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# A Comprehensive Review on Roughness Geometries and Investigation Techniques Used in Artificially Roughened Solar Air Heaters

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**Abstract-** Artificial roughness applied on the absorber plate is the most acclaimed method to improve thermal performance of solar air heaters at the cost of low to moderate friction penalty. Experimental investigations pertinent to distinct roughness geometries unfolds that the enhancement in heat transfer is accompanied by considerable rise in pumping power. In view of the fact, a designer needs to carefully examine shape and orientation of roughness elements in order to choose the best fit roughness geometry for intended application. Moreover it is required to understand how flow field is affected by particular roughness geometry so that direction of future researches could be conceived. So as to elucidate the useful findings an attempt has been made to review roughness geometries employed in solar air heaters. Some distinguished roughness geometries have been compared on the basis of heat transfer enhancements and thermohaydraulic performance to draw attention towards their usefulness for specific applications. Furthermore, light is thrown on different investigation techniques adopted for prediction of heat transfer and friction characteristics of artificially roughened solar air heaters to recognize features and limitations of each technique.

**Keywords-** Artificial roughness, Solar air heater, Roughness geometry, Nusselt number, thermohydraulic performance, Reynolds number.

## 1. Introduction

The role of energy becomes increasingly important to fulfill needs of modern societies and to sustain fast economic and industrial growth worldwide. Conventional energy sources are depleting day by day and seem to be insufficient to fulfill large demand of energy in coming years. As one of long term alternative, the renewable energy sources are having enough potential to occupy the place of conventional energy sources. Solar energy is one of the promising and easily convertible forms of renewable energy available in abundance on earth. Though it is location and time dependent and requires efficient collection and storage systems for economical utilization. Solar air heaters are one of the simplest and cost effective solar energy utilization systems, converts solar radiations into the useful thermal energy being absorbed by fluid medium which can be stored and utilized for various heating and drying applications. The thermal efficiency of solar air heaters is found to be low due to low heat transfer coefficient on the air side. Attempts have been made to enhance the heat transfer rate from the absorber plate to air by extending surfaces in the form of fins so that larger surface area could be available for convection to compensate the lower values of heat transfer coefficient. However the enhancement in heat transfer rate is accompanied by severe pressure drop penalty. In another approach, heat transfer coefficient has been significantly enhanced by providing artificial roughness on absorber plate surface exposed to air. In this approach, the turbulence is promoted by roughened surfaces only in the viscous sub layer region to obtain heat transfer enhancement at the cost of moderate friction penalty. Several roughness geometries have been tested so far with the aim to obtain maximum heat transfer enhancement with consumption of least pumping power. Varun et al. [1] presented a review on roughness geometries used in solar air heaters wherein they discussed the outcomes of different studies concerning with heat transfer enhancement by the use of artificial roughness and showcased the correlations developed by various investigators for prediction of heat transfer and friction factor in roughened solar air heater ducts. In another study, Hans et al. [2] reviewed the roughness element geometries employed by various investigators to improve the thermal performance of solar air heaters. In view of finding optimal roughness pattern, 11 distinct roughness geometries have been compared on the basis of thermohydraulic performance. An attempt has been made by Bhushan et al. [3] to categorize and review the roughness geometries used for creating artificail roughness in solar air heaters. Heat transfer coefficient and friction factor correlations developed by various investigators for roughened duct of solar air heaters have also been reported in this study.

However, there is a need to review the literature on application of roughness elements of different shapes and relative arrangements besides the notable investigation techniques in order to envisage the direction of future researches. The present study presents a detailed review on roughness geometries tested by various investigators and provides critical analysis of data on heat transfer and friction characteristics of roughened surface to facilitate designers and investigators to make best use of it in future.

## 2. Thermal network and performance of conventional solar air heaters

Figure 1 illustrates the thermal network of conventional flat plate solar air heater. The thermal performance of flat plate solar air heater could be observed by considering the energy balance between solar energy absorbed by absorber plate and useful thermal energy output of the system accompanied by some losses. The energy balance equation could be written as follows

$$Q_a = A_p\left[IR(\tau\alpha)_e\right] = Q_u + Q_l \tag{1}$$

Where Qa is the energy absorbed by the absorber plate, Ap is the area of the absorber plate, I is the intensity of insolation, R is the conversion factor to convert radiation on horizontal surface to that on the absorber plane, (τα)e is the effective transmittance absorptance product of the glass cover-absorber plate combination, Qu is the useful energy gain and Ql is energy loss from the collector.

The useful energy gain can be expressed in terms of inlet air temperature Ti and other system and operating parameters as:

$$Q_u = A_p F_R\left[IR(\tau\alpha)_e - U_l(T_i - T_a)\right] \tag{2}$$

Where $F_R$ is given by:

$$F_R = \frac{\dot{m}c_p}{A_p U_l}\left[1 - \exp\left(-\frac{F' U_l A_p}{\dot{m}c_p}\right)\right] \tag{3}$$

Where FR is the collector heat removal factor which indicates the thermal resistance meet by the absorbed solar energy in reaching to the flowing air. Ul is the overall loss coefficient and Ti and Ta are the inlet air and ambient temperatures respectively. F' is termed as collector efficiency factor which provides the relative measurement of thermal resistance between absorber plate and ambient air to that of thermal resistance between the air flowing through collector and the ambient air.

Collector efficiency factor (F' ) is expressed as:

$$F' = \frac{1}{\left(1 + \frac{U_l}{h_e}\right)} \tag{4}$$

Where $h_e$ is the effective heat transfer coefficient between the absorber plate and flowing air.

The thermal efficiency of the collector is the ratio of useful heat gain to the incident solar energy falling on the collector.

Therefore

$$\eta_{th} = \frac{Q_u}{IA_p} = F_R\left[(\tau\alpha)_e - \frac{U_l(T_i - T_a)}{I}\right] \tag{5}$$

According to the above equation, the thermal efficiency of the solar collector could be improved by increasing the value of FR which depends on collector efficiency factor F'. By enhancing the heat transfer coefficient between absorber plate and air, higher values of F' could be achieved. Roughening of absorber surface has been found to be the convenient and effective technique to enhance the convective heat transfer rates from the absorber surface to air.

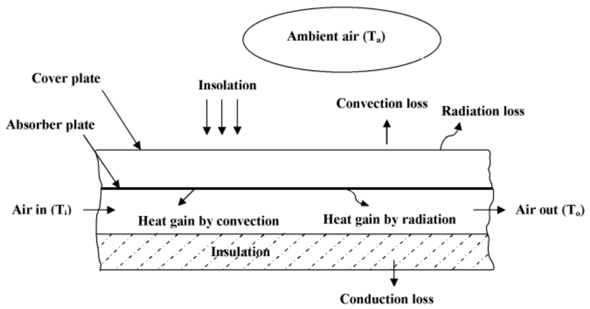



**Fig. 1.** Thermal network of flat plate solar air heater

## 3. Conception of artificial roughness

The concept of artificial roughness was first applied by Joule [4] to enhance heat transfer coefficients for in-tube condensation of steam and since then many experimental investigations were carried out on the application of artificial roughness in the areas of cooling of gas turbine blades, electronic equipments, nuclear reactors, and compact heat exchangers etc. Webb and Eckert [5] conducted experiential study of turbulent air flow in tubes roughened with rectangular repeated ribs and deduced heat transfer and friction factor correlations based on the law of wall similarity and application of the heat-momentum transfer analogy. They considered relative roughness height of 0.01- 0.04 at a relative roughness pitch of 10 - 40 and range of Prandtl number of 0.71-37.6 for this study. Lewis [6] introduced new efficiency parameter for optimizing thermohydraulic performance of roughened surfaces with respect to smooth surfaces.

