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Forced Convection Heat Transfer Augmentation from Ribbed Surface

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ABSTRACT

Heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib tabulators, protrusions, pin fins, and dimples. These heat transfer enhancement techniques have practical application of internal cooling of turbine airfoils, combustion chamber liners and electronics cooling devices, biomedical devices and heat exchangers. The heat transfer can be increased by the following different Augmentation Techniques. They are broadly classified into three different categories as passive techniques, active techniques and compound Techniques.

Passive Techniques generally uses surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behavior (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power. These techniques do not require any direct input of external power; rather they use it from the system itself which ultimately leads to an increase in fluid pressure drop. They generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behavior except for extended surfaces. The objective of the experiment is to find out the convective heat transfer coefficient of ribbed plates test. The varying parameters were i) Ribs arrangement on the plate i.e. staggered and inline arrangement and ii) Heat input iii)pitch between two consecutive ribs on the plate. Convective heat transfer coefficients and Nusselt number were measured in a channel.. The Reynolds number based on the channel hydraulic diameter was varied from 4000 to 9000.Study shown that thermal performance is increasing with Reynolds number. With the inline and staggered ribs arrangement, the heat transfer coefficients, Nusselt number and the thermal performance factors were higher for the staggered arrangement.

Keywords— Convective heat transfer, Ribbed plate, Turbulent flow, Inline arrangement, Staggered arrangement, Forced convection.

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I. INTRODUCTION

The importance of heat transfer enhancement has gained greater significance in such areas as microelectronic cooling, especially in central processing units, macro and micro scale heat exchangers, gas turbine internal airfoil cooling, fuel elements of nuclear power plants, and bio medical devices. A tremendous amount of effort has been devoted to developing new methods to increase heat transfer from fined surface to

the surrounding flowing fluid. Rib turbulators, an array of pin fins, and dimples have been employed for this purpose.

In case of the electronics industry, due to the demand for smaller and more powerful products, power densities of electronic components have increased. The maximum temperature of the component is one of the main factors that control the reliability of electronic products. Thermal management has always been one of the main issues in the electronics industry, and its importance will grow in coming decades.

The use of heat sinks is the most common application for thermal management in electronic packaging. Heat sink performance can be evaluated by several factors: material, surface area, flatness of contact surfaces, configuration, and fan requirement.

Aluminum is the most common material because of its high conductivity (205W/mK), low cost, low weight, and easiness with respect to manufacturability. Copper is also used for heat sinks because of very high conductivity (400W/mK), but its disadvantages include high weight, high price, and fewer choices as far as production methods. To combine the advantages of aluminum and copper, heat sinks can be made of aluminum and copper bonded together.

To improve performance, heat sinks should be designed to have a large surface area since heat transfer takes place at the surface. In addition, flatness of the contact surface is very important because a nominally flat contact area reduces the thermal interface resistance between the heat sink and heat source. A heat sink must be designed to allow the cooling fluid to reach all cooling fins and to allow good heat transfer from the heat source to the fins. Heat sink performance also depends on the type of fluid moving device used because airflow rates have a direct influence on its enhancement characteristics.

To obtain higher performance from a heat sink, more space, less weight, and lower cost are necessary. Thus, efforts to obtain more optimized designs for heat sinks are needed to achieve high thermal performance. One method to increase the convective heat transfer is to manage the growth of the thermal boundary layer. The thermal boundary layer can be made thinner or partially broken by flow disturbance. As it is reduced, by using interrupted and/or patterned extended surfaces, convective heat transfer can be increased. Pin fins, protruding ribs (turbulators), louvered fins, offset-strip fins, slit fins and vortex generators are typical methods. The pattern and placements are suitably chosen based on the required cooling.

Heat transfer augmentation using these methods always results in pressure drop penalties that adversely affect the aerodynamics and efficiencies. In the case of cooling of turbine blades, surface protrusions induce excessive pressure losses, which increase the compressor load. The separated flow field over ribs or pin fins can make significant non-uniform cooling, which leads to thermal stresses.

II. LITERATURE REVIEW

Many experimental investigations have been carried out to determine configurations that produce optimum results in terms of both heat transfer and friction factor. Some of the most important attempts are to examine the rib geometry which gives the best heat transfer performance are discussed below.

