Experimental Examination Of Heat Transfer Characteristics Of Non Conventional- Truncated Stepped Grooved Ribs

Sameer Y.Bhosale*, Prof.Dr.G.R.Selokar

Abstract: In improvement of the thermal performance there is necessity of the heat transfer augmentation. As the series of ribs publicists the turbulence to intensify the heat transfers to the working medium in the duct. For this criteria we made the numerical and experimental examination of non conational type truncated stepped grooved shoe shape rib under forced convection. The study concentrates on truncated ribs to improve thermal performances with continuous ribs. For the experimentation, velocities (1 m/s – 2.5 m/s) and heater inputs (625 W/m^2 - 960 W/m^2) are considered to determine the heat transfer performance. The highest magnitudes of Nusselt Number (Nu), Thermal Enhancement factor (TEF) are 217.01.1.42 respectively. The results of experiments are also validated with Computational Fluid Dynamic (CFD) results and literature results. The results obtained by the experimentation are in good agreement with CFD results and higher than that of results from same domain literature.

Index Terms: Heat Transfer Characteristics, Non Conventional, Numerical, Ribs, Thermal Enhancement factor, Truncated

1. INTRODUCTION

Ribs are a popular heat transfer intensification device used in different 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 increases turbulent heat transfer in the channel. Among the many geometric parameters related to the rib arrangement and configuration, the shape of the rib cross section affects the formation of a separation bubble behind the rib and the amount of turbulent kinetic energy production; thus, the rib shape is a major factor that determines the heat transfer performance of the rib. The heat transfer from surface may in general be enhanced by increasing the heat transfer coefficient between a surface and its surrounding or by increasing heat transfer area of the surface or by both. Ribs used in cooling channel and heat exchanger channel are most commonly used as passive heat transfer techniques. The effects of the rib arrangement and configuration on the heat transfer performance have been inspected by many researchers. Many researchers performed an experimental study on the effects of the pitch-to-hydraulic diameter and rib height-to-hydraulic diameter ratios on the heat transfer performance. Many of them studied the effect of the rib type (continuous and truncated) on the heat transfer performance experimentally an experimental study on the heat transfer performance and friction loss characteristics of a rectangular cooling channel with transverse ribs on one, two, three, and four walls was also conducted. Some researchers investigated the effect of the rib shape on the heat transfer and pressure drop characteristics of a rectangular channel numerically. They tested types of rib shapes: rectangular, triangular, cylindrical, concave–concave, convex–concave, long concave–short concave, and long convex–short concave.

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Mi-Ae Moon, et.al, [1] have determined the friction loss and heat transfer performances from rib mounted in rectangular duct. The simulation of sixteen different type of ribs geometries were performed for the determination of geometry with highest thermal performance. The pitch to height ratio was fixed to 10, width of the rib to hydraulic diameter 0.047. it was observed that the shoe shape was augmented the heat transfer at the same pressure drop comparing with the square cross section rib. Deep Singh Thakur, et.al, [2] studied hyperbolic rib arrays with the height 0.5 mm to 2 mm, with pitch 10 mm to 20 mm. The best results were achieved for e = 1 mm and P = 10 mm at Re = 6000. This rib was evaluated with the comparison of triangular, rectangular and semicircular rib shapes and it was observed that this rib gives the better performance at Re 10,000. Farzad Pourfattah, et.al, [3] the aim of this research work was to analyze the angle of attack of rib with aluminum nano particles for the heat transfer augmentation effect. In this it was accomplished that, with the presence of ribs eddy formation was increased towards the flow direction results into optimum mixing of flow which leads to the enhancement of heat transfer. Sang-Hyo Kim, et.al, [4] analyzed Y-type perforated rib with the shear forces in simulation software to search out the effects of, edge distance, hole diameter transverse rebar diameter on the heat transfer. L. Varshney, et.al, [5] studied twelve types of ribs with different tapered angle employed for the roughness maintenance of the plate. The value of Nussult number and friction factor at constant heat flux was observed. The optimum results found at the Reynolds number 3800 to18000 where the highest performance factor was found as 1.91 with the reference plate. Jinsheng Wang, et.al, [6] in this research rib geometry were analyzed geometry based on width, rib stream wise gap distance, pitch, inner half rib width angle. Best heat transfer performance was at the stream size maintained at 0.2 and inner half angle 45^0 and also it was observed that the width gap also affect to pressure drop. NianbenZheng, et.al, [7] three-dimensional Reynolds-averaged Navier– Stokes equations was used to determine the heat transfer and friction loss performances of rib-roughened rectangular cooling channels having a variety of cross-sectional rib shapes. The Reynolds stress model was used with the pressure strain model to analyze the turbulence. The computational results for the area-averaged Nusselt number were determined and also validated by comparison with the experimental data under the same conditions. The new shoe-shaped rib design showed the best heat transfer performance with a pressure drop similar to that of the square rib. Alessandro Salvagni, et.al, [8] investigated thermal performance of the bottom duct wall equally-distanced square-sectioned ribs heated except for the ribs with an imposed constant heat flux. The outcome for the study was well-resolved LES gave some new insight into the rotation effects on flow and heat transfer, providing information that were not easily accessible to experiments. T. Alam,et.al, [9]studied the application of conical protrusion ribs roughness on the absorber plate of solar air heater duct which showed effective enhancement of the heat transfer rate irrespective of pressure drop penalty. It was claimed that the values of friction factor decrease continuously with increasing the relative pitch ratio, which was due to fact that higher values of relative pitch attributed to low resistance offer to flow. S. Alfarawi et.al, [10] determined heat transfer and flow friction for a fully developed turbulent flow in a rectangular duct with its bottom wall ribbed with three different rib geometries such as semi-circular, rectangular and hybrid ribs of the two. The key result of the analysis was the enhancement in the heat transfer is critically influenced by the flow velocity and the turbulence intensity as well as the rib pitch to height ratio. In this research work the truncated stepped grooved shoe shaped ribs (Fig. 3) are examined experimentally with set of variation of velocities (1 m/sec – 2.5 m/sec) and varying the heater inputs (625W – 960W) to determine the heat transfer performance. The results of experiments are also validated with CFD results and previous literature results.

