# Heat Transfer Enhancement by Turbulators: A Review

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***Abstract*- The use of turbulators in different forms of ribs, baffles, delta winglets, obstacles, vortex generator, rings and perforated blocks/baffles is an effective way to improve the performance of heat exchangers and solar air heaters. Investigators studied the effect of these turbulators for heat transfer and friction characteristics in air ducts. An attempt has been made in this paper to carry out an extensive literature review of turbulators used to investigate heat transfer augmentation and flow structure in air ducts. Based on the review it is found that perforation in ribs/baffles/blocks and combination of combined rib and delta winglet leads to the better thermo-hydraulic performance.**

**Index terms- Ribs, Nusselt Number, Heat transfer, Pressure Drop, CFD**

## I. INTRODUCTION

A common technique to enhance heat transfer is by installing turbulence promotors (rib turbulators) on the smooth walls. The rib turbulators generate secondary flows which increase near-wall shear in the vicinity of the ribs and these secondary flows also interact with channel side walls to increase turbulent transport of energy from relatively hotter walls by forming vortex or vortices. One application of rib turbulators as heat transfer enhancement technique is found in gas turbine airfoils. The gas turbine airfoils are subjected to elevated heat loads on both pressure and suction side walls. Hence rib turbulators are installed on pressure and suction side internal walls to increase heat transfer in order to increase the heat transfer rates between internal walls and coolant. Rib turbulators also result in increase in wetted surface area which enhances the overall conductance. Several studies have been carried out in the past on heat transfer enhancement by various cooling designs, such as, ribbed channel with bleed holes, ribbed channel with grooves, rib dimpled compound channels, jet impingement, jet impingement with effusion holes, dimpled channel, jet impingement onto dimpled target surface etc. The use of artificial roughness on a surface is an effective technique to enhance heat transfer coefficient also has good application in design and development of efficient solar air heaters. Many investigations have demonstrated the effects of different rib configurations on heat transfer coefficient between absorbers plate and air flowing in solar air heaters, numerically and experimentally, in order to improve the heat transfer capability of solar air heater ducts. Heat transfer enhancement due to rib turbulators is affected by several parameters such as rib angle-of-attack, channel aspect ratio, rib pitch-to-height ratio, blockage ratio, and rib shape. Investigations on these aspects of rib turbulator design have been reported. In the past, researchers have studied the flow characteristics of ribbed duct, both, experimentally and numerically. Some studies focused on the relative arrangement of ribs, e.g. parallel, staggered, and criss-cross. Several investigations have been carried out in the past on rib turbulator as a method of enhancing heat transfer [1-11].