Han *et al.* [1] investigated the influence of the surface heat flux ratio on the heat transfer in a square ribbed channel with e/D_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 *et al.* [2] performed an experimental study of fully developed turbulent air flow in square duct with two opposite roughened walls was performed to determine the effects of the rib pitch to height & rib height to equivalent diameter ratios on friction factor & heat transfer coefficient with Reynolds number varied between 7000 to 90000.Based on the four sided smooth duct correlation & the four sided ribbed duct similarity law, a general prediction method for average friction factor in rectangular duct with two smooth & two opposite ribbed walls was developed. There was good agreement between prediction & measurements. The result was useful for gas turbine blade internal cooling design.

Tauscher *et al.* [3] carried out the experimental and numerical investigation of the forced convective heat transfer in flat channels with rectangular cross section. To enhance the heat transfer, rib roughened surfaces are applied to the wider walls of the duct. Various rib shapes, rib spacing and rib arrangement have been investigated. Temperature fields and velocity fields in the heat exchanger channel were obtained by holographic interferometer and laser Doppler anemometry, enabling measurement of the thermo hydraulic behaviour of the flow without disturbing the flow pattern in the channel. By measuring the mean fluid temperature at the entrance and the exit of the test section as well as the pressure drop, the mean heat transfer and the heat exchanger performance could be judged. Simultaneously with the experimental investigations numerical calculations with a commercial code have been performed.

Maab *et al.* [4] studied the effects of inlet temperature and rib height on the fluid flow and heat transfer performances of the ribbed channel inside the high temperature heat exchanger are presented. The inlet temperature varies from 850 K to 1250K and the ratio of rib height to channel height varies from0.083 to 0.333. The results indicate that increasing the rib height can enhance the flow disturbance and hence improve the heat transfer performance. The inlet temperature has little effect on the basic structure of fluid flow and the heat transfer is enhanced due to the increased velocity. Compared to increasing the rib height, more heat can be transferred by increasing the inlet temperature with less pressure drop. The high pressure drop is more serious as the inlet temperature increases. It is proposed to use the compact heat transfer structure at the low temperature region and replace it by loose heat transfer structure at the high temperature region. It also demonstrates that the Nusselt number and friction factor are unsuitable to compare the heat transfer and pressure drop performances among different temperature conditions because the physical properties of fluid change with temperature variation.

Onbasioglu, *et al.* [5]A liquid crystal based experimental investigation of the heat transfer enhancement

supplied by ribs on a vertical plate, has been presented. Since the ribs were adiabatic, they did not work as extended heat transfer surfaces but by redirecting the flow, were used to enhance the heat transfer. Four different rib heights (H is 10, 20, 30, 40 mm) and have different angles of inclination (h is 0^0, 10^0, 20^0, 30^0, 45^0) are considered. Local heat transfer coefficients and the mean Nusselt numbers for the considered experimental cases are compared to those of the plate without ribs. The enhanced flow is always found to have higher heat transfer values. On the other hand, both the rib height and the angle of inclination affect on the magnitude of the local and total heat transfer coefficients.

Tanda, *et al.* [6] studied repeated ribs are on heat exchanger surfaces to promote turbulence and enhance convective heat transfer. Study of heat transfer from a rectangular channel (width-to-height ratio equal to five) is having one surface heated at uniform heat flux and roughened by repeated ribs. The ribs, having rectangular or square sections, were deployed transverse to the main direction of flow. The effect of continuous and broken ribs was also considered. Local heat transfer coefficients were obtained at various Reynolds numbers, within the turbulent flow regime.

Dhanawade *et al.* [7] Rapid heat removal from heated surfaces and reducing material weight and cost become a major task for design of heat exchanger equipments like Cooling of I C engines. Development of super heat exchangers requires fabrication of efficient techniques to exchange great amount of heat between surface such as extended surface and ambient fluid. The present paper reports, an experimental study to investigate the heat transfer enhancement in rectangular fin arrays with circular perforation equipped on horizontal flat surface in horizontal rectangular duct. The data used in performance analyses were obtained experimentally by varying flow, different heat inputs and geometrical conditions.

