribs into passages comes to an account. In principle, ribs disturb the boundary layer due to flow separation and reattachment. The boundary layer separates induced by the upstream rib, forms a separated shear layer, and eventually reattaches on the wall. A reversed flow boundary layer originates at the reattachment point and grows in thickness towards the upstream rib. After reattachment, the flow starts to redevelop until it approaches the next rib subsequently another separation induced by the downstream rib occurs. For this reason, the local heat transfer in the separated-reattached flow is larger than those of an un-disturbed boundary layer.

However, the heat transfer enhancement in the separated-reattached flow is typically accompanied by an increase in the fluid pressure drop and pumping power. Accordingly, the identification of the rib geometry features is fundamental, to obtain the best heat transfer performance by considering both heat and momentum transfer characteristics.

II. EXPERIMENT APPARATUS AND PROCEDURE

A) Experimental Setup

A schematic diagram of the experimental apparatus is presented in Fig. 1 while the details of ribs mounted in rectangular channel are depicted in Fig. 2a,b and c, respectively. In Fig. 1, a circular pipe was used for connecting a 1 HP high-pressure blower where orifice flow meter was mounted in this pipeline while the channel including the test section was employed following to the outlet pipe. In Fig. 2, the channel geometry is characterized by the channel height, H and the axial length of cycle or pitch, P, the respective values of which are 20 mm, 30 mm and 40 mm. The overall length of the channel is 1150 mm which with the channel width, W, of 100 mm and height 25 mm. Each of the ribbed plate was fabricated from 6 mm thick aluminum plates, 100 mm wide and 600 mm long (L). The uniform rib dimensions are 5 mm high and 5 mm wide for square ribs, and other shapes dimensions can be seen in Fig. 2 (b) and (c) respectively. The channel test section is composed of the two parallel walls as shown in Fig. 2, the principal walls. The AC power supply was the source of power for the plate-type heaters, used for heating of the test section to maintain uniform surface heat flux. Asbestos rope and glass wool was considered over the entire test section to minimize the heat loss through convection and radiation. Air as the tested fluid in both the heat transfer and pressure drop experiments, was directed into the systems by a 1 HP high-pressure blower. The operating speed of the blower was varied by using a flow control valve provides desired airflow rates. The pressure across the orifice was measured using U tube manometer. In order to measure temperature distributions on the principal wall, eight thermocouples were fitted in the test plate while one at inlet and two thermocouples at the outlet. The thermocouples were installed in holes drilled at the two sides along the length at a distance 100 mm on one side and 150 mm on other as shown in Fig 2(d). All thermocouples for measuring the tube surface temperatures were K type, 1.5 mm diameter wire. The values measured by the thermocouples were noted through temperature indicator attached to them. 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. The pressure drop was measured by a inclined manometer.

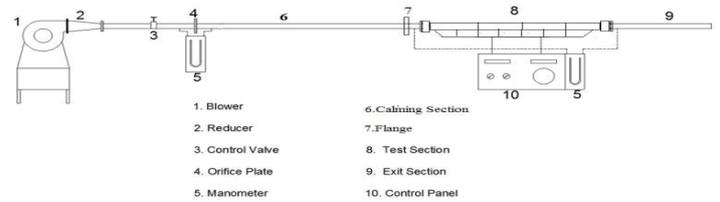


Fig 1 Experimental setup

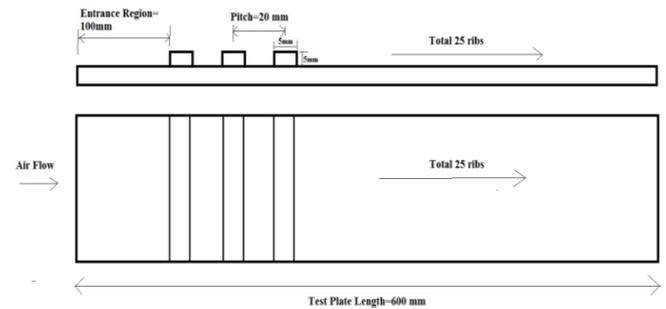


Fig. 2 (a) Square rib test section Configuration

B) Nomenclature:

- Nu : Nusselt number
- Re : Reynolds Number
- h : Convective heat transfer coefficient W/m²K
- ρ_a : Density of air
- V_a : Velocity of air m²
- D_h : Hydraulic diameter m
- μ_a : Kinematic Viscosity
- Δp : Pressure drop Pa
- f : Friction factor
- Pr : Prandtl Number
- T_s : Surface temperature K
- C_d : Coefficient of discharge
- L : Length of test plate m
- p : pitch mm
- Q_{in} : Heat input W

