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RECENT DEVELOPMENTS IN ENHANCEMENT OF HEAT TRANSFER THROUGH ARTIFICIAL ROUGHNESS IN SOLAR AIR HEATER DUCTS

Aditya Takhalate, A. R. Jaurker

Department of Mechanical Engineering, Jabalpur Engineering College, Jabalpur

E-mail of Corresponding Author: adityatakhalate@gmail.com

ABSTRACT

The heat transfer between duct plate and air is low, which leads to higher temperature loss to atmosphere in solar air heaters which result in lower performance of system. It is found that the application of artificial roughness breaks the viscous sub layer in ducts, decrease the thermal resistance and increase the turbulence in flow which enhances the heat transfer rate. The roughness geometry with their parameters and correlations developed by various investigators are tabulated in this paper for suitable selection.

Keywords: solar air heater, artificial roughness, heat transfer, friction factor

INTRODUCTION

A solar air heater generally consists of an absorber plate forming a high aspect ratio through which air to be heated. It is more popular because it utilizes both the beam and diffuse radiation and simple in design, requires little maintenance. The absorber plate is a metal sheet about 1mm in thickness, usually made of GI or steel painted black to obtain higher absorbtivity. Transparent glass sheet having higher transmissivity of thickness 4 to 5 mm is the most commonly used cover plate of the top of the solar air heater. To avoid the heat losses, the sides and bottom are insulated with insulating material like mineral wool or glass wool of thickness 5 to 8cm. Since the heat transfer coefficient of air heater is low which can be improved by active/ passive techniques as listed in table 1.

Artificial Roughness: Artificial roughness is the passive technique of heat transfer enhancement. Further a general classification of ribs according to modifications is listed in table 2.These are defined by non-dimension parameters as follows:-

1) Relative roughness pitch (p/e): It is defined as the ratio of distance between two consecutive ribs to the height of the rib.

2) Relative roughness height (e/D): It is defined as the ratio of rib height to the hydraulic diameter of the duct.

3) Angle of attack (α): It is defined as the inclination of rib with the direction of flow of air.

Three main types of artificial roughness are generally found in literature as follows:

1. Three Dimensional Roughness (Uniform Roughness)
2. Ridge Type Two Dimensional Roughness (Repeated Ribs)
3. Groove Type Two Dimensional Roughness

Table 1: Methods of Heat Transfer Enhancement

PASSIVE TECHNIQUES	ACTIVE TECHNIQUES
Treated surfaces	Mechanical aids
Rough surfaces	Surface vibration
Extended surfaces	Fluid vibration
Displaced enhanced devices	Electrostatic fields
Swirl flow devices	Injection

Table 2: General Classification of Ribs

1. According to Orientation of ribs
   - i. Transverse ribs
   - ii. Inclined ribs
   - iii. Helical ribs
2. According to Cross sectional views
   - i. Circular
   - ii. Rectangular
   - iii. Square
   - iv. Triangular
   - v. Trapezoidal
3. Discrete ribs or Staggered ribs
   - i. Transverse Discrete ribs
   - ii. Inclined Discrete ribs
   - iii. V – shaped Discrete
   - iv. W shaped Discrete
4. Grooves
5. Combined/ compound ribs
   - i. Transverse & Inclined ribs
   - ii. Rib grooves
   - iii. Special shaped rib & grooves
6. Dimple roughness
7. Special Shaped Ribs
   - i. V- shaped ribs
   - ii. W-shapes ribs
   - iii. Wedge Shaped ribs
   - iv. Chamfered
   - v. Arc shaped wire &
   - vi. Multi V ribs
   - vii. Expanded metal mesh

Roughness in the form of sand grains

Nikuradse [1] investigated the effect of roughness on the friction factor and velocity distribution in pipes roughened by sand blasting. Nunner [2] and Dippery and Sebersky [3] developed a friction similarity law and a heat momentum transfer analogy for flow in sand grain roughened tubes.