## II. LITERATURE REVIEW ON RIB TURBULATORS FOR HEAT TRANSFER ENHANCEMENT

Heat transfer coefficients and friction factors of turbulent flow in uniformly heated rectangular channels with transverse ribs applied on two opposite walls were studied systematically for varying rib pitches ($5 \leq p/e \leq 40$), relative roughnesses ($0.0021 \leq e/Dh \leq 0.102$), and channel aspect ratios ($1/4 \leq W/H \leq 4$) at Reynolds numbers of $7.0 \times 104 \leq Re \leq 9.0 \times 104$ by Han [12, 13]. The results showed that the spatially averaged Nusselt number ratio at the rib-roughened wall Nur/Nu0 decreases and friction factor ratio fr/f0 increases for increasing Reynolds numbers within the fully developed flow regime, with the Dittus–Boelter Nusselt number correlation of Nu0 = 0.023•Re°8•Pr°4and the Blasius friction factor correlation of f0B = 0.046•Re−0.2 for smooth circular channel flows. Liou and Hwang [14] carried out wall temperature and pressure drop measurements for fully developed turbulent flow in channels with ribs (e/Dh = 0.063, 0.081, 0.106; p/e = 10, 15, 20) on two opposite walls at Reynolds numbers of 5.0 × 103 ≤ Re ≤ 5.4 × 104. It was found that the friction factor and spatially averaged Nusselt number increase with increasing relative roughness (at a constant Reynolds number) and decrease with increasing pitch (for a constant rib height). The pitch-to-rib-height ratio of p/e =10 was found to be the optimum for reaching maximum heat transfer. Chandra et al. [15, 16] investigated the effect of varying numbers of rib-roughened walls (e/Dh = 0.0625; p/e = 8) in a square channel on heat transfer and flow resistance at Reynolds numbers of 1.0 × 104 ≤ Re ≤ 8.0 × 104. It was shown that heat transfer at the rib-roughened wall was slightly enhanced with increasing numbers of ribroughened walls, but the enormous rise in the corresponding pressure drop leads to a significant efficiency decrease. Thermal performance decreases with increasing Reynolds numbers and with increasing number of rib-roughened walls. Thermal performance of the ribbed cooling channels was evaluated using the performance equation formulated by Webb and Eckert [17]. Experimental studies focusing on both heat transfer and flow characteristics in rib-roughened channels are less frequent. Arts et al. [18] and Rau et al. [19] reported turbulent flow and heat transfer in channels roughened by transverse ribs (e/Dh = 0.1; 6 ≤ p/e ≤ 12) with square cross section on two opposite walls and on one wall, respectively, at a Reynolds number of Re = 3.0 × 104. A counter-rotating vortex pair was found in the cross-sectional channel planes and it was shown that the ribs deflect the fluid toward the smooth side walls and heat transfer is enhanced in regions where the flow impinges on the wall. Vortex shedding, shear layer separation, development of recirculation regions, and secondary flow motion dominate the heat transfer enhancement and are significantly affected by the rib crosssectional shape, rib configuration, and rib angle of attack. Han et al. [20] examined the effect of the inclined angle of trapezoid ribs on heat transfer and friction factor in a channel with ribs on two opposite channel walls at Reynolds numbers of 3.0 × 103 ≤ Re ≤ 3.0 × 104. The rib cross section was found to influence the friction factor, while the influence on the Stanton number was moderate and disappeared at high Reynolds numbers. Lockett and Collins [21] studied local Nusselt number distribution in channels roughened by transverse ribs (e/Dh = 0.067; p/e = 7.1) of square and round-edged square cross section. The results showed that the Reynolds number-dependent heat transfer distribution was more uniform for the round-edged square ribs than for the conventional square ribs. The averaged heat transfer coefficient increased by a factor of about 1.68–2.24 compared to smooth channel flows. Liou and Hwang [22] investigated local heat transfer enhancement and thermal-hydraulic performance of rib-roughened channels with transverse ribs (e/Dh = 0.08; 8 ≤ p/ e ≤ 20) of triangular, semi-circular, and square cross-sectional shape at Reynolds numbers of 7.8 × 104 ≤ Re ≤ 5.0 × 104. The most uniform heat transfer distribution was determined for the semi-circular ribs and the highest heat transfer was obtained for the square ribs. Spatially averaged heat transfer at the rib-roughened wall was increased by about 60–110%, 70–120%, and 90–170% for the semicircular, triangular, and square ribs, respectively. The corresponding averaged friction factor was increased by about 4–8, 5–10, and 7–15 times, respectively. The square ribs produced the highest variation of local heat transfer and heat transfer deterioration. In contrast to this, Ahn [23] found that the highest heat transfer enhancement occurs for ribs with triangular cross section. The thermalhydraulics of rib-roughened channels with semicircular, triangular, and square ribs (e/Dh = 0.0476, p/e = 8) on one channel wall were analyzed at Reynolds numbers of 1.0 × 104 ≤ Re ≤ 7.0 × 104.Ina further study, Ahn et al. [24] performed large eddy simulations (LES) of turbulent flow in a duct with transverse ribs (e/Dh = 0.1, p/e = 10) with square and semicircular cross section applied on two opposite walls at a Reynolds number of