2 EXPERIMENTATION

Experimentation is carried out for the flat plate and finalized geometry. Experimentation is carried out four different air velocities for three different heater inputs. Air velocities are 1 m/s, 1.5 m/s, 2 m/s and 2.5 m/s and the heater inputs are 75 W, 95 W and 115 W. Heater plate connected to the plate carries out the heating purpose. Thermocouples are connected linearly to the plate, the number of thermocouples being 8. These thermocouples show the surface temperatures of the plate across the plate and 2 thermocouple wires are placed at the inlet and outlet of the plate that indicate inlet and outlet temperature of the air inside the duct. The plate is kept inside the duct that had 2.88 m length, 255 mm height and 155 mm width. Air is forced on the plate by a blower provided at one side of the plate. The voltage given to the heater plate is regulated by the voltmeter and air velocities are tested by the use of anemometer.

2.1 Experimental set- up

An experimental setup for analyzing the heat transfer enhancement through the stepped grooved shoe shaped ribs is shown in the Fig. 1. This setup is mainly divided in to three components which are further assembled by using the screw and bolts with asbestos is used to fill the gaps of joints between the intersection of the three components. Also the wall inside the test rig is padded against the wooden material to have uniform cross-sectional length. The components of the test rig are the blower from which the required quantity of the air will be supplied, diverging section and the main duct in between the honeycomb structured section is used to have more uniform velocity profile and also to reduce the eddies from the generated flow. Blower has a hand operated butterfly valve which can be