Roughness in the form of transverse ribs

Ravigururajan and Bergles [4] developed general statistical correlations for heat transfer and pressure drop for four types of roughness, namely semicircular, circular, rectangular and triangular for single-phase turbulent flow in internally ribbed tubes. Han [5] carried out an experimental study in square duct with two opposite rib roughened walls. Son et al. [37] carried out the Particle Image Velocity (PIV)experiments for heat transfer characterization in two-pass square channels with smooth and 90° ribbed walls for high Reynolds number turbulent flow (Re= 30,000).The rib-induced flow turbulence increases the heat transfer mainly because of the enhanced local flow impingement near the rib.

Square transverse, Helically Ribbed, Trapezoidal, Three-Dimensional Ribs.

Firth and Meyer [35] investigated the heat transfer and friction factor performance based on equal pumping power, of four different types of artificially roughened surfaces, namely (i) Square transverse ribbed surface, (ii) Helically ribbed surface. (iii)Trapezoidal ribbed and (iv) Three-dimensional ribs. They found that the helically ribbed surface has a thermal performance which compares closely with the square transverse ribbed surface.

Rectangular, Semicircular and Square cross section ribs

Tanda [11] carried out experimental investigation by using liquid crystal thermography to study of heat transfer from a rectangular channel having width to height ratio of 5 and one surface heated at uniform heat flux and roughened by repeated ribs. The ribs having rectangular or square sections were fixed transverse to the main direction of flow or V-shaped with an angle of 45 or 60 deg relative to flow direction. The results show that for the range of Reynolds number studied, the presence of semicircular, rectangular and square ridges at two walls yield about 4-8 fold, 5-10 fold and 7-15 fold increase in the average friction factor respectively relative to smooth duct. There is about 1.6 to 2.1 fold, 1.7 to 2.2 fold and 1.9 to 2.7 fold increase in the average heat transfer coefficient compared with smooth wall for the semi circular, triangular and square ridge walls respectively.

Square, triangular & trapezoidal

Kamali et al. [18] investigates the effect of square, triangular and trapezoidal ribs with computer code using square duct and the simulation was performed on all four ribs with parameters as relative roughness pitch varies from 8 to 12 and relative roughness height of 0.1 and it was found that the relative roughness pitch of 12 provides the highest heat transfer.

Chamfered Ribs

Layek et al.[14] used chamfered rib groove roughness on one wall of the solar air heater by varying parameters selected and it was found that highest Nusselt number occurs for chamfer angle of 18° but the friction factor increase with chamfer angle. The nusselt number increases 3.24 times and friction increases 3.78 times respectively. The maximum heat transfer enhancement was observed at relative roughness pitch of 6 and relative groove position of 0.4. Again Layek et al.[23] in the next paper investigates the effect of chamfer angle on heat transfer with chamfer angle as 5°, 12°, 15°, 18°, 22° and 30°.It was resulted that the maximum heat transfer occurs at chamfer.

90° Broken Transverse Ribs

Sahu et al.[12]investigated heat transfer on transverse ribs with pitch 10-30mm rib height 1.5mm and it was found that the solar air heater provides maximum efficiency at a pitch of 20mm and heat transfer enhances to 1.4 times as compared to smooth duct.

Inclined Ribs

Han et al. [6] developed a general correlation based on the law of the wall similarity and the application of the heat-momentum analogy for friction and heat transfer to account for rib shape, spacing and angle of attack. In the experimental program, sixteen different geometries were tested for relative roughness pitch of 5,7.5,10,15,and 20, relative roughness height of 0.032,0.042, 0.056, 0.076, and 0.102 and angle of attack of 20°, 45°, 75°, and 90° and Reynolds number from 3000 to 30000. The main conclusions of the investigation are as : For small value of relative roughness pitch (p/e = 5), the flow which separates on each rib does not reattach before it reaches the succeeding rib while a relative roughness pitch value of about

10, the flow does reattach close to the next rib. The maximum value of both the heat transfer coefficient and the friction factor occur at relative roughness pitch (p/e) of 10, and ribs at a 45° angle of attack have superior heat transfer performance at given friction power when compared to ribs at a 90° angle of attack or when compared to sand-grain roughness.