Re = 3.0 × 104. The time-averaged results show, that vortices in the backward-facing and forward-facing rib-corner can be reduced for the semicircular rib and, thus, the local heat transfer peaks in the vicinity of the ribs are weaker for the semicircle ribs than for the square ribs. The averaged Nusselt number is in a comparable range for the square and semicircle ribs, but the reduced friction factor of the latter yield better thermal performance. Chung et al. [25] analyzed the effect of varying pitch-to-rib-height-ratio of semicircle ribs (e/Dh = 0.07; 8 ≤ p/e ≤ 14) in a rectangular channel on Nusselt number and thermal performance at Reynolds numbers of 1.1 × 104 ≤ Re ≤ 2.4 × 104. The results obtained by RANS simulations show that semicircle ribs provide higher Nusselt number ratios than square ribs. The highest heat transfer occurred for a ribpitch of p/e = 11, whereas the thermal performance was found to be the best for p/e = 13. Wang and Sundén [26] examined the effect of rib pitch and Reynolds number on the Nusselt number and friction factor in channels roughened with transverse ribs (e/Dh = 0.1, 8 ≤ p/e ≤ 15) of four different cross-sectional shapes. The highest heat transfer and pressure drop occurred for the trapezoidal ribs with the decreasing rib height. The effect of rib configuration on heat transfer and flow characteristics was analyzed in several studies. Nusselt numbers and friction factors of turbulent flow in heated square channels with crossed, parallel, and upstream or downstream directed V-shaped ribs (e/Dh = 0.0625; p/e = 20) and with varying rib angles of attack at Reynolds numbers of 1.5 × 104 ≤ Re ≤ 9.0 × 104 were studied by Han et al. [27]. Results indicated that the upstream directed V-shaped ribs provide the highest heat transfer, whereas the downstream directed V-shaped ribs produce the highest pressure drop. Further experiments were conducted by Lau et al. [28] to study the effect of angle of attack of upstream and downstream directed V-shaped ribs (e/Dh = 0.0625; p/e = 10, 20) and of parallel ribs on two opposite walls in a square channel on turbulent heat transfer and friction factor at Reynolds numbers of 1.0 × 104 ≤ Re ≤ 6.0 × 104. The results revealed that the upstream directed V-shaped ribs provide a higher heat transfer than the downstream directed V-shaped ribs. Compared to transverse ribs, the upstream directed 45° and 60° V-shaped ribs increased heat transfer and pressure drop. The highest heat transfer at the rib-roughened wall and at the smooth side wall and the best thermal performance were obtained for the upstream directed 60° V-shaped ribs. Taslim et al. [29] investigated the thermal-hydraulics of channels roughened on two opposite walls by variously shaped ribs (e/Dh = 0.083, 0.125, 0.167; p/e = 10) in a staggered array at Reynolds numbers of 1.5 × 104 ≤ Re ≤ 9.0 × 104. In contrast to the aforementioned studies, they reported that 45° parallel ribs performed better than 45° V-shaped ribs. With the exception of small relative roughness, the highest heat transfer is reached by the 45° parallel ribs. The V-shaped and parallel ribs cause stronger secondary flows and, thus, thermal mixing is intensified. High heat transfer coincides with regions of downward flow velocities and reduced heat transfer occurs in regions of upward flow velocities behind the rib, as was pointed out by Liou et al. [30]. They compared thermal performances of twelve different ribs at a Reynolds number of Re = 1.2 × 104. It was found that the rib configuration-dependent secondary flow motion dominates heat transfer at the rib-roughened wall. Under the constraint of constant pumping power, the upstream directed 45° V-shaped rib and a 45° delta wing rib reached the best thermal performances, with the latter showing favorable uniformity of local heat transfer distribution. Averaged friction factors of the 45° and the 60° V-shaped ribs differed considerably by a factor of 2.4 contrary to the small difference reported by Han et al. [27]. Jia et al. [31] performed RANS and LES for different V-shaped ribs (e/Dh = 0.0625, 0.1, 0.125; p/e = 10). It was shown that the V-shaped ribs induce a strong counter-rotation vortex pair and the vortex rotation direction depends on upstream and downstream rib direction. Regions of highest Nusselt number occurred at the rib roughened wall for the downstream directed ribs and at the smooth side walls for the upstream directed ribs. The local Nusselt number on the rib-roughened surface varied significantly for upstream and downstream directed V-shaped ribs. Sharma et al. [32] analyzed the heat transfer and friction factor development of upstream directed V-shaped ribs for varying angle of attack, ribpitch and relative roughness (30°-75°, 5 ≤ p/e ≤ 20, 0.022 ≤ e/Dh ≤ 0.044) at Reynolds numbers ranging from 4.9 × 103 to 1.4 × 104. As reported for transverse ribs by Liou and Hwang [25], the Nusselt number increases with increasing relative roughness and decreases with increasing pitch for p/e ≥ 10 at a constant Reynolds number. The 60° V-shaped ribs with p/e = 10 and e/Dh ≤ 0.044 provide the highest heat transfer and the highest friction factor. Satta et al. [33] investigated mean flow velocities and local heat transfer in a rectangular channel with 45°