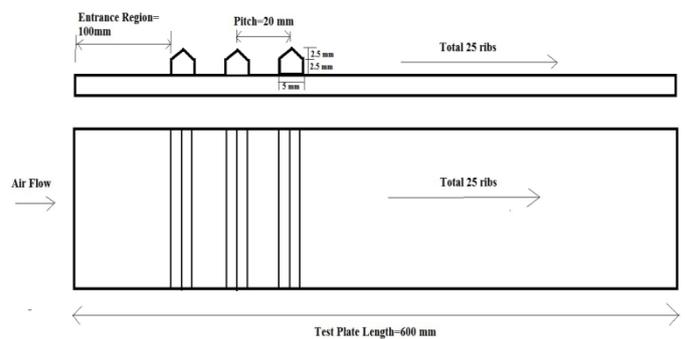


Fig. 2 (b) House Shaped ribs test section Configuration

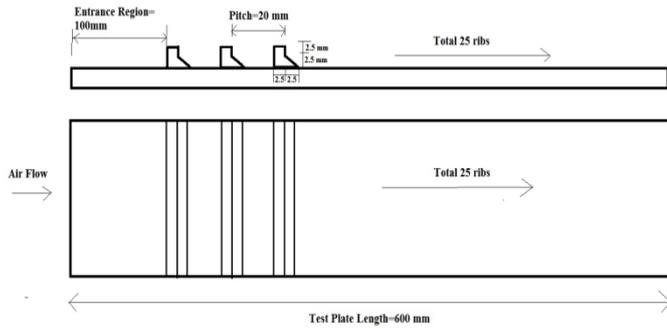


Fig. 2 (c) Boot shaped ribs test section Configuration

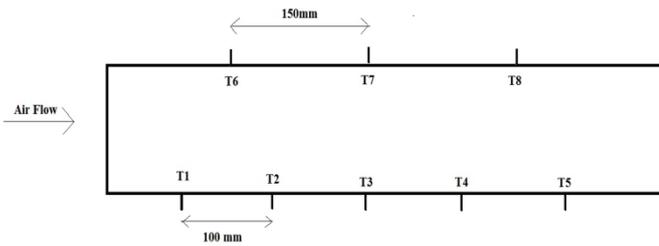


Fig. 2 (d) Thermocouple location

C) Experimental procedure

The thermocouples are connected to the reference junction and to the Test plate. Connection for electrical heater input is given to the sandwich heater from the control panel dimmerstat through a voltage servo stabilizer. After the heater input is given by setting a particular value on the dimmerstat a minimum of two hours is given for the heat sink to reach a steady state condition. Temperatures are then recorded and noted down. This procedure is followed for various heater input values for each step mentioned above.

III. DATA REDUCTION

The aim of the experiment is to investigate the Nusselt number in square, house shaped and boot shaped ribbed channels. The independent parameters are Reynolds number and ribs pitch ratios. Reynolds number based on the channel hydraulic diameter is given by

$$Re = \frac{\rho_a V_a D_h}{\mu_a} \tag{1}$$

The local heat transfer coefficients are evaluated from the measured temperatures and heat inputs. With heat added uniformly to fluid (Q_{air}) and the temperature difference of wall and fluid ($T_w - T_b$), average heat transfer coefficient will be evaluated from the experimental data via the following equations.

$$\text{Heat gain by the Air} = \dot{m} C_p \left(\frac{T_{10} + T_{11}}{2} - T_9 \right) \tag{2}$$

Rise in temperature of air = ΔT

$$\text{Where } \Delta T = \left(\frac{T_{10} + T_{11}}{2} - T_9 \right) \tag{3}$$

Properties of Air at $T = \frac{(T_{10} + T_{11} + T_9)}{2}$

$$\dot{m} = \text{Volume of Air} \times \rho \text{ (Density of air)} \tag{4}$$

$$\text{Volume of Air} = C_d \times A_{\text{Orifice}} \times \sqrt{2gH_{air}} \tag{5}$$

$$H_{air} = \left(\frac{\rho_{water} \times H_{water}}{\rho_{air}} \right) \tag{6}$$

$$Q_{rib} = h \times \text{Area of Orifice} \times \Delta T \tag{7}$$

$$\Delta T = (T_s - T_f) \tag{8}$$

$$T_s = \left(\frac{T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7 + T_8}{8} \right) \tag{9}$$

$$T_f = \left(\frac{\frac{T_{10} + T_{11}}{2} + T_9}{2} \right) \tag{10}$$

Calculate “h”

Then, average Nusselt number is written as:

$$Nu = \frac{h D_h}{k} \tag{11}$$

The friction factor is evaluated by:

$$f = \frac{D_h \Delta p}{2 \rho_{air} L U^2} \tag{12}$$

IV. RESULT & DISCUSSION

A) Verification of smooth channel

The present experimental results 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 correlations of Dittus–Boelter and Blasius for turbulent flow in ducts.