Angled Rib Turbulators

Tanda et al.[27] employed inclined ribs to investigate heat transfer in rectangular channel by using it in one and two walls of the channel. For one ribbed wall the optimum performance was found at relative roughness pitch of 13.33 while for two ribbed wall the optimal performance was observed at relative roughness pitch of 6.66-10 for an angle of attack of 45°.

Inclined Discrete ribs

Aharwal et al. [21] used inclined discrete ribs on absorber plate and the effect of width and gap position has been investigated under the parameters as Reynolds number from 3000 to 18,000 with the relative roughness pitch (P/e) range of 4–10, relative roughness height (e/D) range of 0.018–0.037, and angle of attack (α) ranges from 30–90°. Result shows that Nusselt number increases 2.83 times and friction factor 3.60 times under the range of parameters investigated. The maximum heat transfer occurs at the relative gap position of 0.25 with the relative gap width of 1.0 for the relative roughness pitch of 8.0, angle of attack of 60° and relative roughness height of 0.037. The maximum value of friction factor occurs for discrete transverse ribs with relative roughness pitch of 8.0.

Combined transverse and inclined ribs

Varun et al.[15] investigates the effect of combined transverse and inclined ribs on absorber plate with Reynolds number (Re) ranges from 2000 to 14 000, relative roughness pitch (p/e)ranges from 3–8 and relative roughness height e/D_h= 0.030.Results shows relative roughness pitch of 8 have the maximum thermal efficiency. Again Varun et al [19].shows the effect of inclined and transverse ribs and optimized the efficiency of solar air heater by taguchi method and the maximum value of effective efficiency has been found for relative roughness pitch (p/e) of 8 with parameters as Reynolds number (Re) 2000–14,000, relative roughness pitch (p/e) 3–8 and a fixed value of relative roughness height (e/D) of 0.030.

V Shaped

Lau et al. [7] observed that the average Stanton number is 60, 45 or 30 deg discrete rib case is about 25 to 35 percent higher than in the 90 deg discrete rib case. In all the above configurations, the crossed ribs were found to perform poorly compared with other configurations. Momin et al. [9] investigated 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 at Reynolds number of 17034.

Wedge shaped ribs

Bhagoria et al.[10] used wedge shaped ribs with parameters as Re= 3000–18000,relative roughness height (e/Dh) varies from 0.015–0.033,Relative roughness pitch $60.17\Phi^{-1.0264}$<p/e<12.12 rib wedge angle Φ= 8°–15°,Aspect ratio of duct W/H 5 and it was found Nusselt number increases up to 2.4 times while the friction factor rises up to 5.3 times.

Rib groove

Zhang at el. [38] carried out on experimental investigation on the effect of ribbed-grooved roughness on heat transfer and pressure drop in rectangular channels for Reynolds numbers between 10000 and 50000.They observed that for similar rib height and spacing, the rough side of the ribbed-grooved duct enhances heat transfer 3.4 times while the rough side of the ribbed duct enhances heat transfer 2.4 times. The turbulence is larger for the ribbed-grooved wall than for the ribbed wall. The additional vortices created by the grooves are responsible for the higher turbulence in the ribbed-grooved wall. Jaurker et al. [13] investigated the characteristics of rib-grooved artificial roughness. The parameters having Reynolds number range from 3000 to 21,000, relative roughness height 0.0181–0.0363, relative roughness pitch 4.5–10.0 and groove position to pitch ratio 0.3–0.7. The effect of important parameters on the heat transfer coefficient and friction factor has been discussed. The results shows nusselt number increases 2.7 times and the friction factor increases up to 3.6 times.