parallel ribs on one or two opposite channel walls at Reynolds numbers of Re = 2.9 × 104. The results showed that one or two cell secondary flow pattern occur, respectively, which transport cold core fluid toward the rib-leading end regions and heated fluid to the opposite trailing end regions. Thus, local heat transfer at the ribroughened channel walls varies significantly in spanwise direction. In a parallel study, Tanda [34] analyzed the effect of varying rib-pitch of 45° parallel ribs (p/e = 6.7, 10, 13.3 and 20) on the thermalhydraulics and heat transfer performance at Reynolds numbers of 9.0 × 103 ≤ Re ≤ 35.5 × 104. Similar to transverse ribs, the heat transfer enhancement is accompanied by an increase in the pressure drop. It was found that under the constraint of constant pumping power, optimum values of rib-pitch varies for one and two sided rib-roughened channels. In recent studies (time-resolved) Particle-Image-Velocimetry (PIV) measurements was applied to provide deep insights into the complex flow motion around ribs at moderate Reynolds numbers under isothermal conditions. Fang et al. [35] investigated the turbulent flow around 30°, 45° and 60° upstream directed V-shaped ribs (e/Dh = 0.1, p/e = 8) in a square channel at Reynolds number of 1.15 × 104 ≤ Re ≤ 1.3 × 104.As a result of the interaction between different mean vorticity components, the secondary flow motion forms a counter rotating vortex pair. It was found that ejection events which were identified by Wang et al. [36] by PIV measurements (e/Dh = 0.2, p/e = 10, Re = 2.2 × 104) as the main contributor to the Reynolds shear stress at the rib-edge of transverse ribs were significantly suppressed for the V-shaped ribs. Compared to transverse ribs, turbulent structures above the V-shaped ribs are less originated in vortex shedding over the ribs due to the strong vertical downward flow motion. In a companion study, Fang et al. [37] investigated the turbulent flow around upward directed V-shaped ribs (45°, 60°, and 90°, e/Dh = 0.1, p/e = 8) at different Reynolds numbers by means of LES. The results revealed that the high friction factors of the V-shaped ribs are cause by an increased in friction and form drag due to the secondary flows.