Maa *et al.* [8] studied air side heat transfer and friction characteristics of 14 enhanced fin-and tube heat exchangers with hydrophilic coating under wet conditions are experimented. The effects of number of tube rows, fin pitch and inlet relative humidity on airside performance are analysed. The test results show that the influences of the fin pitch and the number of tube rows on the friction characteristic under wet conditions are similar to that under dry surface owing to the existence of the hydrophilic coating. The friction characteristic is independent of the inlet relative humidity.

III. EXPERIMENTAL RIG

Actual experimentation will be conducted to investigate the effect of ribs on the heated plate in the forced convection heat transfer from heated plate. The heat will be generated within the plate with plate type heating element located below heating plate. An arrangement will be made to measure and vary the heat input with the help of transformer, and additional measuring instruments like voltmeter and ammeter. A provision is made to measure the mass flow rate of flowing air with orifice meter and U-tube manometer. The surface temperatures will be measured with the help of RTD's (PT-100 sensors) mounted at different locations of plates. Ten RTD's are to be fixed on the plate (5 on test sides) in order to measure the base temperature and one RTD is to be used to measure air temperature. Each RTD (PT-100 sensor) is to be fixed to the surface of the test plate at equal space locations along the plate length.

Another inclined tube manometer is used to measure pressure drop across the test section. The apparatus will be allowed to run until the steady state for particular heat input. The recording of temperature will begin after steady state has been reached. Same procedure will be repeated for different heat input and readings will be noted at steady state. Heat input to heater will be varied from 20W to 100 W in the step of 20W.

The procedure is used to take the reading on setup includes

1) Make all electrical connections i.e. main supply to Dimmerstat from Dimmerstat to voltmeter and ammeter after this supply is connected to the plate type heater. For blower and digital temperature indicator also required electrical connections.
2) After connection all start blower adjust mass flow rate as 5×10^{-3} kg/s and adjust the wattage given to heater as 20 W and for same wattage adjust the mass flow rate as 6×10^{-3}, 7×10^{-3}, and 8×10^{-3} and for constant wattage of 20W and for different mass flow rate note down corresponding temperature readings till steady state is reached with respect to time.
3) Next step is to note down the properties of air at bulk mean temperature properties include density, specific heat, Dynamic viscosity etc.
4) This stage we have to calculate Reynolds number, Prandtl number from available data. And find out the convective heat transfer coefficient and this experimental convective heat transfer coefficient is compared with Theoretical convective heat transfer coefficient.
5) Calculate the drop of pressure across the test plate from readings obtained in inclined manometer.

The schematic of experimental set up is as shown in figure 1 and The test plates used for experimentation is as shown in figure 2 to 6. The data reduction has been carried out from the observed readings to come out with effect of variation in ribbed plate on the forced convection heat transfer characteristics.

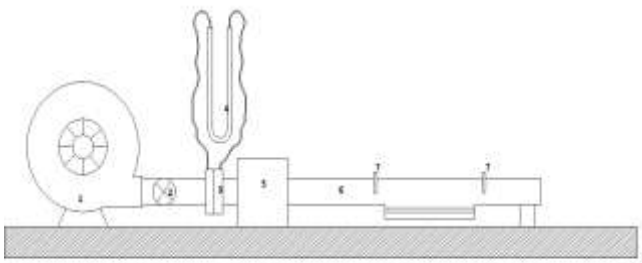

Fig. 1. Schematic of Experimental setup

1- Blower
2- Flow Control valve
3- Orifice meter
4- U- tube manometer
5- Planiuum Box
6- Rectangular duct of acrylic sheet
7- RTD for I/L and O/L temperature measurement

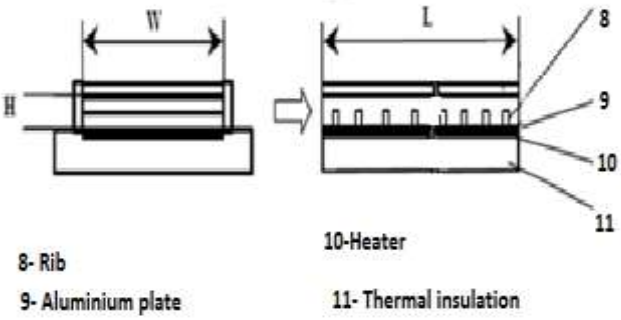

Fig. 2 Schematic diagram of Plate assembly

Fig.3 Image of Rectangular plate with Continuous rib large gap distance of 37 mm (CONFIGURATION – A)