The experimental study carried out by Han [7 -10] in search of the effect of rib shape, angle of attack, pitch to height ratio and spacing in square duct with two opposite rib roughened wall revealed that the maximum value of heat transfer and friction factor occurs for the ribs oriented at $45^o$ angle with a relative roughness pitch of 10. Han et al. [11] investigated the effect of parallel and V-shaped broken rib orientation on the local heat transfer distribution and pressure drop in a square channel with two opposite ribbed walls and reported that $60^o$ staggered discrete V-shaped ribs provide higher heat transfer than parallel discrete ribs. Han et al. [7] investigated the effect of angle of attack and apex orientation in case of V-shaped ribs and found that $45^o$ and $60^o$ V-shaped ribs facing upward show higher heat transfer rates compared to corresponding V-shaped ribs facing downward. They observed that V-shaped ribs facing upward forms two pairs of rotating cells along each divergent axis of rib, while in the case of V-shaped ribs facing downward, two pairs of counterrotating cells merge resulting in a higher pressure drop and lower heat transfer.

Ravigururajan and Bergles [12] developed general statistical correlations for heat transfer and pressure drop for single-phase turbulent flow in tubes roughened with semicircular, circular, rectangular and triangular shape ribs. Liou and Hwang [13] conducted experimental study on heat transfer and friction for turbulent flow through channel with two opposite walls roughened with semicircular, square and triangular shape ribs. They reported that among three types of ribs, the square rib yielded the maximum 1.9-2.7 fold increase in average Nusselt number while friction factor increased by 7-15 folds respectively.

Zhang et al. and Kiml et al. [14, 15] reported that the thermal performance of V-shaped ribs with $60^o$ angle of attack is better than that of 45o angle for the same range of flow parameters. Lau et al. [16] observed that the replacement of continuous transverse ribs by inclined ribs in a square duct results in higher turbulence near the roughened wall due to interaction of the primary and secondary flows which goes in favour of better thermal performance. Lau et al. [17, 18] studied the heat transfer and friction characteristics of fully developed flow in a square duct with transverse and inclined discrete ribs. They reported that a five-piece discrete rib with $90^o$ angle of attack shows 10-15% higher heat-transfer coefficient as compared to the $90^o$ continuous ribs, whereas inclined discrete ribs give 10-20% higher heat transfer than that of the $90^o$ discrete rib. Lau et al. [17], Taslim et al. [19] and Olssom and Sunden [20] investigated the effect of V-shaped ribs in square channel and found fair enhancement in heat transfer as compared to inclined and transverse ribs. They observed that V-shaped ribs pointing downward have a much higher heat transfer coefficient because the warm air being pumped toward the rib leading region increases the apex region heat transfer coefficients as compared to that of the leading end region. Gao and Sunden [21] also reported that V-shaped ribs pointing downward perform better than the ribs pointing upward in rectangular ducts.

A study by Hu and Shen [22] presented the effect of inclined discrete ribs with and without groove and revealed that the performance of inclined discrete rib without groove has been found best arrangement. In a recent study, Cho et al. [23] investigated the effect of a gap in inclined ribs on heat transfer for a fluid flow through square duct and reported that a gap in the inclined rib accelerates the flow and enhances the local turbulence, which results in an increase in the heat transfer. They reported that the inclined rib with a downstream gap shows significant enhancement in heat transfer compared to that of continuous inclined rib arrangement. Moon et al. [24] investigated effect of channel height on heat transfer in a rectangular duct with a dimpled surface and observed enhancement in heat transfer by about 2.1 times regardless of channel height and friction factor of 1.6 - 2.0 times that of smooth channel. Mahmood and Ligrani [25,26] measured local heat transfer on opposite walls with dimple type roughness with various temperature ratios having ratio of channel height to dimple print diameter of 0.5. They observed that the vortex structures augment local Nusselt number near downstream rim of each dimple. Burgess et al. [27] conducted an experimental study to investigate effect of dimple depth on heat transfer with aspect ratio of 8 and for Reynolds number range of 12000-70000 and reported that Nusselt number increases with increase in dimple depth.

Sang et al. [28] investigated heat transfer with dimple/protrusion arrays in a rectangular duct with low Reynolds number range and observed heat transfer enhancement of 14 and 7 times for double protrusion wall and double dimpled wall at Reynolds number of 1000. However at high Reynolds number of 10000, enhancement level observed was from 2 to 3. Chang et al. [29] examined heat transfer characteristics for four sets of dimpled channels with Reynolds number ranging from 1500 to 11000 and determined effect of dimpled arrangement, fin length to channel hydraulic diameter ratio and Reynolds number on heat transfer over the dimpled fin channel. Varun et al. [30] also reported different investigations on roughness geometries carried out in heat exchangers as well as in air heaters. Application of artificial roughness on one broad wall i.e. absorber plate of solar air heaters to improve thermal performance found its root from the above investigations.

## 4. Methodology of artificial roughness

When air flows over a heated surface, a thin layer called sub layer exists beneath the core turbulent flow region in which the flow remains predominantly laminar due to supremacy of viscous effects. Due to presence of this viscous sub layer over the heated surface, the heat transfer rates from surface to air is poor due to low values of heat transfer coefficient. The application of artificial roughness on the heated surface induces local wall turbulence due to separation and reattachment of flow between two consecutive roughness elements. The turbulence promoted by roughness elements significantly enhances the heat extraction rates from the heated surface. It has been observed that the region affected by this turbulence depends mainly on the height of roughness element. If the roughness height is greater in comparison to thickness of laminar sub layer, the turbulence created by roughened surface promotes disturbances in the turbulent core region and raises fluid friction which results in higher pumping power requirements. To ascertain benefit of artificial roughness in promoting higher heat transfer rates at the cost of minimum friction losses, it is desirable to create turbulence only in the vicinity of heat transferring surface, i.e., in the viscous sub layer region. Figure 2 shows the flow behavior in the viscous sublayer region due to presence of repeated roughness elements.

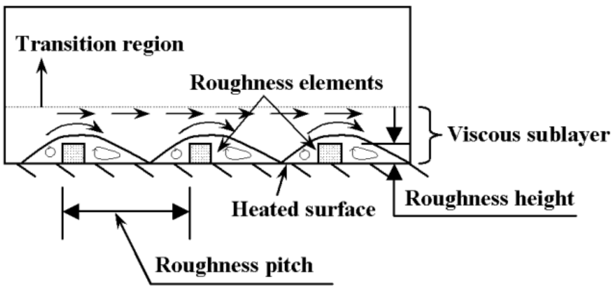



**Fig. 2.** Effect of roughness elements on flow field

## 5. Experimental approach

Performance estimation of artificially roughened solar air heater requires generation of experimental data on heat transfer and friction for specified roughness geometry. To ensure the reliability of experimental data it is essential to perform experimentation on validated experimental setup under standard test conditions.

Figure 3 shows the schematic diagram of indoor experimental setup to generate data on heat transfer and friction in artificially roughened solar air heater as per the guidelines of ASHRAE standard 93-77 [64].

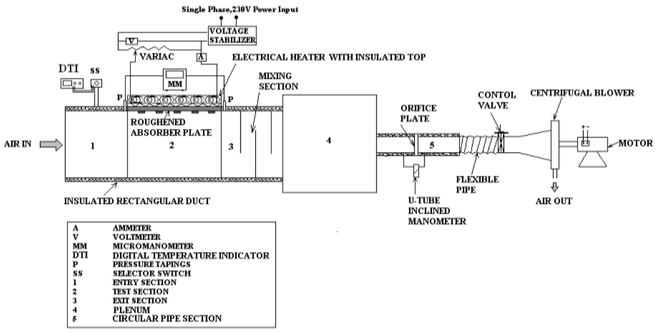



**Fig. 3.** Schematic diagram of experimental setup

The main elements of the system comprise of insulated rectangular duct with metallic artificially roughened absorber plate. Uniform heat flux is supplied over the top surface of the plate by means of electrical heater and the bottom surface is modified by providing artificial roughness elements. The duct is connected to a circular pipe which accommodates the flow measurement device and flow control valve. The other end of pipe is connected to the suction side of centrifugal blower which exhales the air to the surrounding thus forming an open loop system.