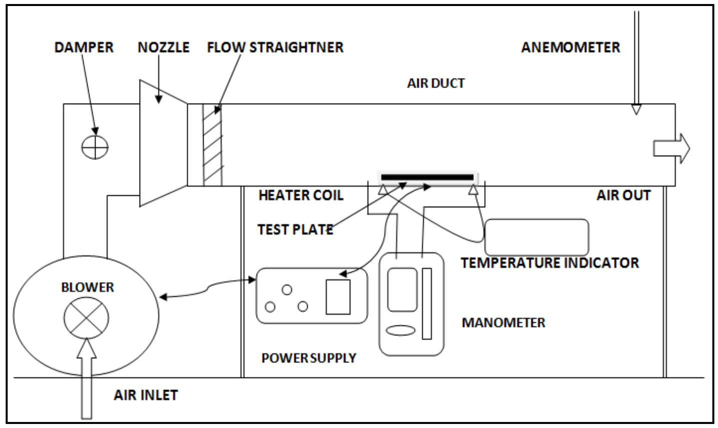

Fig.1 *Experimental Set-up*

opened and closed to get different velocities. A blower is powered with a motor for the purpose of adequate supply of air stream. As we go on closing the valve, the velocity of the air flow goes on reducing and as we go on opening the valve, the velocity of the air flow increases. To make the air flow velocity accurate and make the controlling or setting the velocity of the air flow use of digital Anemometer gives the reliability. The coil electric heater is used to provide the constant heat flux to the experimental plate on which the ribs are mounted. The heater is placed inside the main duct with proper fitment by using the screws and nuts. Heating coil used to supply constant heat flux to the object plate is controlled by the constant voltage transformer. Constant voltage transformer is used to impede the current fluctuation and to supply uniform current to the object plate to avoid the temperature surges. Three dimensional turbulent fluid flow and convection from the ribs is studied experimentally. Plate we are using is made of aluminum 0.8m×0.150m×0.004m and the heat input is given to the plate with the help of dimmer stat and control panel. Aluminum flat plate with and without ribs Fig.2 used for the experimentation and is attached with the heater plate for the heating purpose. The heater plate is clamped to the aluminum plate to get the maximum thermal contact and to avoid the losses due to thermal contact resistance. The dimension of the plate is as follows: length of the plate is 0.8 mm, thickness of the plate is 4 mm, width of the plate is 0.150 m. Eight holes of M5 size are drilled in a line along the center line of the plate where the thermocouple wires are attached. These thermocouple wires are attached to the plate with washer and bolts. To measure the inlet temperature, a thermocouple wire is suspended from the duct from a hole that is drilled at the upper surface of the duct. To measure the outlet temperature, anemometer is used. By knowing these temperatures, mean surface temperature of the duct can be calculated. For three heat input readings were recorded, 626 W/m^2, 802 W/m^2 and 902 W/m^2. For each hear flux, velocity is varied by keeping the heat flux constant at the interval of 0.5 m/s, i.e. 1 m/s, 1.5 m/s, 2m/s and 2.5 m/s, as expressed in the matrix below (table 1).

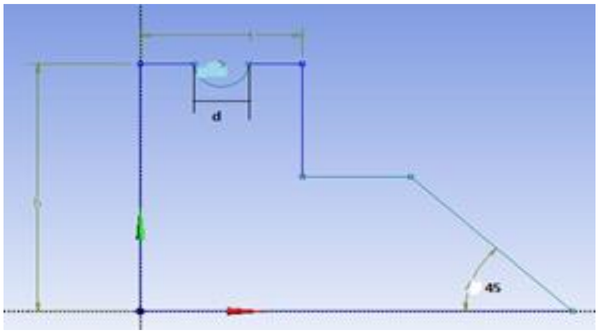

Fig.2 *Rib Arrangement on plate*

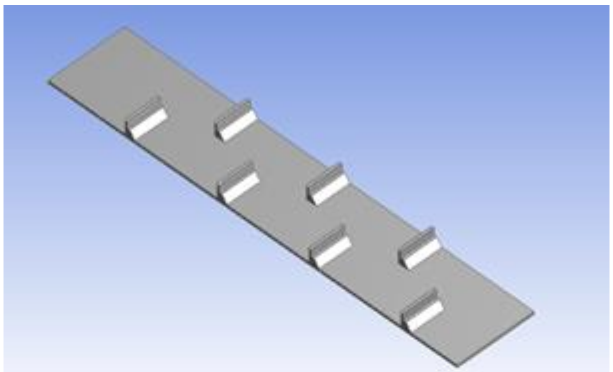

***Fig.3**. Rib Geometry*

Table 1 : *Test matrix for experimental analysis*

Sr. No.	Geometry	Heater Input Q (W/m^2)	Velocity V (m/s)
1	1] Plain Plate 2] Truncated stepped grooved shoe shape with grooved rib gap=20mm t=5mm d=3mm	626	1
2			1.5
3			2
4			2.5
5		802	1
6			1.5
7			2
8			2.5
9		902	1
10			1.5
11			2
12			2.5

3 NUMERIACAL ANALYSIS

The cross section of experimental air domain is maintained with rectangular shape of dimensions of 180mm×290mm and the length of it is 2.8 m. In this stepped grooved shoe shape rib mounted plate is placed for the analysis of heat transfer enhancement. The study and analysis is done with simulation software with considering the K-ξ model. The temperature velocity and pressure profiles are plotted from CFD software. The thermal performance parameters are determined based on these results.