Correlation of Dittus–Boelter,

$$Nu_{th} = 0.023 Re^{0.8} Pr^{0.4} \tag{13}$$

Correlation of Blasius,

$$f = 0.079 Re^{-0.25} \tag{14}$$

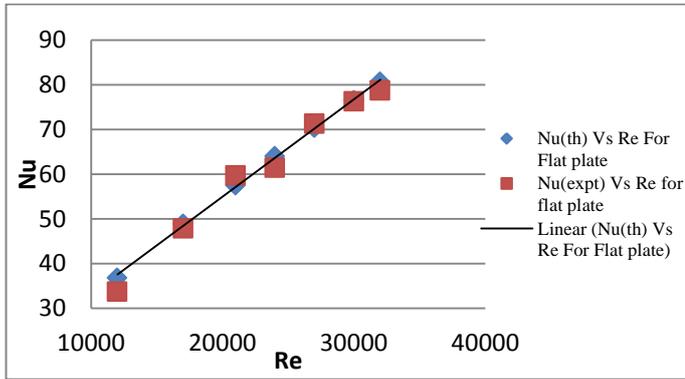


Fig 3(a).Nu Vs Re for validation of test setup

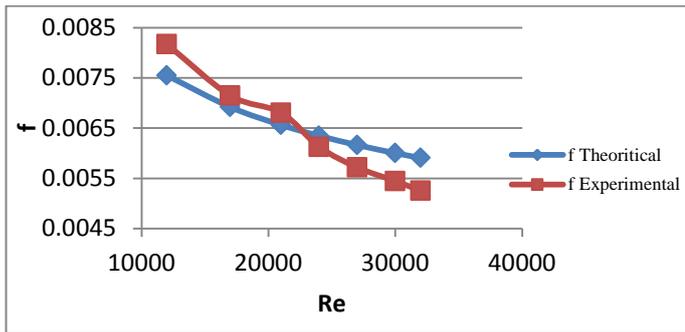


Fig 3(b).f Vs Re for validation of test setup

Fig. 3a and b show, respectively, a comparison of Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (13) and (14). In the figures, the present results reasonably agree well within $\pm 15\%$ deviation for both the friction factor and Nusselt number correlations.

B) Heat transfer enhancement and friction factor Results

Augmenting any test plate would result in increase in surface area providing more contact of air and an added benefit of increase in heat transfer rate due to continuous separation of boundary layer due to ribs with the cost of frictional losses. Thus the geometry with optimum heat transfer rate and less frictional losses has to be obtained. Experimental results show various heat transfer enhancements results as well as corresponding friction loss values. Fig 4(a) shows the Nusselt number for various pitches of ribs with increase in Reynolds number. As the Reynolds number advances the corresponding values of Nusselt number increases. It can be seen that maximum value of Nu is obtained for square ribs placed at 40 mm pitch which is around 155 while the ribs placed at 20mm pitch and 50 mm pitch shows identical results. Also Fig 4(b) shows the comparison of Nusselt number ratio to Reynolds number stating Square ribs at 40mm pitch shows better heat transfer performance.