The temperature of absorber plate at different locations and inlet and exit temperatures of air are measured with the aid of thermocouples. Pressure drop across the test section of the duct is measured to find the friction losses during the flow over roughened test section.

## 6. Roughness geometries used in Solar air heaters

### *6.1. Transverse ribs in the form of small diameter wires*

Prasad and Mullick [31] were the first who introduced the application of artificial roughness in the form of small diameter wire attached on the underside of absorber plate to improve the thermal performance of solar air heater for drying purposes. Relative roughness height and relative roughness pitch of 0.019 and 12.7, respectively, with wire diameter of 0.84 mm were considered for this study.

Prasad and Saini [32] investigated the effect of protrusions from underside of absorber surface in the form of small diameter wires on heat transfer and friction factor for fully developed turbulent flow in a solar air heater duct. The investigations were carried out for the relative roughness pitch of 10, 15, and 20 and relative roughness height of 0.020, 0.027 and 0.033 to predict the effect of height and pitch of the roughness elements on heat transfer and friction. Study shows that both Nusselt number and friction factor increase with increasing relative roughness height but the rate of heat transfer enhancement diminishes with increase in relative roughness height while the rate of increase of friction factor was found to be nearly constant. The maximum value of Nusselt number and friction factor were reported to be 2.38 and 4.25, respectively, at the pitch of 10. The study unfolds that the reattachment of free shear layer is necessary between the consecutive ribs for heat transfer enhancement. It was observed that for the relative roughness pitch less than 8, reattachment of free shear layer does not occur while the pitch beyond 10 produces lesser number of reattachment points, leads to poor heat transfer rates. The study also suggests that the net benefit of artificial roughness could be exploited till it disturbs the region up to transition limit without disturbing the turbulent core region. It finds that the roughness height should be equal to or slightly higher than the laminar sublayer thickness to achieve the desired results.

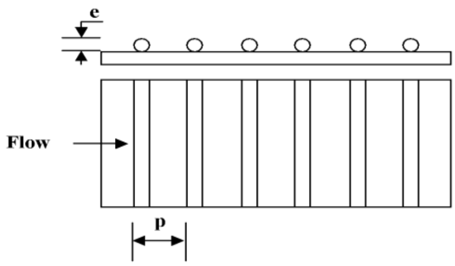



**Fig. 4.** Transverse ribs [32]

### 6.2. *Transverse and inclined ribs in the form of small diameter wires*

Gupta et al. [33] investigated the effect of transverse and inclined wire roughness on fluid flow characteristics for solar air heater. They achieved maximum heat transfer coefficient at an angle of attack of $60^{o}$and pitch of 10. The thermal efficiency of the system is enhanced by a factor of 1.16 - 1.25. The study suggests that the system should be operated for Reynolds number ranges from 13000 to19000 in order to achieve healthier thermal performance.

It has been stated that the effective efficiency increases as the insolation increases for the Reynolds number higher than 10000. Maximum enhancement inheat transfer coefficient and friction factor in roughened duct was reported to be 1.8 and 2.7 times of smooth duct at an angle of attack of 600 and 700 respectively. The roughened surfaces with relative roughness height (e/D) of 0.033 corresponding to Reynolds number (Re) around 14,000 unveiled the best thermohydraulic performance in the range of parameters investigated. The investigation emphasized that the secondary flow rolling along the inclined rib is responsible for higher heat transfer rates.

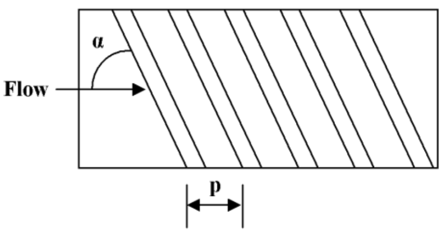



**Fig. 5.** Inclined ribs [33]

### 6.3. *Roughness in the form of expanded metal mesh*

Saini and Saini [34] explored the heat transfer and friction characteristics for flow inside a large aspect ratio rectangular duct with roughness in the form of expanded metal mesh geometry. They reported that the average Nusselt number attains maximum value at the relative longway length of mesh (L/e) of 46.87 and relative shortway length (S/e) of 25 at an angle of attack of $61.9^{o}$. The maximum friction occurs for an angle of attack of $72^{o}$ for relative longway length of 71.87 and relative shortway length of 15.62. The maximum enhancement in Nusselt number and friction factor were found to be 4 and 5, respectively.

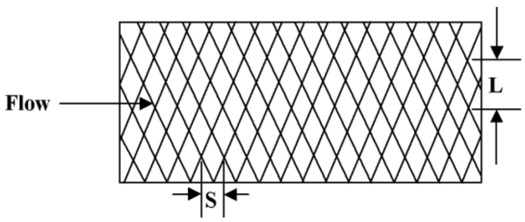



**Fig. 6.** Wire mesh roughness [34]

### 6.4. *Repeated chamfered rib roughness*

Karwa et al. [35] conducted experimental study to predict the thermo-hydraulic performance of solar air heater duct roughened with chamfered rib roughness applied in transverse direction with respect to the flow of air. The chamfered ribs on one broad wall yields up to about two-fold and three-fold increase in the Stanton number and the friction factor, respectively, for the range of Reynolds number (3000-20000), chamfer angle ($-15^{o}$-$+18^{o}$), relative roughness height (0.014-0.033), relative roughness pitch (4.6-8.5) and duct aspect ratio (4.65-12). Square ribs with $15^{o}$ chamfer angle observe maximum heat transfer rates and highest friction.

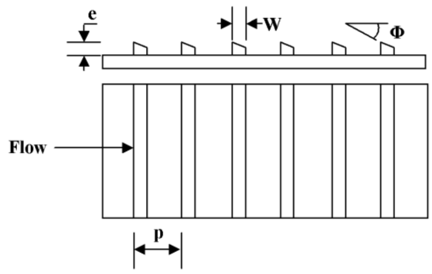



**Fig. 7.** Chamfered ribs [35]

It is seen that Stanton number and friction factor values increase with increase in the chamfer angle from -15 to $+15^{o}$ and beyond $+15^{o}$ chamfer angle, the rate of change of Stanton number and friction factor decrease. The heat transfer function increases with the increase in the aspect ratio from 4.65-9.66 and the roughness function decreases with the increase in the aspect ratio from 4.65-7.75 and thereafter both the functions attain nearly a constant value. It has been pointed out that positive chamfer encourages frequent shedding of vortices causing greater heat removal from the surface and higher frictional losses while in case of negative chamfer, the Stanton number and friction factor decreases due to suppression of the shedding of vortices as a result of skimmed flow over the ribs.

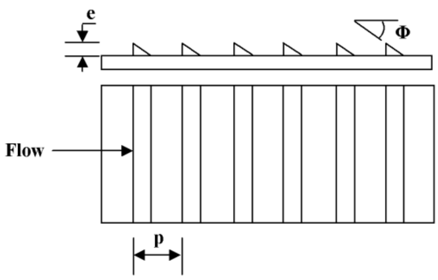



**Fig. 8.** Wedge shape ribs [36]

### 6.5. *Wedge shape transverse rib roughness*

Bhagoria et al. [36] applied wedge shaped transverse repeated rib roughness on one broad heated wall of solar air heater duct and generated data pertinent to friction and heat transfer. They reported that the presence of wedge shape ribs yield maximum enhancement in Nusselt number and friction factorby about 2.4 and 5.3 times as compared to smooth duct. The maximum Nusselt number was obtained at a relative roughness pitch of about 7.57 and decreases further with increasing values of the relative roughness pitch from 7.57 to 12.12, while the friction factor continuously decreases as relative roughness pitch increases from 5.67 to 12.12. The

Nusselt number increases and attains maximum value at a wedge angle of about 10° and then sharply decreases with increasing wedge angle beyond 10° while the friction factor increases as the wedge angle increases. Wedge shape ribs showed significant enhancement in heat transfer over square and chamfered ribs due to relatively lesser chances of eddy formation downstream of the ribs.