3.1 Velocity Profile

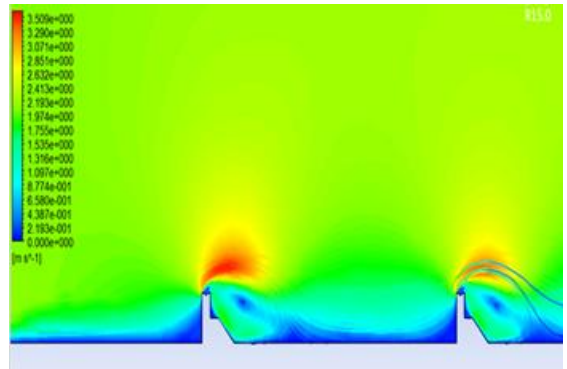

Fig. 4 *Velocity profile of Truncated stepped grooved shoe shape rib*

Fig. 4 indicates vortex generation after the ribs and subsequent reattachment of it further downstream between the two ribs which is responsible for increase in heat transfer. In case of while in case of stepped grooved shoe ribs there is secondary vortex generation along with primary vortex which leads to localized hot spots in groove. This gives the more variation in groove shape on the rib head gives the significant increase in heat transfer coefficient.

3.2 Pressure Profile

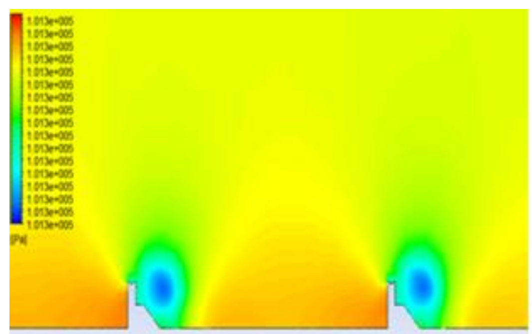

Fig. 5 *Pressure profile of Truncated stepped grooved shoe shape rib*

Shoe shape gives the best energy transfer as the pressure developed by the ribs converted in to the velocity which gives the increase in the heat transfer. This pressure drop gives the friction factor enhancement on the increase in the same. But the increment in friction factor is not very high as compare to thermal enhancement therefore thermal enhancement factor increases.

3.3 Temperature Profile

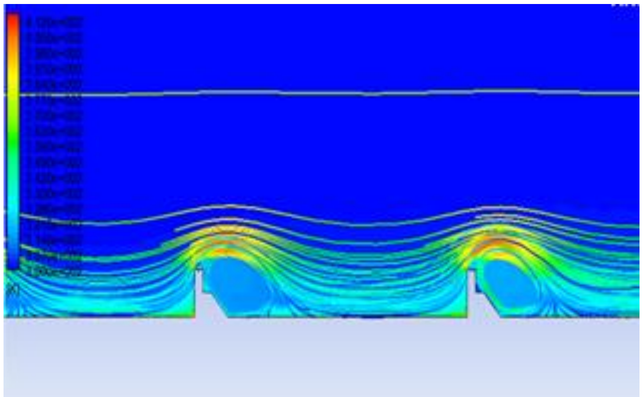

Fig. 6 *Temperature profile of Truncated stepped grooved shoe shape rib*

Fig. 6 shows temperature contours for stepped grooved shoe shaped rib. The average surface temperature obtained is 336.915 K respectively. Stepped shoe shape ribs with groove gives the smallest average temperature of the surface with which results in higher heat transfer coefficient across the plate.

4 DATA REDUCTION

1] Average surface temperature (Ts)
It can be taken from the obtained temperature plots

2] Nusselt number
The Nusselt number is a measure of the convective heat transfer occurring at the surface and is defined as hd/k, where h is the convective heat transfer coefficient, d is the diameter of the tube and k is the thermal conductivity.

$$Nu = \frac{qD_h}{k(T_S - T_O)} \quad (1)$$

3] Friction factor
The friction factor is a measure of head loss or pumping power.

$$f = \left(\frac{\Delta P}{\frac{1}{2}\rho U_m^2}\right)\frac{D_h}{L} \quad (2)$$

4] Thermal enhancement factor
The thermal enhancement factor 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.