Fig. 4 Image of Rectangular plate with broken rib with large gap distance of 37mm (CONFIGURATION – B)

Fig. 5. Image of Rectangular plate with continuous rib with small gap distance of 18mm (CONFIGURATION – C)

Fig. 6 Image of Rectangular plate with broken rib with small gap distance of 18mm (CONFIGURATION – D)

IV. DATA REDUCTION

To obtain Experimental convective heat transfer coefficient following method is employed.

$Q = h_{exp} \times A \times \Delta T$

$h = Q/(A \times \Delta T)$

Where,

Q= Total heat supplied in Watt considering losses of conduction and radiation

A= Surface area of plate in m^2

And $\Delta T = T_w - T_b$

Where T_w= Surface wall temperature

T_b= bulk mean temperature

$$Tb = \frac{Tin + Tout}{2}$$

(T_{in}, T_{out}) are inlet and outlet temperatures and T_w is the tube wall temperature which is the mean value measured by the four surface RTDs.

$$Tw = \frac{T1 + T2 + T3 + T4}{4}$$

To obtain Experimental convective heat transfer coefficient following method is employed.

We know that correlation for Nusselt number for ribbed surface

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

Reynolds number is calculated as follows

$$Re = \frac{m \times Dch}{Ach \times \mu}$$

Where, m = Mass flow rate of air in kg/s

D_{ch} = Hydraulic mean diameter in m

A_{ch} = C/S Area of rectangular channel in m^2

μ = Dynamic viscosity in Kg/ms

Prandtl Number is calculated as,

$$Pr = \frac{\mu \times Cp}{K}$$

Where,

μ = Dynamic viscosity in Kg/ms

Cp = Specific heat of air in J/Kg K

K = Thermal conductivity W/mK

Nusselt number is given by

$$Nu = \frac{hth \times Dch}{K}$$

$$h_{th} = \frac{Nu \times K}{Dch}$$

After completing all calculations we compared $\mathbf{h_{th}}$ and $\mathbf{h_{exp}}$. As both values are near to each other hence developed setup is validated and can further proceed for readings on dimpled plates.

V. RESULTS AND DISSCUSSION

A number of experimental runs are carried out on the setup with plain Aluminium plate and by changing the wattage input and varying mass flow rate of the air by the mentioned testing methodology the obtained results are presented in the following section.

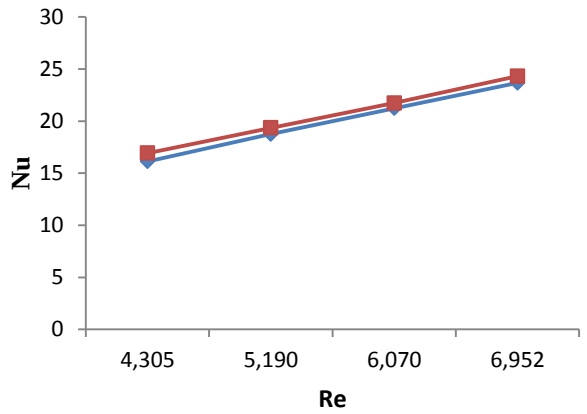

Fig No. 6:- Graph of Nu V/S Re for plain Plate no. 01 @ P= 20 W

From above graphs of Nu V/S Re for plain aluminium we clearly identify that there is slightly variation in the values of theoretical Nusselt number and experimental Nusselt number for Reynolds number. Range is of Reynolds number various from 4000 – 7500 and for these values curves of Nusselt number theoretical and experimental variation is there but from this we can conclude that setup is validated and further readings on same setup for ribbed plates taken and convective heat transfer coefficient is calculated and compared with the plain plate.