### *6.6. V shaped rib roughness*

Momin et al. [37] performed experimentations on flow through duct roughened with V-shape ribs attached to the underside of one broad wall of the duct, to collect data on heat transfer and fluid flow characteristics. They observed that the Nusselt number increases whereas the friction factor decreases with an increase of Reynolds number. It was found that for relative roughness height of 0.034 and for angle of attack of 60°, the V-shaped ribs enhance the values of Nusselt number by 1.14 and 2.30 times over inclined ribs and smooth plate case at Reynolds number of 17034. The rate of increase of Nusselt number with an increase in Reynolds number is lower than the rate of increase of friction factor as the re-attachment of free shear layer might not occur at higher Reynolds number. The thermo-hydraulic performance parameter increases with increasing the angle of attack of flow and the maxima occurs at an angle of attack of 60°. Study shows that besides flow separations and reattachments, the secondary flow moving along the two limbs of V-rib roughness contributes much in obtaining higher values of Nusselt number and friction factor.

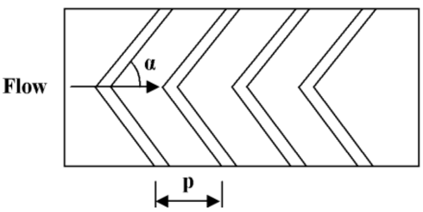



**Fig. 9.** V shape ribs [37]

### *6.7. Transverse, Inclined, V-continuous and V-discrete ribs*

An experimental study conducted by Karwa [38] on heat transfer and friction in a high aspect ratio duct with transverse, inclined, V-up continuous, and V-down continuous, V-up discrete ribs, and V-down discrete ribs shows that the enhancement in Stanton number over smooth duct was found to be 65-90%, 87-112%, 102-137%, 110-147%, 93-134%, 102-142% respectively. Study revealed that the V-down discrete rib roughness secured the best thermal performance for the same power consumption.

It has been reported that in V-up continuous rib pattern, the divided secondary flow directed towards the side walls which allows higher heat extraction rates in the central region while in case of V-up continuous rib pattern, the secondary flow is towards the central axis where it interacts with the axial flow creating additional turbulence leading to

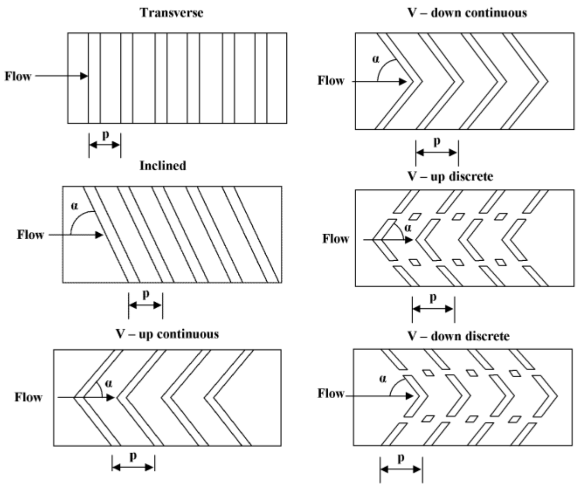



**Fig. 10.** Roughness geometries investigated by Karwa [38]

higher heat transfer rates. The study also emphasized that discrete rib arrangements have lower friction losses as compared to continuous ribs due to change in flow behavior as result of modified secondary flow field.

### *6.8. Broken transverse rib roughness*

Sahu and Bhagoria [39] varied the pitch for 90o broken transverse rib roughness and examined its effect on thermal performance of solar air heater. Experiments were conducted for the range of Reynolds number 3000–12000, pitch of ribs from 10–30 mm, roughness height 1.5 mm and aspect ratio of eight. They observed that the Nusselt number increases sharply at low Reynolds number and remain constant for higher values of Reynolds number. The maximum enhancement in heat transfer was reported at the pitch of 20 mm. It has been highlighted that smooth duct performs better than the roughened duct at low Reynolds number (below 5000). Experimental results revealed that roughened absorber plates increase the heat transfer coefficient 1.25–1.4 times as compared to smooth rectangular duct and maximum thermal efficiency of roughened solar air heater lying in the range of 51–83.5%, depending upon the flow conditions.

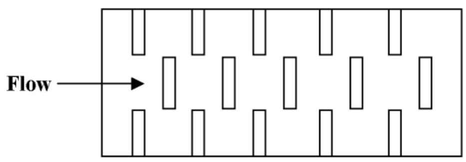



**Fig. 11.** Broken transverse ribs [39]

### *6.9. Rib-grooved roughness*

Jaurker et al. [40] experimentally generated the friction and heat transfer data for turbulent flow through a rectangular duct with rib-grooved transverse repeated rib roughness produced on one broad heated wall. The application of rib grooved artificial roughness enhances Nusselt number up to 2.7 times while the friction factor rises up to 3.6 times as compared to smooth surface in the range of parameters investigated. The maximum heat transfer and friction factor were observed at relative roughness pitch of about 6 and relative groove position of 0.4. The additional vortices induced in and around the grooves were found to be responsible for the higher turbulence intensity between the ribs results in higher heat transfer rates.

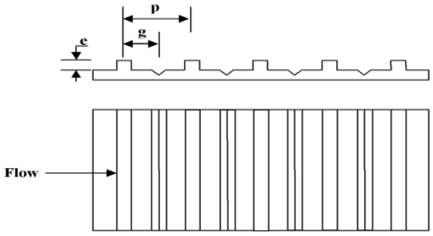



**Fig. 12.** Rib grooved roughness [40]

*6.10. Metal grit ribs*

Karmare et al. [41] experimentally determined the thermal performance of a rectangular duct with metal grit rib roughness employed on heated wall of solar air heater duct. The enhancement in the Nusselt number and friction factor in case of roughened duct was observed nearly 200 and 300%, respectively, in comparison to smooth duct. The maximum heat transfer rate was reported for the set of roughness parameters (l/s = 1.72, e/Dh = 0.044, p/e = 17.5), and highest friction factor was observed for the set of roughness parameters (l/s = 1.72, e/Dh = 0.044, p/e = 12.5). For the roughened surface, rate of increase in Nusselt number with the Reynolds number was registered more than that of smooth surface. At low Reynolds number, roughened and smooth both surfaces have similar values of Nusselt number while at higher Reynolds number, the roughened surfaces owned higher Nusselt number as compared to smooth surface.

It has been stated that at lower Reynolds number the laminar sublayer region is comparatively thick and therefore contribution of roughness elements is insignificant. The laminar sublayer thickness is reduced at higher Reynolds number which improves the effectiveness of roughened surface in promoting higher disturbances in viscous zone leads to higher heat transfer rates. In addition to this, local contribution to the heat removal process by the vortices originated from the roughness element is joined by frequent shedding of vortices cause additional loss of energy resulting in higher heat transfer and fluid friction.

