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A Review on Comparative Study of Thermal Performance of Artificially Roughened Solar Air Heaters

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Abstract—*Provision of artificial roughness is made to enhance the heat transfer rate in solar air heaters. Information regarding different configurations and geometries of roughness elements producing different quality and quantity of heat transfer is available in literature. This paper reviews the comparative rate of heat transfer quantitatively and qualitatively in artificially roughened solar air heaters of various configurations. Analytical and experimental result as well as worked out values of result with respect to heat transfer data in terms of Nusselt number, utilizing the equations or correlations developed by various researchers and investigators have been represented with respect to the roughness and flow parameters (p/e, e/D and Re), for the comparison of rate of heat transfer. The experimental values of average Nusselt number for multi V-roughness has been found to be the highest. However, the analytical values of average Nusselt number in the end of side roughened solar air heaters are the maximum.*

Keywords— *Relative roughness pitch (p/e), Relative roughness height (e/D), Flow Reynolds number (Re) and Average Nusselt number ($\overline{Nu_r}$).*

Nomenclature:

A_c collector area, m^2
B solar air heater duct height, m
D hydraulic diameter of solar air heater duct, m
e artificial roughness height, m
g heat transfer function
R roughness function

e^+ roughness Reynolds number $= e/D\sqrt{\left(\frac{f_r}{2}\right)}$ (Prasad et. al., 2014)

e^+ roughness Reynolds number $= e/D\sqrt{\left(\frac{\bar{f}}{2}\right)}$ (Prasad and Saini, 1991)

e^+_{opt} optimal roughness Reynolds number $= {}^{e}/_{D}\sqrt{\frac{f_r}{2}}\,Re = 23$ (Prasad et. al., 2015)

e/D relative roughness height
d/W relative gap position
F' plate efficiency factor
F_R heat removal factor
$\bar{f}$ average friction factor $= (f_s + f_r)/2$, in roughened collector
f_s friction factor in four sided smooth duct
f_r friction factor in four sided rough duct
$\overline{f_r}$ average friction factor in three sided rough duct(Prasad et. al., 2014)
e/H baffle blockage ratio
P/H baffle pitch spacing ratio
B/s boosted Stanton number ratio
g/e relative gap width
H solar air heater duct height, m (Referred cases)
h convective heat transfer coefficient, W/m^2 K
L collector length, m
${}^{l}/_{e}$ relative long way length of mesh
$\overline{Nu}$ average Nusselt number for top side roughened duct
$\overline{Nu_r}$ average Nusselt number for three sided roughened duct (Prasad et.al., 2014)
Nu_s Nusselt number for four sided smooth duct
p roughness pitch, m
p/e relative roughness pitch
Pr Prandlt number
Re Reynolds number
$\overline{St_r}$ average Stanton number for three sided roughened duct (Prasad et. al., 2014)
$\overline{St}$ Stanton number for top side roughened duct
St_r Stanton number in a four sided rough duct
${}^{s}/_{e}$ relative short way length of mesh
W solar air heater duct width, m
w width of single V-rib
${}^{W}/_{w}$ relative roughness width
${}^{W}/_{H}$ aspect ratio of duct
U_L overall heat transfer coefficient, W/m^2 K
ρ fluid density, Kg/m^3
α angle of attack (^{o})
φ chamfer/ wedge angle (^{o})
${}^{\varphi}/_{90}$ relative arc angle
η_{th} thermal efficiency

I. INTRODUCTION

Literature on use of artificial roughness of different geometries to enhance heat transfer in solar air heaters is widely available. (Prasad and Mullick, 1983) utilized small diameter wires on the top absorber plate to enhance the heat transfer in a solar air heater used for drying purposes. The effect of protrusions on 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 was analyzed (Prasad and Saini, 1988). The investigations were carried out (Prasad, 2013) for the relative roughness pitch of 10, 15 and 20 and relative roughness height of 0.020, 0.027 and 0.033 to see the effect of height and pitch of the roughness elements on heat transfer and friction. Where, the maximum value of Nusselt number and friction factor were reported to be 2.38 and 4.25 respectively, for relative roughness pitch of 10.