Many investigations [38–47] have demonstrated the effects of different rib configurations on heat transfer coefficient between absorbers plate and air flowing in solar air heaters, numerically and experimentally, in order to improve the heat transfer capability of solar air heater ducts. It is well known that in a turbulent flow a laminar/viscous sublayer exists in addition to the turbulent core. The artificial roughness on heat transfer surface breaks up the laminar boundary layer of turbulent flow and makes the flow turbulent adjacent to the wall. The artificial roughness that results in the desirable increase in the heat transfer also results in an undesirable increase in the pressure drop due to the increased friction; thus, the design of the flow duct and absorber surface of duct should, therefore, be executed with the objectives of high heat transfer rates and low friction losses. It is therefore desirable that the turbulence must be created only in the region very close to the heat transferring surface i.e. in the viscous sub-layer only where the heat transfer takes place and the core flow should not be unduly disturbed so as to avoid excessive friction losses. This can be done by keeping the height of the roughness elements to be small in comparison with the duct dimensions.

Heat transfer enhancement due to rib turbulators is affected by several parameters such as rib angle-of-attack, channel aspect ratio, rib pitch-to-height ratio, blockage ratio, and rib shape. Investigations on these aspects of rib turbulator design have been reported in [48–57]. In the past, researchers have studied the flow characteristics of ribbed duct, both, experimentally and numerically [58–63]. Some studies focused on the relative arrangement of ribs, e.g. parallel, staggered, and criss-cross [64–66]. Gao and Sunden [64] used particle image velocimetry to reveal the flow characteristics in rectangular channels with aspect ratio of 1:8. Six different rib configurations were experimentally investigated, namely staggered parallel ribs, staggered-single parallel ribs, inline parallel ribs, crossed ribs, V-ribs pointing downstream, and pointing upstream. Ekkad and Han [65] performed a detailed study on heat transfer characteristics with all rib parallel to each other in a non-rotating square channel using transient liquid crystals technique. Lee et al. [66] used naphthalene sublimation technique to measure detailed heat transfer enhancement contours using heat-mass transfer analogy. They tested continuous V shaped ribs and discrete V shaped ribs. Several investigations have been carried out in the past on rib turbulator as a method of enhancing heat transfer.

In past, several numerical investigations have been carried out to understand turbulent heat and fluid flow in rib roughened ducts. Murata and Mochizuki [67] carried out numerical investigation of laminar and turbulent heat transfer in a square duct featuring angled rib turbulators. Lu and Jiang [68] carried out experimental and numerical study on a rectangular channel featuring angled rib turbulators. The authors concluded that the SST k- turbulence model was more suitable for the prediction of turbulent heat transfer compared to RNG k-ε model. Al-Qahtani et al. [69] used Reynolds Stress turbulence model in conjunction with near-wall second –moment turbulence closure to study heat transfer in rotating rectangular channels with rib turbulators. Eiamsa-ard and Promvonge [70] studied four turbulence models for prediction of heat transfer in a rectangular duct featuring grooves. The authors carried out computations using standard k-ε, RNG k-ε, standard k, SST k-ω turbulence models and found that the k-ε model was better than the other turbulence models. For computations of heat transfer in a square duct roughened by discrete V-shaped ribs, Promvonge et al. [71] used RNG k-ε model. Acharya et al. [72] compared nonlinear and standard k-ε models for prediction of periodically developed heat and flow transfer in ribbed duct and concluded that the nonlinear model predicted realistic Reynolds stresses in the core flow region than the standard model. Peng et al. [73] studied different rib shapes numerically using SST k-ω turbulence model. Sewall and Tafti [74] carried out Large Eddy Simulations (LES) on a two-pass rib roughened duct featuring 90° rib turbulators and demonstrated that LES predicted flow and heat transfer was very accurate. However, LES comes at a significant computational cost. In order to maintain a balance between computational accuracy and cost, SST k-ω turbulence model was used in the present study. Further, the choice of the turbulence model in the present study is based on relative comparisons of heat transfer predictions by four other turbulence models and experimentally obtained heat transfer data.

## III. CONCLUSION

It can be concluded that there is definite increase in heat transfer in duct when its surface is roughened. However, the different investigators find different values of increment in heat transfer with increase in friction factor for each experiment. We can see that the increase in heat transfer is achieved but the friction factor is also increasing simultaneously. The various correlations developed for the Nusselt number and friction factor for the range of parameters considered will give insight to the designers and other investigators in their work for finding out the optimum values. These correlations and heat transfer values can help the beginners of this field will help to know about the various geometries used till now and encourage them for investigating new roughness elements or even combinations of already existing elements for their work. The use of artificial roughness on a surface is an effective technique to enhance heat transfer to fluid flowing in the duct. Artificially roughened duct have enhanced rate of heat transfer as compared to the smooth solar air heaters under the same geometric/ operating conditions. It has been found that roughness geometries being used in solar air heaters are of many types depending upon shapes, size, arrangement and orientations of roughness elements on the absorber plate. There are several parameters that characterize the roughness elements, but for duct the most preferred roughness geometry is repeated rib type, which is described by the dimensionless parameters viz. relative roughness height (e/D), relative roughness pitch, (P/e), angle of attack (α) and channel aspect Ratio (W/H) etc. Transverse rib roughness enhances the heat transfer coefficient by flow separation and generation of vortices on the upstream and downstream of rib and reattachment of flow in the inter-rib spaces. It can be concluded that the use of artificial roughness results in higher friction and hence higher pumping power requirements. It is desirable that design of solar air heater should be made in such a way that it should transfer maximum heat energy to the flowing fluid with minimum consumption of blower energy.

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