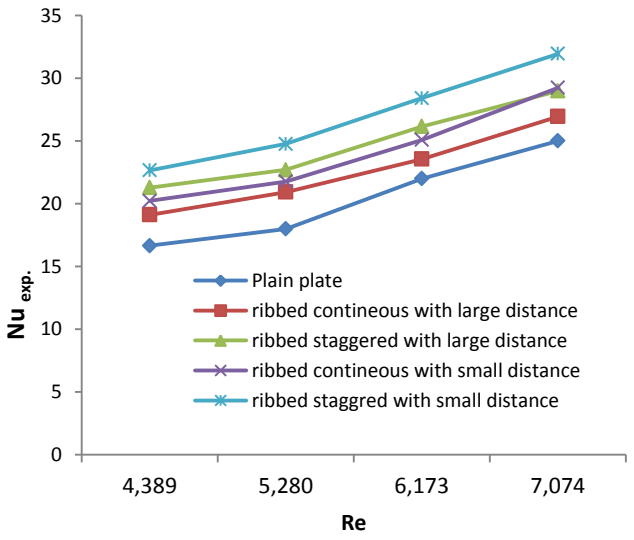

Fig No. 7:- Graph of $Nu_{exp.}$ V/S Re for all Plates @ P= 20 W

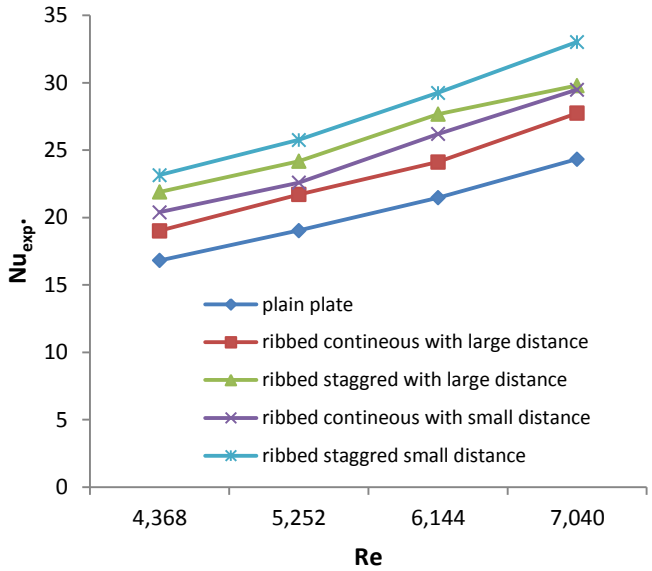

Fig No. 8:- Graph of $Nu_{exp.}$ V/S Re for all Plates @ P= 40 W

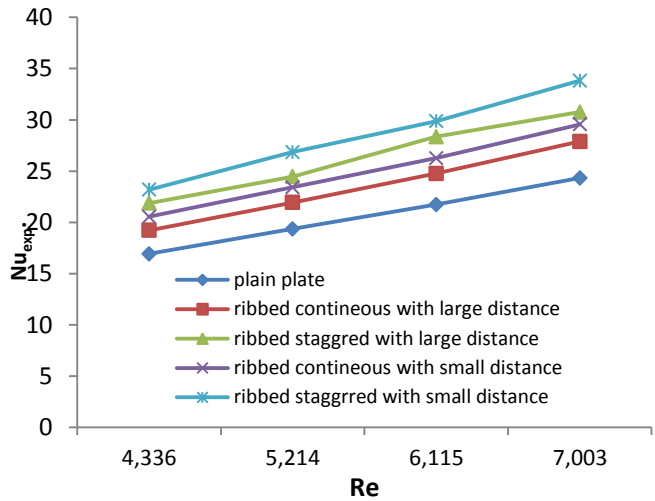

Fig No. 9:- Graph of $Nu_{exp.}$ V/S Re for all Plates @ P= 60 W

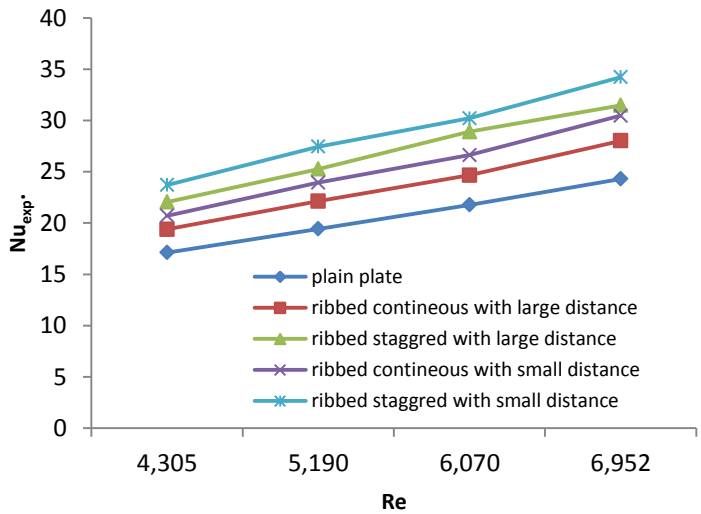

Fig No. 10:- Graph of $Nu_{exp.}$ V/S Re for all Plates @ P= 80 W

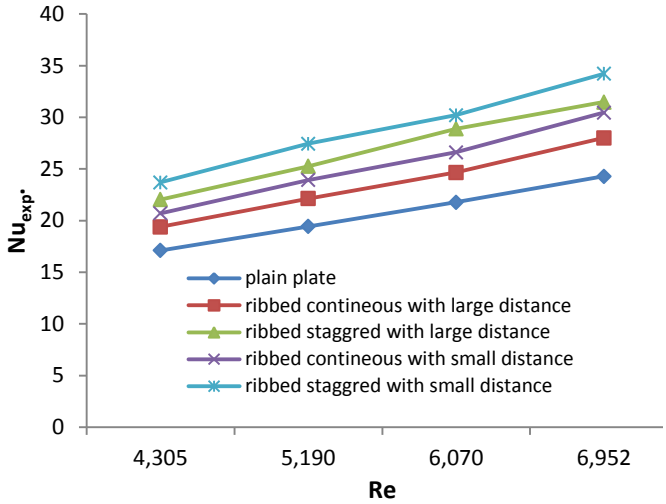

Fig No. 11:- Graph of $Nu_{exp.}$ V/S Re for all Plates @ P= 100 W

From all above graphs of $\mathbf{Nu_{exp}}$ V/S Re for all plates we clearly identify that there is more enhancement in the convective heat transfer coefficient for plate having staggered ribs with smaller gap distance apart (20 mm) than compared with all. After that plate having ribbed staggered arrangement of large gap distance between two ribs (37mm) is second in heat transfer enhancement. And after that continuous ribbed plate with larger gap distance and after this lastly continuous ribbed with small gap distance. From above graph we conclude that maximum heat transfer enhancement takes place with staggered arrangement ribs with smaller gap distance. As Reynolds number increases the Nusselt number also increases as shown in above graph.

VI. CONCLUSION

The experimental investigation of forced convective heat transfer from ribbed surface is calculated for different rib configurations with either continuous or broken ribs and for single plain plate with varying heat supplied and different mass flow rates of air after completing experiment for all readings of plain plates and ribbed plates of different rib configurations theoretical and experimental convective heat transfer coefficient is calculated the results obtained were found to agree with literature data and the following conclusions were made.

1. Among four different configurations of the ribbed plates the maximum convective heat transfer coefficient when compared with plain plate is for configuration with staggered rib with smaller gap distance i.e. configuration “D”.
2. The results were plotted and from graph it shows that as Reynolds number increases with increase in mass flow rate of air for all wattage ranges the Nusselt number also increases and shows the enhancement of convective heat transfer coefficient.
3. The configuration “A” having continuous rib with large gap distance shows 13% enhancement in convective heat transfer coefficient when compared with plain plate.
4. The configuration “B” having staggered rib with large gap distance shows 25 % enhancement in convective heat transfer coefficient when compared with plain plate, And 12% more than configuration “A”.
5. The configuration “C” having continuous rib with smaller gap distance shows 20 % enhancement in convective heat transfer coefficient when compared with plain plate, And 7% more than configuration “A”, And 5% less than configuration “B”.

Finally we conclude that the rib configuration “D” shows the maximum enhancement in convective heat transfer rate.

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