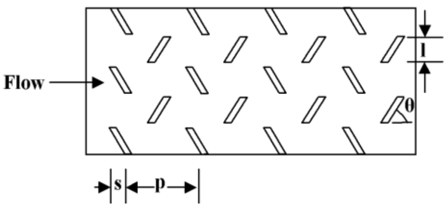



**Fig. 13.** Metal grit ribs [41]

*6.11. Inclinedbroken rib roughness*

Aharwal et al. [42] carried out experimental investigation on the performance of solar air heater ducts, having the absorber plate with inclined rib, provided with and without a gap. The increase in Nusselt number and friction factor was in the range of 1.48–2.59 times and 2.26–2.9 times of the smooth duct, respectively, for the range of Reynolds numbers from 3000 to 18,000. The Nusselt number and friction factor were found to be highest corresponding to a relative gap position of 0.25 and a relative gap width of 1.0. The thermo-hydraulic performance parameter also shows maximum value for the above set of parameters. It was observed that due to gap in inclined rib the secondary flow along the rib joins the main flow through gap accelerating the flow field behind the rib, which energizes the retarded boundary layer flow along the surface and enhances the heat transfer rates.

It was mentioned that the relative gap width beyond 1.0 reduces the flow velocities through the gap which reduces heat transfer rates as compared to the continuous ribs. While the relative gap width lower than 1.0, shrinks the passage for secondary flow release which results in low turbulence behind the gap and hence reduces the heat transfer rates. In case of relative gap position of 0.25, the contribution of secondary flow in energizing the main flow through the gap and recirculation loop in the remaining part of the rib is maximum, which produces highest values of Nusselt nuber.

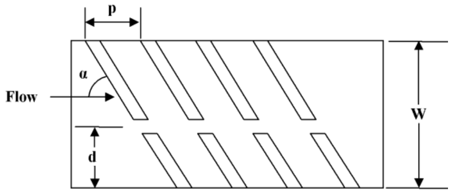



**Fig. 14.** Inclined broken ribs [42]

*6.12. Combined inclined and transverse ribs*

Varun et al.[43] in their experimental study, used a combination of transverse and inclined ribs as roughness geometry and examined the thermal performance for the range of Reynolds number (Re) 2000–14000, pitch of ribs ( p), 5–13mm, roughness height (e), 1.6mm and aspect ratio (W/H) of 10. Results show that the collector roughened with this type of roughness provides best performance at relative roughness pitch (p/e) of 8.

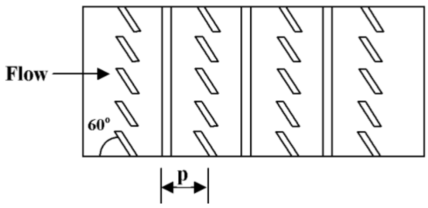



**Fig. 15.** Combined inclined and transverse ribs [43]

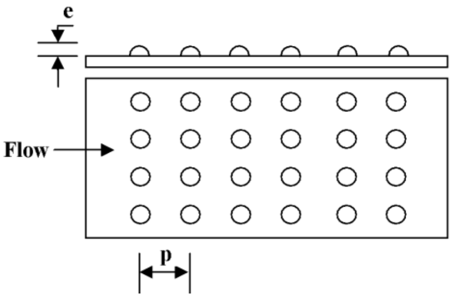



**Fig. 16.** Dimple shape roughness [44]

*6.13. Dimple shape roughness*

Saini et al. [44] conducted experimentation on dimple-shape roughness geometry provided on to the underside of absorber plate of the solar air heater duct, and generated heat transfer and fluid friction data. The maximum value of Nusselt number has been obtained at relative roughness height (e/D) of 0.0379 and relative pitch (p/e) of 10. While minimum value of friction factor has been found correspond to relative roughness height (e/D) of 0.0289 and relative pitch (p/e) of 10.

*6.14. Arc shape roughness*

Saini and Saini [45] investigated the performance of solar air heater duct roughened with arc shape roughness elements. Experimentations were conducted to predict the effect of various roughness parameters such as relative roughness height, relative arc angle on heat transfer coefficient and friction factor for the range of Reynolds number (Re) from 2000 and 17,000, Relative roughness height (e/D) 0.0213 to 0.0422 and relative arc angle (α/90) 0.3333 to 0.6666. The study showed considerable enhancement in heat transfer by applying arc-shape parallel wire roughness geometry in solar air duct. The maximum enhancement in Nusselt number has been obtained as 3.80 times that of smooth surface corresponding to the relative arc angle of 0.3333 and relative roughness height of 0.0422. However, the increment in friction factor corresponding to these parameters has been observed 1.75 times only.

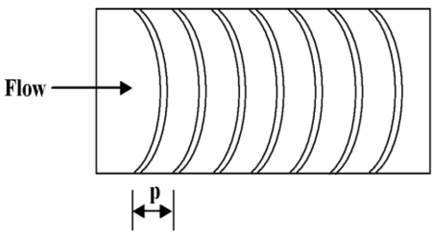



**Fig. 17.** Arc shape roughness [45]

*6.15. Chamfered rib groove roughness*

Layek et al. [46] in their experimental work on heat and fluid flow characteristics of artificially roughened surface with chamfered rib-grooved roughness yielded a maximum of about 2.6 fold and 3.35 fold increase in the Nusselt number and friction factor respectively as compared to smooth surface. The maximum enhancement in heat transfer has been obtained at chamfer angle of 18o whereas the friction factor increases monotonously with increase in chamfer angle. The thermo-hydraulic performance is lying between 1.4 and 1.76 for the range of parameters investigated. It was reported that more frequent shedding of vortices with chamfering of the ribs and an additional vortex created by the presence of grooves between ribs results in greater heat removal from the surface accompanied by higher frictional losses.

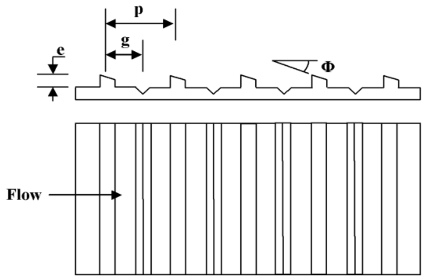



**Fig. 18.** Chamfered rib groove roughness [46]

Results showed that for higher Reynolds number the friction factor attains an asymptotic value and remains almost constant with further increase in Reynolds number. The study concludes that formation of grooves near reattachment points induces greater turbulence which significantly enhances the heat transfer rates.

*6.16. Discrete W shaped roughness*

Arvind kumar et al. [47] investigated the heat transfer and friction characteristics of artificially roughened ducts with discrete W shaped ribs. The study revealed that the flow separation and generation of secondary flow cells followed by the movement of resulting vortices were contributed simultaneously in enhancing heat transfer rates by promoting higher wall turbulence. The maximum enhancement of Nusselt number was found reported as 2.16 times that for smooth duct for corresponding to angles of attack of 60° and relative roughness height of 0.0338. For relative roughness height of 0.0338, the maximum friction factor was found to be 2.75 times that of smooth duct for angles of attack of 60°.

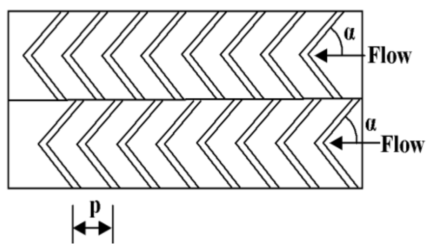



**Fig. 19.** Discrete W shape ribs [47]

*6.17. Multi V ribs*

Recently Hans et al. [48] introduced multiple V-rib roughness and performed extensive experimentation to collect data on heat transfer and fluid flow characteristics of a roughened duct. Maximum enhancement in Nusselt number and friction factor due to presence of multi V-rib roughness has been found to be 6 and 5 times, respectively, in comparison to the smooth duct. The maximum enhancement in heat transfer has been achieved corresponding to relative roughness width (W/w) of 6 while friction factor attains maximum value for relative roughness width (W/w) of 10. It has been observed that Nusselt number and friction factor both attain maxima corresponding to 60° angle of attack.