U Shaped ribs

Bopche et al. [20] used U shaped turbulators on the absorber plate of solar air heater. It was found that the heat transfer increases 2.82 times while friction increases 3.72 times as compared to that of smooth duct.

Multi V ribs

Hans et al. [26] investigated multiple V ribs and the presence of this rib provides increase in nusselt number and friction factor of 6 and 5 times respectively. The maximum heat transfer occurs at relative roughness width of 6 while maximum friction occurs at relative roughness width of 10 and it was also resulted that the maximum enhancement in nusselt number and friction was found at angle of attack of 60°.Promvonge et al.[24] used multi 60 V baffles on absorber plate to find out the enhancement of heat transfer and it was found that thermal performance with PR=1 and e/H=0.10 results maximum thermal enhancement factor of about 1.87 at lower Reynolds number.

V Discrete

Gao and Sunden [36] carried out an experimental investigation on heat transfer and pressure drop in the cases of two wide rib roughened walls of rectangular duct with three rib configurations parallel ribs and V-shaped ribs pointing upstream or downstream of the main flow direction. The system parameters include e/D_h of 0.06, α of 60 degree and p/e of 10 for the Re ranges from 1000 to 6000.The V-ribs pointing downstream produced the highest heat transfer enhancement and friction factors and provided the best thermal performance over the Reynolds number range tested. Parallel ribs provided better performance than that of V-ribs pointing upstream at high Reynolds numbers. Karwa et al. [25] investigated the effect of V down discrete ribs by using a mathematical model. And it was found that at sufficiently lower mass flow rates i.e. less than $0.04 kgs^{-1}m^2$ the thermal and effective efficiency differ marginally but at higher mass flow rates i.e. greater than $0.045\ kgs^{-1}m^2$ the thermal efficiency is higher than effective efficiency. Singh et al.[30] investigated experimentally the effect of V down discrete ribs on one wall of the duct. The maximum heat transfer is 3.04 times and friction factor is 3.11 times higher to that of smooth duct.

W shaped ribs

Lanjewar et al.[31] used W shaped ribs to increase the performance of air heater and it was resulted that heat transfer increases 2.36 and friction factor increases 2.1 times at an angle of attack of 60°.Kumar et al.[22] employs discrete W ribs on absorber plate having aspect ratio of 8:1.The maximum heat transfer occurs at 30°, 45°, 60° and 75° e/D_h of 0.0168 but at e/D_h of 0.0338 the maximum heat transfer was 1.88, 1.99, 2.16 and 2.08 times as compared to smooth duct. It was also resulted hat there was decrease in friction with increase in reynolds number. The effect of orientation of W shaped ribs on heat transfer was observed by lanjewar et al.[28] and it was found that better thermo hydraulic performance is obtained at α=60°.And it was concluded that W down ribs gives better performance than W up and V ribs.

Dimple shape geometry

Saini et al. [17] used dimple shape geometry as artificial roughness with parameters as Reynolds number 2000 to 12,000, relative roughness height of 0.018 to 0.037 and relative roughness pitch of 8 to 12.The maximum heat transfer was obtained at the relative roughness height of 0.0379 and relative pitch of 10.And the minimum friction was found at relative roughness height of 0.0289 and relative roughness pitch of 10.

ARC shaped wire

Saini et al. [16] investigates the effect of arc shaped wire as artificial roughness on duct and increase in heat transfer is 3.80 time while friction factor increases 1.75 times. The maximum heat transfer occurs at relative arc angle of 0.3333 and relative roughness height of 0.0422.