(3)

5 I

$$TEF = \frac{\overline{Nu}/\overline{Nu}_o}{(\bar{f}/\bar{f}_o)^{\frac{1}{3}}}$$

JSSION

5.1 Effect of variation of Reynolds number on Nusselt number (Experimental results)

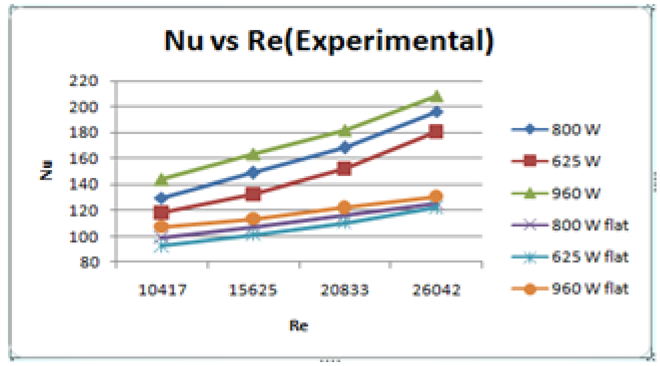

Fig. 7*. Effect of Re on Nu for different heater inputs for truncated stepped grooved rib and Flat plate*

Trend of Nusselt number with respect to Reynolds number is examined experimentally for Reynolds number 10417, 15625, 20833, 26042. Nusselt number shows the increasing trend with the Reynolds number this is due to turbulence introduced due to rib intervention. The maximum values gives of the Reynolds number 26042 are 182.07, 199.76, and 217.01 for heater input of 75W, 95W and 115W respectively. Experimental analysis observed 27% to 59% enhancement of Nussult number of stepped shoe shape rib with respect to flat plate.

5.2 Effect of variation of Reynolds number on heat transfer enhancement (Experimental results)

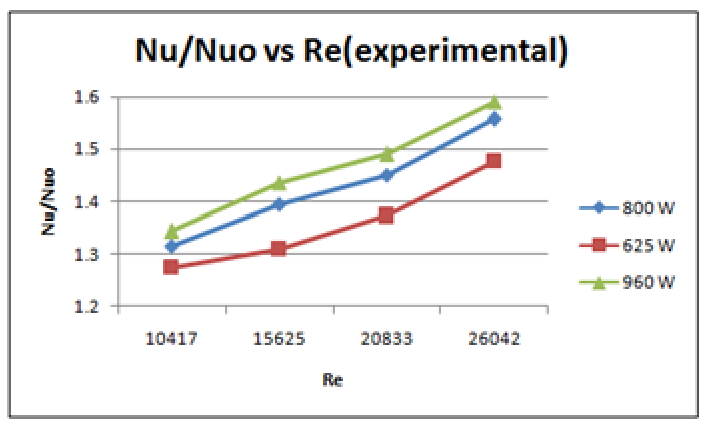

Fig. 8 *Effect of Re on heat transfer enhancement for different heater inputs for truncated stepped grooved rib*

Experimental results of Nu/Nuo versus Reynolds number are examined which shows the increasing trend of the Nu/Nuo for the Reynolds number from 10417 to 26042. The Reynolds number of 26042 it gives highest value of 1.47, 1.55 and 1.59 for the heater input of 75W, 95W and 115W respectively.

5.3 Effect of variation of Reynolds number on frictional characteristics (Experimental results)

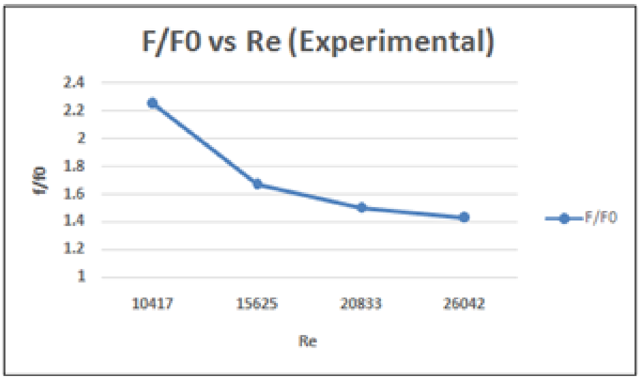

Fig. 9 *Effect of Re on friction factor ratio for truncated stepped grooved rib*

The ratio of friction factor deceases with increase in Reynold number. The trend of the graph remains same in all three cases of heater inputs. Temperature has no significant effect on the pressure drop or friction factor. Effect of f/fo on Re is presented in Fig 8. Minimum value is obtained at Re of 26041.7 which is 1.42.