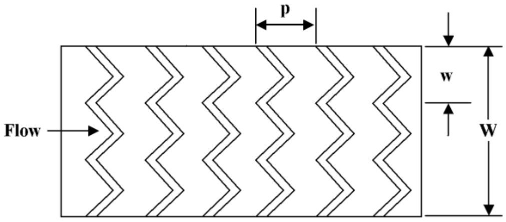



**Fig. 20.** Multi V rib roughness [48]

The study revealed that as the relative roughness width (W/w) is increased, heat transfer also increases and attains maximum value at relative roughness width (W/w) of 6, and thereafter it decreases. The reason may be due to higher number of leading ends and secondary flow cells produced favorable conditions to incur maximum wall turbulence at relative roughness width (W/w) of 6. Beyond relative roughness width (W/w) of 6, results in separation of flow from rib top surface and formation of boundary layer impeding heat transfer rates. However, friction factor goes on increasing with increase in relative roughness width (W/w) on account of vortices formed by separation of flow.

## 7. Performance of roughness geometries

Studies on artificially roughened solar air duct predominantly concerned with the effect of shape and arrangement of roughness elements on heat transfer and friction. It is therefore reasonable to compare the performance of some distinct roughness geometries separately on the above basis.

Figure 21 shows the variation in Nusselt number as a function of Reynolds number for roughness geometries tested by different investigators. Considerable jump in Nusselt number can be seen in all the cases as compared to the smooth surface. Plot reveals that the value of Nusselt number is highest in case of multi V shape roughness and lowest in case of transverse ribs for all the values of Reynolds number. Multi V shape roughness shows discrete performance among all the roughness geometries and provides almost double increment in heat transfer rates in comparison to other roughness geometries. It can be observed that roughness geometries other than V shape rib, multi V shape rib, inclined rib and transverse rib roughness showing that the Nusselt number increases at faster rate for higher values of Reynolds number. While in case of V shape rib, multi V shape rib, inclined rib and transverse rib roughness, a steep rise in Nusselt number has been registered up to around 12000 of Reynolds number and after that the increment rate of Nusselt number slightly diminishes. On the basis of this observation one can choose preferably wire mesh roughness or arc shape roughness for systems working at higher flow rates to achieve higher heat transfer rates. But for the systems with low and moderate flow rates, multi V shape roughness is the best option as far as heat transfer enhancement is concerned. A significant rise in Nusselt number can be seen in case of inclined ribs with gap as compared to continuous inclined rib roughness. It is therefore expected that the geometries like V shape ribs and multi V shape ribs with gap at suitable location could exhibit considerable enhancement in heat transfer rates as compared to continuous V or multi V ribs. It is believed that the secondary flow cells are responsible for higher heat transfer rates. Since inclined rib has only one secondary flow cell while V rib got two secondary flow cells and multi V rib roughness can generate multiple secondary flow cells, and therefore multi V rib roughness shows maximum heat transfer augmentation.

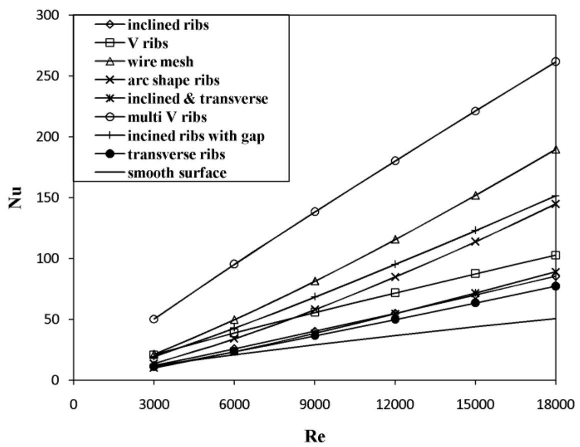



**Fig. 21.** Effect of Reynolds number on Nusselt number for different rib geometries

Figure 22 shows the effect of Reynolds number on Nusselt number enhancement ratio for different roughness geometries. One can observe that multi V rib roughness outperforms the other roughness geometries with the enhancement ratio greater than 4. The enhancement ratio

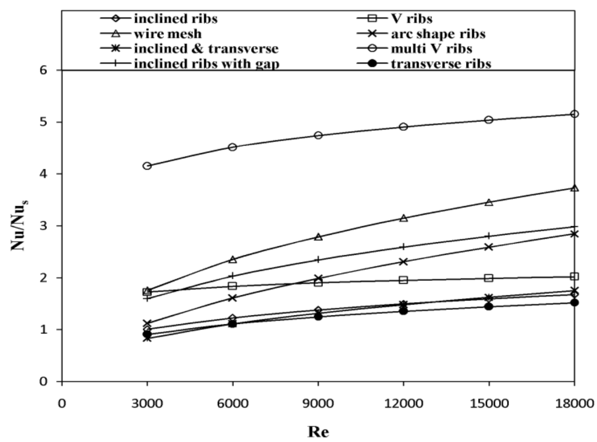



**Fig 22.** Effect of Reynolds number on Enhancement ratio for different rib geometries

increases with increase in Reynolds number in almost all the cases but highest increment rate is registered in case of wire mesh roughness and arc shape roughness while the enhancement ratio for V rib roughness remains constant for the entire range of Reynolds number.

It is clear from the figure that Multi V rib roughness provides double enhancement as compared to the single V rib roughness. The combination of inclined and transverse rib roughness performance lies in the close vicinity of inclined roughness though it can provide higher rate of enhancement.

Figure 23 shows the comparison of various roughness geometries on the basis of thermohydraulic performance. It can be seen that the multi V ribs outperforms the other roughness geometries. The arc shape and wire mesh roughness are showing similar kind of performance over the entire range of Reynolds number.

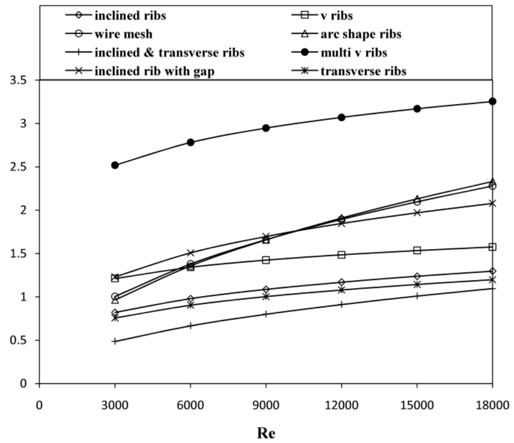



**Fig. 23.** Effect of Reynolds number on Thermohydraulic performance for different rib geometries

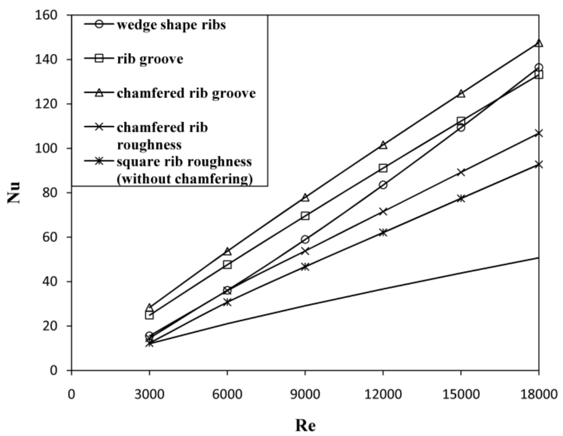



**Fig. 24.** Effect of Reynolds number on Nusselt number for different rib shapes

The performance of broken inclined ribs is far better than continuous inclined rib roughness and also showing clear distinction while compared with V shape rib roughness geometry. The value of thermohydraulic performance parameter for the combination of inclined and transverse rib roughness is lying in the range where the enhancement in heat transfer is achieved at the cost of severe pressure drop penalty.

Figure 24 shows the comparison of different transverse roughness geometry shapes on the basis of Nusselt number variation corresponding to Reynolds number.