Table. 4 Heat Transfer And Friction Factor Correlations				
Author[1]	**Optimum value of parameter[2]**	**Range of parameters[3]**	**Geometry used[4]**	**Results on heat transfer and friction/correlations [5]**
Han et al. (1978)	p/e=10, α=45°	p=5,7.5,10, 15, 20 e/D_h=0.032,0.042,0.056, 0.076, and 0.102 α=20°, 45°, 75°, and 90° Re=3000 to 30000	Transverse and inclined	$S_t=\frac{f}{[He^{+}-Re^{+}](2f^{1/2})+2}$ $Re^{+}=4.9(e^{+}/35)^{m}/[(\Phi/90)^{0.35}(10/(p/e))^{n}(\alpha/45)^{0.57}]$
Han and Park (1988)	α=45°	Re=10000-60000,α=90°-30°,W/H=1 to 2 and to 4	Inclined and transverse ribs	$R/[(p/e)/10]^{0.35}(W/H)^{m}]=12.31-27.07(\alpha/90°)+17.86(\alpha/90°)^{2}$ $G=R+\{[\bar{f}+(H/W)($ $\bar{f}-f(FD)]/(2S_t)-1\}/\{[\bar{f}+(H/W)(\bar{f}-\bar{f}(FD))]/2\}^{1/2}$
Han (1984)		Re=7000 to 90000,p/e=10,e/D_h =0.021 to 0.063	Inclined ribs	$f_r=\frac{2}{\left[0.95\frac{p}{e}^{0.53}-2.5ln\frac{2e}{D_e}-2.5-2.5ln\frac{2B}{A+B}\right]}2$ $S_{tr}=\frac{fr/2}{[1+\sqrt{\frac{fr}{2}}[G_H(e^{+},Pr)-R_m(e^{+})]}$
Karwa et al. 2001	e/D_h=0.0441	p/e=4.58-7.09 Ø=15°, e= 0.74-1.68 p=5.25-7.69	integral chamfered rib	$R=\sqrt{\frac{2}{f}}+2.5\ln(2e/D_h)+3.75$ $g=[\{f/(2St)-1\}\sqrt{2/f}+R]$
[1]	**[2]**	**[3]**	**[4]**	**[5]**

Malik et al. 2002	$p/e=7.09$ $e/D_h=0.0256$	$Re=2500–18000$ $e/Dh=0.02–0.034$ $\alpha=30°–90°$	V shaped rib	$Nu_r=0.067 \times (Re)^{0.888} \times (e/D_h)^{0.424} \times (\alpha/60)^{-0.077} \times \exp[-0.782 \times \ln(\alpha/60)^2]$ $f=6.266 \times (Re)^{-0.425} \times (e/D_h)^{0.565} \times (\alpha/60)^{-0.093} \times \exp[-0.719 \times (\ln \alpha/60)^2]$
Bhagoria et al. 2002	$p/e=7.57$ $e/D_{h=}0.033$	*Re* 3000–18000 e/Dh 0.015–0.033 $\alpha=8°–15°$ $p/e=$ $60.17Ø^{-1.0264}$ $<p/e<12.12$	Wedge shape rib	$Nu_r=1.89 \times 10^{-4}(Re)^{1.21}(e/D_h)^{0.426}(p/e)^{2.94}[\exp(0.71(\ln(p/e))^2)]$ $(\Phi/10)^{-0.018}[\exp(-1.50(\ln(\Phi/10))^2)]$
Sahu et al. (2005)	$p/e=13.33$	$Re=3000–12000$ W/H 8.0 e (mm) 1.5 e/D= 0.0338 p=10, 20 and 30	90° broken transverse ribs	Efficiency at p=20mm is 83.5%.
Jaurker et al. (2006)	$p/e=6$ $e/D_h=0.036$	Re 3000–21,000 (e/D) 0.0181–0.0363 (p/e) 4.5–10.0 (g/p) 0.3–0.7	Rib grooved	$f = 0.001227(Re)^{-0.199}(e/D)^{0.585}(p/e)^{7.19}(g/p)^{0.645} \times \exp(-1.854\{\ln(p/e)\}^2) \times \exp(1.513\{\ln(g/p)\}^2 + 0.8662\{\ln(g/p)\}^3)$ $Nu = 0.002062\ Re^{0.936}(e/D)^{0.349}(p/e)^{3.318} \times \exp[-0.868\{\ln(p/e)\}^2](g/p)^{1.108} \times \exp[2.486\{\ln(g/p)\}^2 + 1.406\{\ln(g/p)\}^3]$
Layek et al. (2007)	$p/e=8$ $e/D_h=0.04$	$Re=3000–21,000$ $p/e=4.5–10$ $e/D_{h=}0.022–0.04$ $Ø=5°–30°$ $g/p=0.3–0.6$	transverse chamfered rib-groove	$Nu=0.00225Re^{0.92}(e/D_h)^{0.52}(p/e)^{1.72}(g/p)^{-1.21}\Phi^{1.24} \times [\exp\{-0.22(\ln\Phi)^2\}][\exp\{-0.46(\ln p/e)^2\}] \times [\exp\{-0.74(\ln g/p)^2\}]$ $f=0.00245\ Re^{-0.124}(e/D_h)^{0.365}(p/e)^{4.32}(g/p)^{-1.124} \times \exp[0.005\Phi]\exp[-1.09(\ln p/e)^2]\exp[0.68(\ln g/p)^2]$
[1]	**[2]**	**[3]**	**[4]**	**[5]**
Varun et al. (2008)	$p/e=8$	Re=2000-14000 p/e=3-8 $e/D_h= 0.030$	Inclined and transverse ribs	$Nu = 0.0006 \times Re^{1.213} \times (p/e)^{0.0104}$ $f=1.0858 \times Re^{-0.3685} \times (p/e)^{0.0114}$