5.4 Effect of variation of Reynolds number on Thermal enhancement factor (Experimental results)

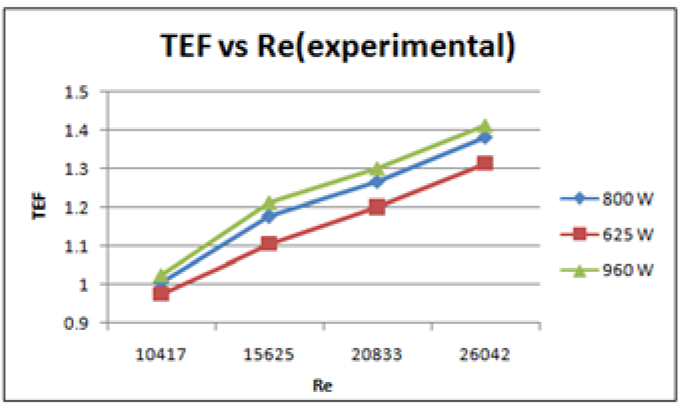

Fig. 10 *Effect of Re on Thermal Enhancement factor for truncated stepped grooved rib for all heater inputs*

Experimental analysis is carried out for the TEF verses Reynolds number. Reynolds number is varied is from 10416.7 to 26041.7 for different heater inputs of 75 W, 95 W and 115 W. Fig.10 shows the graph between TEF and Re. An increasing trend for TEF is observed with increase in Reynolds number as TEF is directly proportional to Re. Maximum values of TEF are obtained for highest Re of 26041.7which are 1.311, 1.3824 and 1.4124 for heater inputs of 75 W, 95 W and 115 W respectively.

5.5 Effect of variation of Reynolds number on Nusselt number (Numerical results)

Fig. 11 *Effect of Re on Nu for different heater inputs for truncated stepped grooved rib and Flat plate*

Variation for Nu versus Re is studied numerically (Fig 11), Re is varied is from 10417 to 26042 for different heater input of 75 W, 95 W and 115 W. Fig.10 shows the graph between Nu and Re, an increasing trend for Nu is observed with increase in Reynolds number. Maximum value is obtained for highest Re of 26042 which is 192.07, 213.46 and 225.01 for heater input of 75 W, 95 W and 115 W respectively. Numerical analysis observed 45% to 63% enhancement of Nussult number of stepped grooved shoe shape rib with respect to flat plate. Trends observed in both cases (numerical and experimental) are similar with experimental results deviating from numerical with small variation of 7.2% only.

5.6 Effect of variation of Reynolds number on heat transfer enhancement (Numerical results)

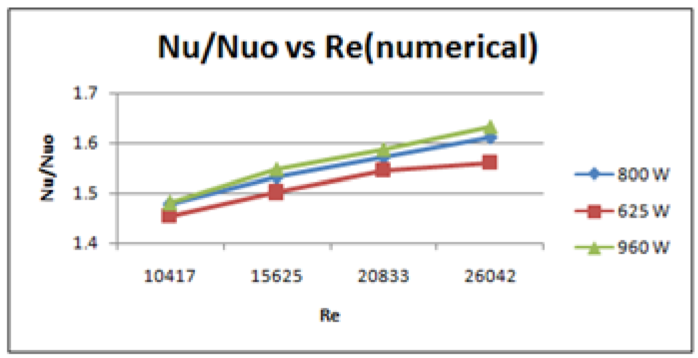

Fig. 12 *Effect of Re on heat transfer enhancement for different heater inputs for truncated stepped grooved rib*

Trend of Nussult number with respect to Reynolds number is examined numerically (Fig 12) for Reynolds number 10417, 15625, 20833, 26042. Nussult number shows the increasing trend with the Reynolds number this is due to turbulence introduced due to rib intervention. Maximum values of heat transfer enhancement ratio achieved at the Reynolds number 26042 are 1.56, 1.61 and 1.64 for heater input of 75W, 95W and 115W respectively.