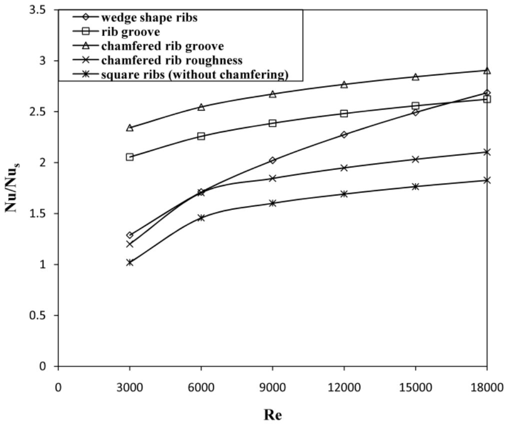



**Fig. 25.** Effect of Reynolds number on Enhancement ratio for different rib shapes

It shows that chamfered rib roughness is better than square rib roughness and observes significant increment in heat transfer rates when combined with grooves. The rate of increase in Nusselt number is of greater order in case of wedge shape ribs while comparing with chamfered rib roughness. The higher heat transfer rates are observed at low to moderate Reynolds number in case of rib groove and chamfered rib groove roughness which fulfill the need of suitable shape and combination of roughness geometry for solar air heater applications.

Figure 25 shows the variation of Nusselt number enhancement ratio with Reynolds number for different types of roughness shapes. A significant enhancement in heat transfer rates are achieved at lower Reynolds number by combining grooves with square and chamfered rib roughness as compared to square and chamfered ribs without grooves. The rate of enhancement in heat transfer is highest in case of wedge shape roughness which finds its suitability for system operating at comparatively higher range of Reynolds number.

Figure 26 shows the variation of thermohydraulic performance with respect to Reynolds number for different

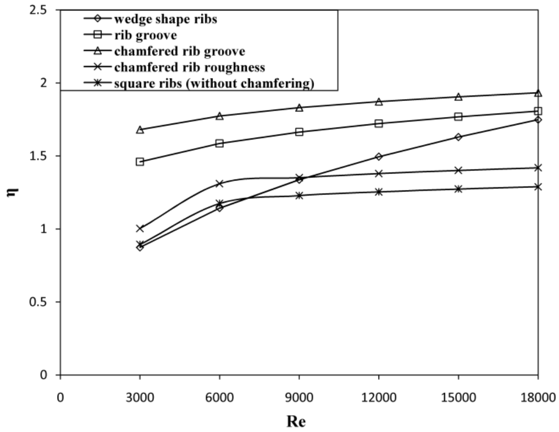



**Fig. 26.** Effect of Reynolds number on Thermohydraulic performance for different rib shapes

rib shapes. It follows the same trend as observed in the last plot. Chamfered rib roughness combined with groove shows best thermohydraulic performance and steep rise in performance is obtained in case of Wedge shape rib roughness.

## 8. Flow visualisation techniques

Gao and Sunden [49] employed PIV system to investigate the effect of rib inclination on the flow field. Experiments have been conducted for flow through rectangular ducts with ribs inclined at 30°, 45°, 60°, and 90° for ratio of rib height to hydraulic diameter as 0.06, pitch to rib height ratio as 10 and Reynolds number as 5800. They analyzed that the strength of the secondary flow is highest at an angle of attack of 60°. The PIV results obtained for 90° ribs were in good agreement with the data collected with Laser Doppler Velocimetry (LDV) system. Bonhoff et al. [50] conducted numerical and experimental investigation to analyze the flow field inside a roughened channel with 45° inclined ribs on two opposite surfaces using PIV system. The results generated by numerical model were validated with the observations of PIV system. Gao and Sunden [51] have analyzed and generated results for the flow field in a rectangular duct with 60º inclined and V-shaped ribs by using PIV system and reported that the PIV technique is capable of obtaining the detailed flow structures between two consecutive ribs with reasonable accuracy.

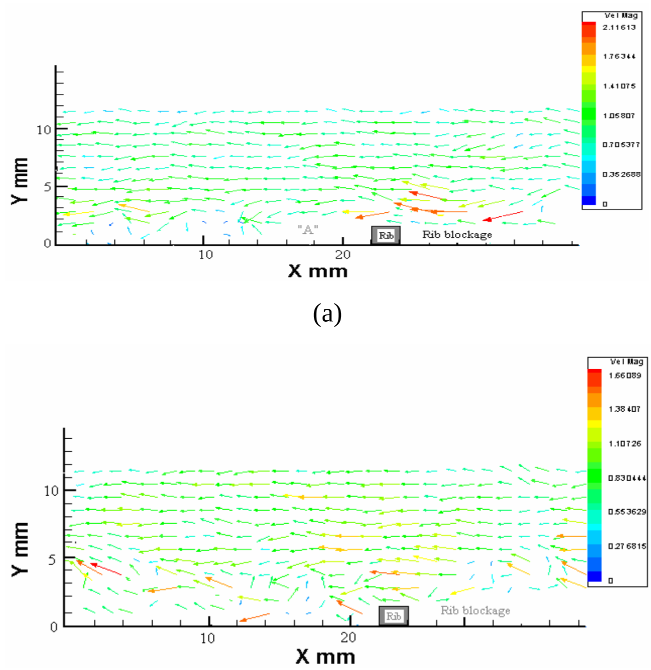



**Fig. 27. (a)** Velocity profile for flow over an inclined continuous rib **(b)** Velocity profile for flow over an inclined rib with a gap

Aharwal et al. [52] investigated the effect of gap width on heat transfer in a repeated inclined ribbed wall in view of obtaining higher thermal performance of solar air heaters. Experiments have been carried out for the relative gap width of 0.5 to 2.0 for the Reynolds number range of 3000-18000 at relative roughness height of 0.037. It was found that inclined rib with relative gap width (g/e) of 1.0 provides highest heat transfer enhancement compared to the other relative gap width.

A Two Dimensional Particle Image velocimetry (2-D PIV) system has been used to investigate the effect of gap width on the flow field and the turbulence intensity at different relative gap width has been measured. They observed that the highest value of turbulent intensity is measured at the relative gap width of 1.0 which promotes higher surface heat extraction rates in the region behind the gap. The flow field visualized through the side plane 'A' inside a rectangular duct roughened with an inclined continuous rib is shown in Figure 27 (a). The flow is seen to be separated at the rib which appears to be reattached with the surface at around 5 times of the rib height. It is seen that the separated boundary layer behind the rib results in a dead zone as marked "A", which is non participative in heat transfer process. The flow field for a gap in the inclined rib, visualized through plane 'A' passing through the middle of the gap is presented in Figure 27 (b) for similar flow conditions as applied to continuous rib. It shows that the dead zone "A" shown in case of continuous rib completely vanished due to the release of main and secondary flow through the gap.

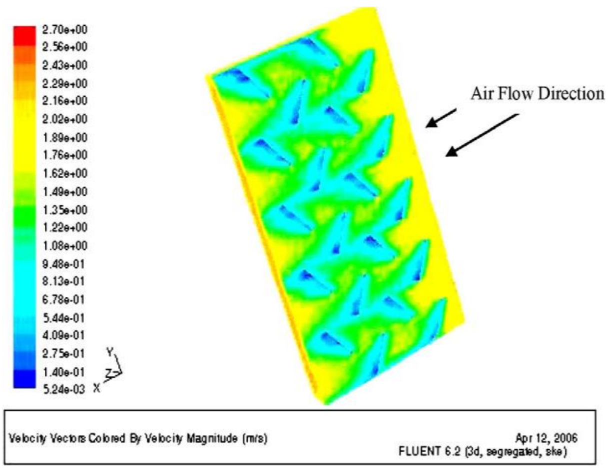
Lockett et al. [53] employed the non-invasive optical method of holographic interferometry to investigate the effect of rib shape on heat transfer in turbulent flow field. They observed that the heat transfer distribution depends on the Reynolds number for the rounded rib, but independent for square rib geometry. In both cases, the minimum heat transfer occurred at the base of the rear facing rib wall. Han et al. [54] applied modeling clay to fill the corners of the rectangular ribs to create two distinct geometries. It was found that the clay has much more modest effect on the heat transfer coefficients than on the friction factor, and the influence of the rib shape on the heat transfer coefficient disappears at higher Reynolds number where the flow is in the completely rough regime. Arman and Rabas [55] used a nonorthogonal, body-fitted numerical code to predict the effect of the rib shape on the thermal-hydraulic performance in a circular tube. They reported that the increase in heat transfer were in the order of arc, semicircle, sine and trapezoid.