(Prasad and Saini, 1991) analyzed for the optimal thermo hydraulic performance of top side artificially roughened solar air heaters, covering a wide range of the values of relative roughness pitch (p/e), relative roughness height (e/D) and flow Reynolds number (Re), to arrive at the conclusion that the value of the parameter, roughness Reynolds number, $e^{+}\approx 24$ gives the optimal value of thermo hydraulic performance. The effect of transverse wire type roughness geometry fixed on underneath of an absorber plate was studied by Gupta et al., (1993) .The experiment conducted by using aspect ratio of 6.8-11.5, relative roughness height of 0.018-0.052 at a relative roughness pitch of 10 with a range of roughness Reynolds number in between 5 and 70 and Reynolds number in the range of 3000-18000. It was found that the Stanton number increased initially with an increase in Reynolds number up to 12000 and decreased by further increment of Reynolds number. (Gupta et al., 1997) examined 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 60^0 and relative roughness pitch of 10. It has been found that maximum enhancement in heat transfer coefficient in roughened duct at an angle of 60^0 and 70^0 respectively, that roughened surfaces with relative roughness height (e/D) of 0.033 corresponding to Reynolds number (Re) around 14000 gives the best thermo hydraulic performance in the range of parameters investigated.

(Saini and Saini, 1997) studied the heat transfer and friction characteristics for flow inside a solar air heater duct of large aspect ratio with roughness in the form of expanded metal mesh geometry. They explored that the average value of Nusselt number attains the maximum value at the relative longway mesh (L/e) of 46.87 and relative shortway length (S/e) of 25 at an angle of 61.9^0. The comparative study for the thermal performance of roughened absorber plate fixed with staged discrete V-apex has been done by (Muluwork et al., 1998) and (Muluwork, 2000). Boosted Stanton number ratio was found to be of order of 1.32-2.47. Due to the increment of relative roughness length ratio in the range of 3-7, there was an increment in Stanton number. (Karwa et al., 1999) developed heat transfer coefficient and friction factor correlations in rib-roughened solar air heater duct for transitional flow. (Verma and Prasad, 2000) experimentally verified for the optimal thermohydraulic performance in artificially roughened solar air heaters and arrived at the conclusion that the value of flow Reynolds number, $e^{+}\approx 24$, gives the optimal thermo hydraulic performance.

(Bhagoria et al., 2002) used 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 analyzed that the presence of wedge shape ribs yield maximum enhancement in Nusselt number, about 2.4 times as compared to smooth duct. Nusselt number increases and attains maximum value at a wedge angle of about 10^0 and then sharply decreases with increasing wedge angle beyond 10^0. (Saini and Saini, 2008) explored 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 for the range of Reynolds number (Re), from 2000 and 17000, relative roughness height (e/D), 0.0213 to 0.0422 and relative arc angle (α/90), 0.3333 to 0.6666. 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. (Varun et al., 2008) experimentally 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. Table I represents various roughness geometries of transverse or inclined, wedge shaped rib, rib-grooved, dimple-shape, arc-shaped wire, compound turbulation, metal grit ribs and w-shape deal for the analysis and investigation towards heat transfer enhancement in solar air heaters.

(Momin et al., 2002) experimented 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 with an increase of Reynolds number. It was found that for relative roughness height of 0.034 and for angle of attack of 60^0, the V-shaped ribs enhance the values of Nusselt number by 1.14 and 2.3 times respectively over inclined ribs and smooth plate case at Reynolds number of 17034. The effect of 90^0 broken transverse ribs on heat and fluid flow characteristics was investigated by (Sahu and Bhagoria, 2005), with a roughness height of 1.5 mm, aspect ratio of 8, pitch in the range of 10-30 mm and Reynolds number in the range of 3000-12000. The rate of heat transfer was enhanced by 1.25-1.4 times over smooth duct. Maximum thermal efficiency found during experimentation was 83.5 %. (Varun et al., 2007) reviewed on roughness geometries used in solar air heaters. The thermo hydraulic performance of a solar air heater has been investigated by (Aharwal et al., 2008) for inclined continuous rib with a gap with parameters W/H=5.84, p/e=10, e/D_h= 0.0377, $\alpha=60^0$, g/e=0.5-2, d/W=0.1667-0.667, Re=3000-118000. It was found that for the optimum values of parameters (g/e=0.5 and d/W=0.25), the increase in Nusselt

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number and friction factor equal to 2.59 and 2.87 respectively, was maximum. An investigation for the thermo hydraulic performance of solar air heaters with inverted U-shaped ribs on the absorber plate has been carried out by (Bopche and Tandale, 2009). The roughness element was found so efficient even at lower Reynolds number (Re<5000) and was further concluded that the turbulence was created only in viscous sub-layer resulting in higher thermo hydraulic performance over smooth solar air heaters. (Hans et. al., 2009) have reviewed on performance of artificially roughened solar air heaters. (Hans et al., 2010) developed 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. It has been observed that Nusselt number attain maximum value at 60^0angle of attack.