5.7 Effect of variation of Reynolds number on Thermal enhancement factor (Numerical results)

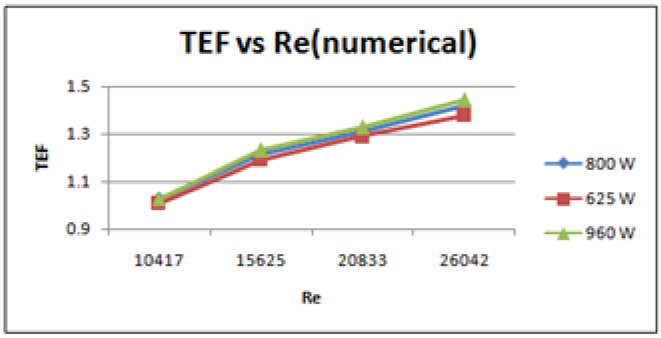

Fig. 13 *Effect of Re on Thermal Enhancement factor for truncated stepped grooved rib for all heater inputs*

Fig.13 shows the graph between TEF and Reynolds number. Reynolds number is varied is from 10416.7 to 26041.7 for different heater inputs of 75 W, 95 W and 115 W. An increasing trend of TEF is observed with increase in Reynolds number. Maximum values of TEF are obtained for highest values of Re i.e. 26041.7 are 1.3771, 1.4217 and 1.441 for heater input of 75 W, 95 W and 115 W respectively. Trends observed in both cases (numerical and experimental) are similar with experimental results deviating from numerical with small variation of 3.5% only.

5.8 Comparison of Experimental Results with results from literature (Validation)

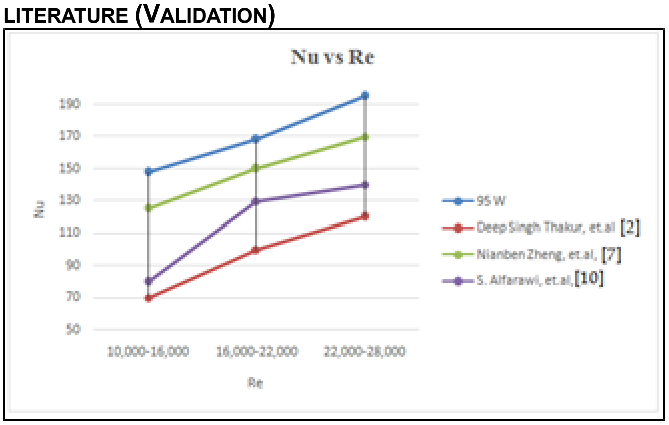

Fig. 14 *Validation of Experimental Results with results from previous literature*

The comparison of experimental results is done with results of literature for the purpose of validation. Reynolds Number is plotted against Nu for the above purpose. In above graph, the blue line shows maximum Nusselt Number obtained from the present work are plotted. These results are taken at heater input of 95 W and Reynolds Number varying from 10416 to 26041. Deep Singh Thakur, et.al, [2], had performed a CFD analysis on hyperbolic ribs to study flow characteristics and heat transfer characteristics. The rib is tested for different pitch by rib height ratio and by varying Re. The highest nusselt number obtained is 170. NianbenZheng, et.al, [3], had carried out a numerical investigation on the flow pattern and heat transfer in an internally ribbed heat exchanger tube. Two types of ribs, namely parallel type ribs (P-type ribs) and V shape type ribs (V-type ribs) were imployed for the above purpose. The points plotted showed the results obtained in case of P-type ribs. The highest Nusselt number obtained is 140.

In second case, the authors S. Alfarawi, et.al, [12], had carried out an experimental investigation for a fully developed turbulent flow in a rectangular duct with its bottom wall ribbed with three different rib geometries: semi-circular, rectangular and hybrid ribs of the two. The analysis showed that the hybrid ribs showed the optimum performance, thus the points are plotted for the hybrid ribs. The highest Nusselt number obtained is 120.

6 CONCLUSIONS

1. The new Stepped Grooved shoe shape rib can be considered as a heat transfer augmentation tool. The shoe shape thickness and groove diameter plays main role in heat transfer increment.
2. Vortex generation after the ribs and subsequent reattachment of it further downstream between the two ribs which is responsible for increase in heat transfer.
3. In case of truncated stepped with grooved shoe shape rib heat transfer increases due to development of turbulence which gives the heat transfer enhancement with respect to the flat plate.
4. In experimental analysis it is observed that Nussult number increased by 27% to 59% for truncated stepped shoe shape rib with respect to flat plate.
5. Nusselt number variation trends observed in both cases (numerical and experimental) are similar with experimental results deviating from numerical with small variation of 7.2% only.
6. In experimental analysis the highest magnitude of Thermal enhancement factor is 1.42.
7. Thermal enhancement factor variation Trends observed in both cases (numerical and experimental) are similar with experimental results deviating from numerical with small variation of 3.5% only.

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