## 9. Computational techniques

Computational approaches have also been adopted to predict the effect of rib roughened duct on fluid flow and heat transfer. Despite of limited repeatability and accuracy, calculations using turbulence models are having wide acceptability since they could produce fairly good results with less effort and time. Studies using standard k-ε turbulence model have been suffered with serious errors as it does not account for anisotropy in the turbulence when simulating ribbed duct flows. Liou et al. [56] has identified the most significant problem areas with high anisotropy, namely the separated flow and large recirculation zone downstream of the ribs, producing faulty results due to incorrect prediction of Reynolds stresses in these regions.

Ooi et al. [57] compared a standard k-ε model with the one equation Spalart–Allmaras (SA) model and the v2 − f model and found that the v2 − f model was capable of producing best results compared to others, though it could not predict and analyze the secondary flows with accuracy, which have a significant impact on the heat transfer. Liou et al. [58] used a k-ε-A model instead of k-ε model, which is a hybrid model of standard k-ε model and Algebraic Stress Model (ASM), and was able to account for the anisotropy of turbulence.

A study comparing several k-ω models with a Reynolds Stress Model (RSM) with enhanced wall treatment showed good agreement between the RSM results and experiments in predicting mean flow and smooth side wall heat transfer with some differences in predicting ribbed wall heat transfer [59]. In addition Layek et al. [60] performed large-eddy simulations (LES) to assess the occurrence of hot spots on the rib roughened wall by investigating the effects of rib shapes on the local heat transfer. They produced local heat transfer results for prediction of the flow and heat transfer characteristics in ribbed passages, and compared the thermal performance of the two types of ribbed duct. Simulated results are compared with the experimental measurements, and showed fairly good agreement.




(a)

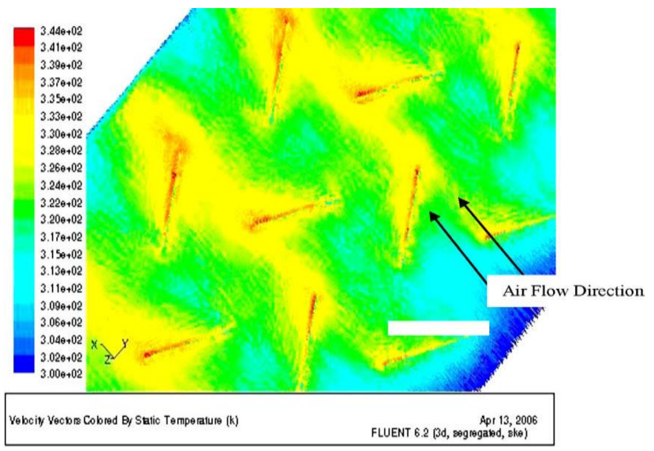



**Fig. 28. (a)** Velocity vectors of air on rib roughened plate **(b)** Static temperature on rib roughened plate

Karmare et al. [61] conducted CFD investigation of heat transfer and fluid flow in a rectangular duct with metal grit ribs as roughness elements employed on one broad wall of a solar air heater. The circular, triangular and square shape rib grits with the angle of attack of 54o, 56 o, 58 o, 60 o and 62 o has been tested where one broad wall is subjected to uniform heat flux with flow range of Reynolds numbers 3600–17,000 is maintained. The CFD results showed good agreements with experimental results. Amongst the different shape and orientations analyzed, square ribs with 58 o angle of attack produced best results. The maximum enhancement in the heat transfer is found to be about 1.3 times of smooth surface. Figure 28 (a) shows velocity vectors of air on the surface of roughened test plate. Figure 28 (b) shows static temperature on the rough surface.It shows that temperature of air increasing along the length of the rib, because the heated primary air moves along the rib due to inclination. Hence the temperature at the tip of the rib is higher. As heated primary air is removed by the rib then low temperature secondary air comes in contact with hot surface results in higher heat transfer coefficient.

Sharad [62] analyzed the effects of arc shaped roughness geometry on heat transfer and friction by computational fluid dynamics (CFD). The results obtained from different models of computational analysis were compared with Dittus–Boelter empirical relationship for smooth duct and it was found that the Renormalization (RNG) k- ε model had least variation as compared to other CFD models.

Chaube et al. [63] carried out a computational analysis using Fluent 6.1 software to investigate the flow and heat transfer characteristics of two-dimensional rib roughened rectangular ducts with one wall subjected to uniform heat flux of 1100 W/ m2. They compared the predictions of different turbulence models with experimental results available in the literature and reported good matching of experimental results and predictions of shear stress transport (SST) K-ω turbulence model. They used SST K- ω turbulence model for analyzing the performance of nine different roughness elements and compared the predictions on the basis of heat transfer enhancement, friction characteristics and performance index. The results obtained from two-dimensional model were reported to be closer to the experimental results and these models required less memory and computational time as compared to three-dimensional models. The highest heat transfer was reported in case of chamfered ribs, however, the best performance index was found to be with rectangular rib of size 3mm× 5 mm within the range of parameters investigated.

## 10. Conclusion

- Use of artificially roughened surfaces with different type of roughness geometries is found to be the most effective technique to enhance the heat transfer rates from the heated surface to flowing fluid at the cost of moderate rise in fluid friction.

- Roughness in the form of ribs, wire matrix, and dimples were mainly suggested by different investigators to achieve better thermal performance. Among all, rib roughness was found the best performer as far as thermal performance is concerned.

- Multi V rib roughness is one of the most acclaimed forms of roughness tested recently showed distinct performance in case of solar air heaters.

- The mechanism of fluid turbulence and heat transfer in case of different rib geometries has been presented in various studies employing computational, PIV and other flow visualization techniques but still clear interpretation of secondary fluid flow and its effects on flow field has to be explored in order to select the best fit roughness geometry in solar air heaters.

- Experimentations on rib roughened duct employing roughened plates are to be prepared by either fixation of wires or machining on metallic plates. Both techniques seem to be obsolete as they involve tedious fabrication steps which involve higher cost and prolonged time duration. Therefore computational approach could be adopted for the analysis of flow field as it could produce results with reasonably good accuracy with consumption of least time and money.

- Existing turbulence models could be modified in order to assess the flow behavior close to actual phenomenon and generate heat transfer and friction results with good accuracy.

## 11. Scope for Future work

- Broken inclined rib roughness performed excessively well as compared to continuous inclined rib roughness and therefore the performance of multi V rib roughness shall be improved by introducing gap at suitable location.

- Multi V ribs arranged in transverse direction which were tested recently showed outstanding performance and in future these multi V rib arrays could be arranged inclined to the direction of flow and subsequently arrays arranged in V type fashion could be tested in the quest of higher heat transfer rates.

- It was observed that wedge shape rib roughness performed better than other rib shapes as far as heat transfer enhancement rate is concerned and therefore wedge shape ribs combined with grooves could be the better combination in order to get better enhancement rates in heat transfer.

- Ribs of different shapes like chamfered, wedge are to be used in making inclined and V type roughness geometries to achieve higher heat transfer enhancements.

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