Saini et al. (2008)	$e/D_h=0.0422$	W/H= 12 P/e= 10 e/d= 0.0213–0.0422 (α/90)= 0.3333–0.6666 Re=2000–17 000	arc-shaped wire	$Nu=0.001047\ Re^{1.3186}\ (e/d)^{0.3772}(\alpha/90)^{-0.1198}$ $f=0.14408Re^{-0.17103}(e/d)^{0.1765}(\alpha/90)^{0.1185}$
Kamali et al. (2008)	p/e=12	$e/D_h=0.1$,p/e=8-12,	square, triangular & trapezoidal	Trapezoidal ribs have highest value of heat transfer for range of parameters studied.
Saini et al. (2008)	$e/D_h=0.0379$ p/e=10	Re=2000 to 12,000, e/D=0.018 to 0.037 p/e=8 to 12.	Dimple shape geometry	$Nu=5.2 \times 10^{-4}Re^{1.27}(p/e)^{3.15}x[\exp(-2.12)(\log(p/e))^2](e/D)^{0.033} x[\exp(-1.30)(\log(e/D))^2]$ $f=0.642Re^{-0.423}(p/e)^{-0.465}[\exp(0.054)(\log(p/e))^2] \times (e/D)^{-0.0214}[\exp(0.840)(\log(e/D))^2]$
Varun et al. (2009)	p/e=8	Re= 2000–14,000 p/e=3 to 8 e/D= 0.030	Transverse and inclined ribs	$Nu=0.0006 \times Re^{1.213} \times (p/e)^{0.0104}$ $f=1.0858 \times Re^{-0.3685} \times (p/e)^{0.0114}$
Bopche et al. (2009)	$e/D_h=0.03986$ p/e =8.3	Re=3800 to 18000 e/Dh= 0.0186 to 0.03986 p/e = 6.67 to 57.14,α=90°	U shaped ribs	$Nu_r=0.5429x\ Re^{0.7054} \times (p/e)^{-0.1592} \times (e/D_h)^{0.3619}$ $f=1.2134 \times Re^{-0.2076} \times (p/e)^{-0.4259} \times (e/D_h)^{0.3285}$
[1]	**[2]**	**[3]**	**[4]**	**[5]**
Aharwal et al (2009)	$e/D_h=0.037$ p/e=8	W/H= 5.83 d/W=0.16 to 0.5 g/e=0.5to2.0 Re=3000 to 18,000 P/e=4–10; e/D= 0.018–0.037	Inclined discrete ribs	$Nu=0.0102Re^{1.348}(e/D)^{0.51}[\{1-(0.25-\frac{d}{W})^2\{0.01(1-\frac{g}{e})^2\}\}]$ $f=0.5Re^{-0.0836}(e/D)^{0.72}$