(Sharma et al., 2010) analyzed that small diameter protrusion wires are better in the flow Reynolds number limited to 10000. (Sethi et al., 2010) predicted for effective efficiency of solar air heater using discrete type of ribs. An experimental investigation has been done by (Promvonge, 2010) to evaluate turbulent forced convection heat transfer and friction losses behavior for air flow through a channel fitted with a multiple 60^0 V-baffle tabulators. Experiment was investigated by using a channel of aspect ratio of 10 and height of 30 mm with three different baffle blockage ratios of 0.10, 0.20 and 0.30 and three baffle pitch spacing ratios of 1, 2, and 3, whereas the transverse pitch of the V-baffle is kept constant. (Kumar et al., 2011) have investigated for the enhancement of heat transfer coefficient of a solar air heater having roughened air duct with a provision of artificial roughness in the form of 60^0 inclined discrete ribs. For the relative roughness pitch of 12, relative gap position of 0.35 and relative roughness height of 0.0498, the maximum heat transfer enhancement takes place. Statistical correlations for friction factor and Nusselt number have been derived as a function of gap position, rib depth, pitch and Reynolds number. The correlations have been established to predict the values of friction factor and Nusselt number with an average absolute standard deviation of 3.4% and 3.8% respectively. An experimental investigation for heat transfer and friction factor was carried out by (Lanjewar et al., 2011), having the characteristics of a rectangular duct roughened W-shaped ribs, that is arranged at an inclination with respect to the flow direction. The duct used, had a width to height ratio of 8, relative roughness pitch of 10, relative roughness height of 0.03375 and angle of attack of flow of 30^0-75^0. (Patil et al., 2012) have also reviewed the types of roughness geometries and investigation techniques used in artificially roughened solar air heaters. If the value of heat transfer coefficient between the absorber plate and flowing air is low, the value of thermal performance is also low. So, the provision of artificial roughness in the form of thin wires of varying diameters at different relative roughness pitch and relative roughness height on the absorber plate has resulted in increase of the value of heat transfer coefficient leading to increase the value of thermal efficiency of such collectors compared to the smooth ones.

(Chamoli et al., 2012) reviewed regarding approach for further research on use of turbulence promoters for heat transfer enhancement in solar air heaters, as also the performance of double pass solar air heaters. (Bhusan et al., 2012) developed the thermal and thermo hydraulic performance of protruded solar air heater. (Karwa et al., 2013) developed the correlations for Nusselt number and friction factor for multi V-shaped roughness for solar air heaters. (Karwa et al., 2013) found 12.5-20% enhancement in thermal efficiency for 60^0 V-down discrete rib roughnesses in solar air heaters. (Saurabh et al., 2013) reviewed for heat transfer and thermal efficiency in solar air heater having artificial roughness. (Prasad, 2013) represented the experimental results on heat transfer and thereby thermal performance of artificially roughened solar air heaters for fully developed turbulent flow. Such solar air heaters have been found to give considerably high value of collector heat removal factor (F_R), collector efficiency factor (F') and thermal efficiency (η_{th}) as compared to the corresponding values of those of smooth collectors. In the range of the operating parameters investigated, the ratio of the respective values of the parametersF_R, F' and $\eta_{th}$for the roughened collectors to the smooth collectors have been found to be 1.786, 1.806 and 1.842 respectively.