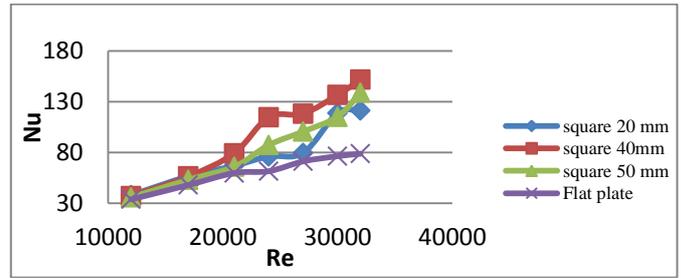


Fig 4(a)Nu Vs Re for Square Ribs

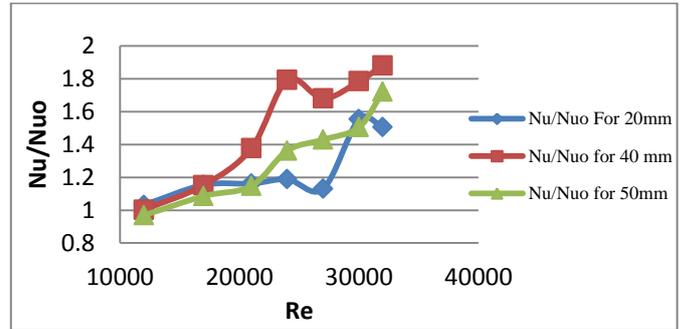


Fig 4(b)Nu/Nu0Vs Re for Square Ribs

Fig 5(a) shows the similar results for House shaped ribs but it is found to have comparatively less heat transfer enhancement than that of Square shaped ribs with a maximum of only 115 but similar to square ribs the house shaped ribs at a pitch of 40mm shows better results. Fig 5(b) represents results for Nusselt number ratio with Reynolds number.

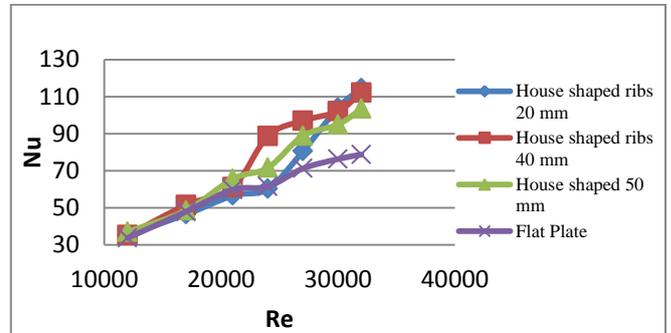


Fig 5 (a)Nu Vs Re for House shaped Ribs

Fig 5(b) represents results for Nusselt number ratio with Reynolds number.

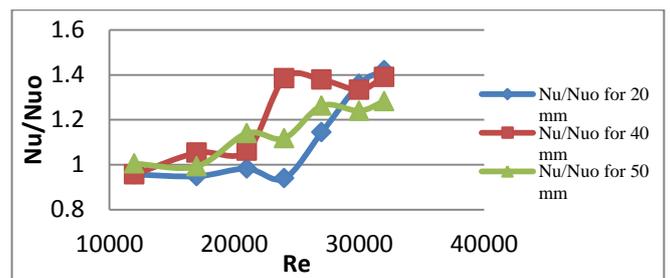


Fig 5 (b)Nu/Nu0Vs Re for House shaped Ribs

Fig 6(a) shows the results for House shaped ribs and it is found to have comparatively higher heat transfer

enhancement then that of Square shaped ribs and House shaped ribs with a maximum of 220 but similar to square and house shaped ribs ,the boot shaped ribs at a pitch of 40mm shows better results. Fig 6(b) represents results for Nusselt number ratio with Reynolds number for Boot shaped Ribs.

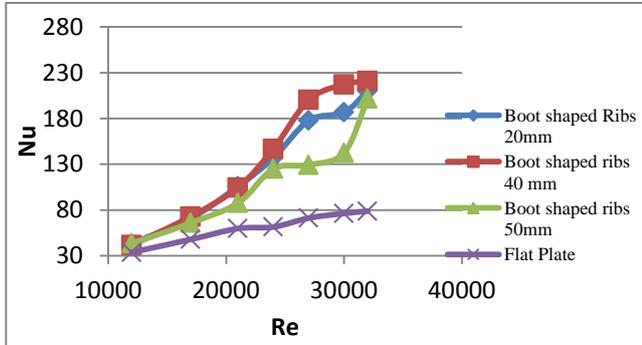


Fig 6(a)Nu Vs Re for Boot shaped Ribs

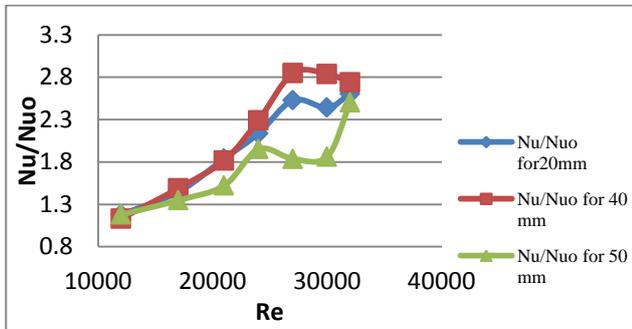


Fig 6(b)Nu/Nu0Vs Re for Boot Ribs

Fig 7(a) shows friction factor results compared to Reynolds number determines that the friction losses decreases with increase in Reynolds number, the square shaped ribs shows the highest friction losses followed by boot shaped and then house shaped ribs. Fig 7(b) represents the ratio of friction factor stating maximum friction factor reaches 4 for square ribs while the lowest is 1.8 for house shaped ribs.