		α=30–90°		
J.L. Bhagoria et al (2009)	e/D_h =0.038	Re=3000 to 15,000 e/Dh=0.0168–0.0338 p/e= 10 α=30° to 75°	Discrete W shaped ribs	$Nu_r=0.105 x Re^{0.873} \times (e/D_h)^{0.453} \times (\alpha/60)^{-0.081} \times \exp[-0.59 \times (\ln(\alpha/60))^2]$ $f=5.68 \times Re^{-0.40} \times (e/D_h)^{0.59} \times (\alpha/60)^{-0.081} \times \exp[-0.579 \times (\ln(\alpha/60))^2]$
Solanki et al (2009)	e/D_h =0.03 p/e=10 chamfer angle=18°	Re=3000–21,000 P/e=10 e/Dh=0.03 Φ=5°,12°, 15°, 18°, 22° and 30 60° V groove	Chamfered rib groove roughness	Nusselt number and friction factor increases 2.6-fold and 3.35-fold in the range of parameters investigated.
Pongjet Promvonge et al. (2010)	p/e=8 e/D_h =0.043	e/H=0.10, 0.20 and 0.30 Re=5000 to 25,000 AR=10 H=30mm	Multi V 60° baffles	$Nu=0.147Re^{0.763}Pr^{0.4}(1- e/H)^{-1.793}(PR + 1)^{-0.42}$ $f=0.48 Re^{-0.038}(1-e/H)^{-5.428}(PR + 1)^{-0.833}$
Karwa et al (2010)	e/D_h =0.07	L= 1–4 m,H =5–20 mm,W= 1 m,L/H 200 e/Dh 0.02–0.07 Re =1070–26350	V down discrete ribs	
[1]	**[2]**	**[3]**	**[4]**	**[5]**
Hans et al. (2010)	α=60°,W/w=6	Re=2000 to 20000 e/D=0.019– 0.043 P/e=6–12, α=30°–75° W/w=1–10.	Multi V ribs	$Nu=3.35 \times 10^{-5}Re^{0.92}(e/D)^{0.77}(W/w)^{0.43}(\alpha/90)^{-0.49} \times \exp^{[0.1177(\ln(W/w))2]}\exp^{[0.61(\ln(\alpha/90))2]}(p/e)^{8.54} \times \exp^{[-2.0407(\ln(p/e))2]}$ $f=4.47 \times 10^{-4}Re^{-0.3188}(e/D)^{0.73}(W/w)^{0.22}(\alpha/90)^{-0.39} \times \exp[-0.52(\ln(\alpha/90))^2](p/e)^{8.9}\exp^{[-2.133(\ln(p/e))2]}$
Giovanni Tanda (2011)	p/e=13.33	AR = 5 rib turbulators, inclination =45° Re=9000 to 35,500. e/D=0.09, p/e=6.66,10.0,	45° angled rib turbulators	

		13.33,and 20.0		
J.L. Bhagoria et al (2011)	e/D_h =0.03375	W/H=8.0, p/e=10, e/Dh= 0.03375 α=30°-75°. Re=2300–14,000.	W rib roughness at different angles	$R=\sqrt{(2/f)} + 2.5\ln(2e/D_h) + 3.75$ $e^{+}=\sqrt{f/2}\,Re(e/D_h)$
Brij Bhushan et al (2011)	S/e=31.25,d/D=0.294	e/D= 0.03 W/H=10.00 Re=4000–20,000 S/e=18.75–37.50 L/e=25.00–37.50 d/D= 0.147–0.367	Protrusion as artificial roughness	$Nu=2.1 \times 10^{-88} Re^{1.452}(S/e)^{12.94}(L/e)^{99.2}(d/D)^{-3.9} \times \exp[(-10.4\{\log(S/e)\}^{2}]\exp[(-77.2)\{\log(L/e)\}^{2}] \times \exp[(-7.83)\{\log(d/D)\}^{2}]$ $f=2.32Re^{-0.201}(S/e)^{-0.383}(L/e)^{-0.484}(d/D)^{0.133}$
[1]	**[2]**	**[3]**	**[4]**	**[5]**
Sukhmeet Singh et al (2011)	e/D_h =0.043 p/e=8	Re=3000-15000 P/e=4-12 α=30°-75° d/w= 0.2-0.8 g/e=0.5-2.0 e/Dh=0.015-0.043	Discrete V down ribs	$Nu = 2.36 \times 10^{-3} Re^{0.90}(p/e)^{3.50}(\alpha/60)^{-0.023}(d/W)^{-0.043}(g/e)^{-0.014} x(e/D_h)^{0.47} \exp(-0.84(\ln(p/e))^{2})\exp(-0.72(\ln(\alpha/60)^{2}) \times \exp(-0.05(\ln d/w)^{2})\exp(-0.15(\ln g/e)^{2})$ $f=4.13 \times 10^{-2} Re^{-0.126}(p/e)^{2.74}(\alpha/60)^{-0.034}(d/w)^{-0.058}(g/e)^{0.031} x (e/D_h)^{0.70} \exp(-0.685(\ln p/e)^{2})\exp(-0.93(\ln \alpha/60)^{2}) \times \exp(-0.058(\ln d/w)^{2})\exp(-0.21(\ln g/e)^{2})$
Atul Lanjewar et al (2011)	e/D_h=0.03375, α=60°	p/e =10 e=0.8 mm, 1.0 mm, 1.3 mm & 1.5 mm e/Dh=0.018, 0.0225, 0.02925 & 0.03375 W/H=8 Re=2300-14,000 α=30°, 45°, 60° & 75°	W shaped roughness	$Nu_r=0.0613(Re)^{0.9079}(e/D_h)^{0.4487}(\alpha/60)^{-0.1331} \times [\exp(-0.5307(\ln(\alpha/60))^{2})]$ $f=0.6182(Re)^{-0.2254}(e/D_h)^{0.4622}(\alpha/60)^{0.0817} x[\exp(-0.28(\ln(\alpha/60))^{2}$

CONCLUSION

It is found that most of the investigators developed their correlations experimentally for the solar air heaters ducts having different geometries as artificial roughness in order to enhance the heat transfer. But some researchers have attempted to numerically model the roughness effects. And it was observed that it is the most promising technique to improve the performance of solar air heaters.

Nomenclature

D_h	hydraulic diameter of duct,m
e	rib height, mm
g	groove position, mm
h	heat transfer coefficient, W/m^2K
H	height of air duct, mm
P	pitch, m
W	width of duct, m

Dimensionless Parameters

d/D	Relative print diameter
d/W	relative gap position
e^+	roughness reynolds number
e/D, e/D_h	relative roughness height
e/H	rib to channel height ratio
f	friction factor
G (e^+)	Heat transfer function
G_H	Heat Transfer Roughness Function
g/p	relative groove position
L/e	relative long way length of mesh
p/e	relative roughness pitch
Re	Reynolds number
R (e^+)	momentum transfer function
R	Roughness function
St	Stanton number
S/e	relative short way length of mesh
W/H	Aspect Ratio

Greek Symbols

α	angle of attack
Φ	wedge angle
α/90	arc angle

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