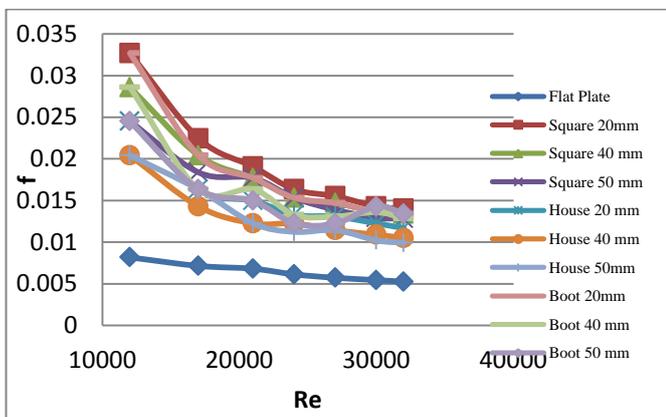


Fig 7(a) f Vs Re for each arrangement of Ribs

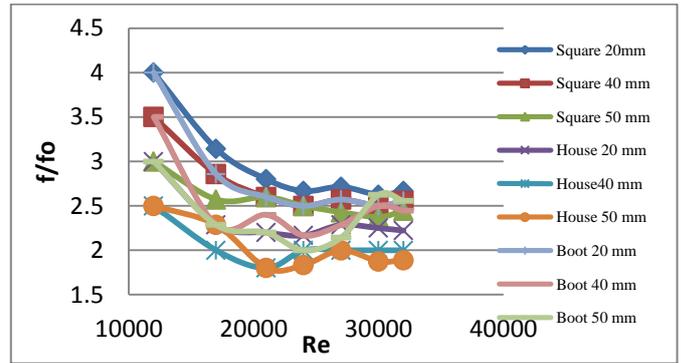


Fig 7(b) f Vs Re for each arrangement of Ribs

C)Thermal Performance of Various Ribs shapes

Thermal performance of any geometry is given by the ratio of thermal enhancement factor to Reynolds number given as

$$\text{(Thermal enhancement factor)} \eta = \left(\frac{Nu}{Nu_0} \right) \left(\frac{f_0}{f} \right)^{\frac{1}{3}} \quad (15)$$

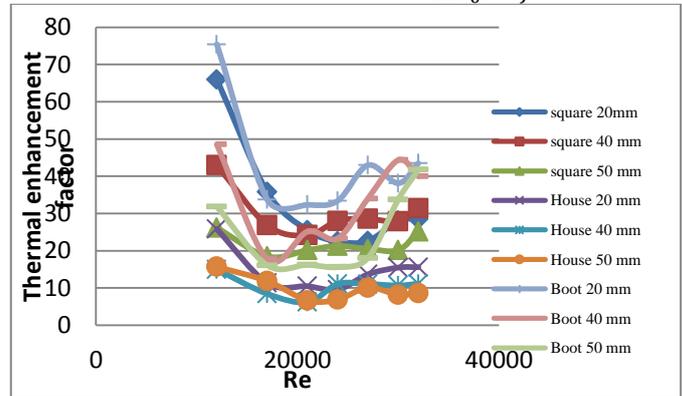


Fig 8 Thermal enhancement factor Vs Re for each arrangement of ribs

It covers both the significance of heat transfer enhancement and frictional losses combined to show the best optimum geometries. Results shows that the boot shaped ribs has the better thermal performance than that of house shaped or square shaped ribs. While the house shaped ribs shows the least thermal performance.

V.CONCLUSION

Heat transfer enhancement and Friction losses performances for three ribs shapes at various pitch of 20mm , 40mm, and 50 mm respectively was experimentally determined along with the thermal performance evaluation of each ribbed geometry. It was found that with increase in Reynolds number the heat transfer performance increases whereas the friction factor decreases for every geometry. The arrangement which showed the highest heat transfer enhancement was boot shaped ribs with 40mm pitch and worst was the house shaped ribs with 50 mm pitch. For all the geometries the heat transfer enhancement was found for ribs at 40mm pitch. In terms of friction factor the square ribs shows the highest frictional losses followed by the boot shaped ribs and then the house shaped ribs. Thermal performance result concludes that the boot shaped ribs has

the highest thermal performance than that of other two ribs shapes. This is because the rearpart of the boot-shaped rib induced a flow reattachment on the roughened surface earlier than the other shapes. So these new shape could be an advantage since it has the highest heat transfer rate with acceptable frictional losses and could be used for future applications.

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