(Prasad et al., 2014) analyzed with respect to fluid flow and heat transfer in a novel solar air heater having artificial roughness on three sides (the two side walls and the top side) of the rectangular solar air heater duct, with three sides glass covers. Equations for friction factor and heat transfer parameter have been developed. The analytical values of friction factor and heat transfer parameter have been found to be 2 to 40% more and 20 to 75% more than those of the respective values of (Prasad and Saini, 1988) for the same range of the values of operating parameters p/e, e/D and Re and fixed values of W and B. Prasad et al., (2015) has analyzed the thermo hydraulic optimization of three sides (the two side walls and the top side) artificially roughened solar air heater to get the maximum heat transfer for the minimum pumping power (friction factor) and found that the optimal thermo hydraulic performance condition corresponds to the optimal value of roughness Reynolds number, , $e^+_{opt} = {}^e/_D \sqrt{\frac{f_r}{2}}\, Re = 23$, for a specific set of values of roughness and flow parameters *p/e*, *e/D* and *Re.*

Table I represents roughness geometries, fluid flow directions, roughness and flow parameters analyzed, investigated and the equations developed for the heat transfer and friction parameters by various authors to have a more comparative view

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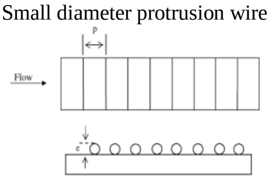
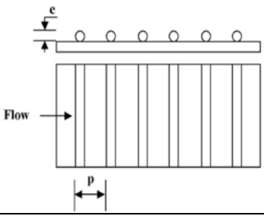
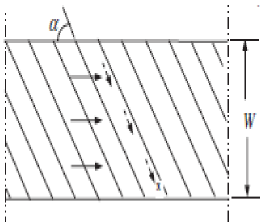
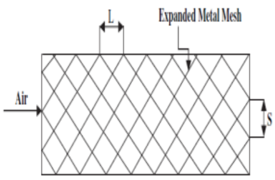
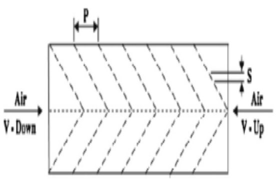
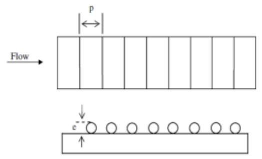
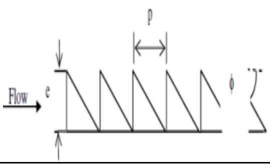
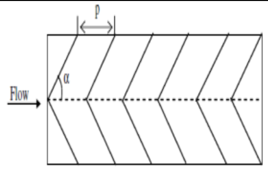
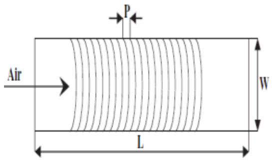
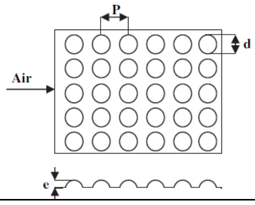
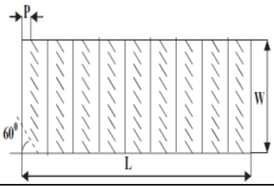
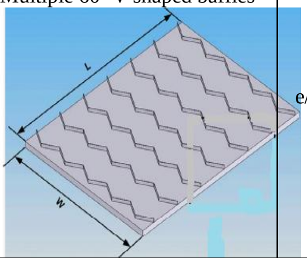
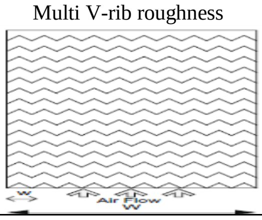
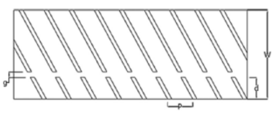
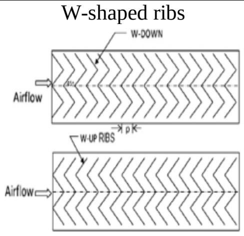
TABLE I.

S. No.	Reference	Roughness Geometry	Parameter Investigated	Associated Equations / Correlations
1	Prasad and Saini (1988)	Small diameter protrusion wire 	e/D=0.020-0.033 p/e=10-20 Re=5000-50000	$\overline{St} = \dfrac{\bar{f}/2}{1+\sqrt{\dfrac{\bar{f}}{2}}\left[4.5(e^{+})^{0.28}Pr^{0.57}-0.95\left(p/e\right)^{0.53}\right]}$ $\bar{f} = \dfrac{(W+2B)f_s + W\left[\dfrac{2}{\left\{0.95\left(p/e\right)^{0.53}+2.5\ln\left(D/2e\right)-3.75\right\}^{2}}\right]}{2(W+B)}$ $\overline{Nu} = \bar{S}_t Re P_r$
2	Gupta et al., (1993)	Transverse ribs 	e/D=0.018-0.052 Re=3000-18000 W/H=6.8-11.5	$Nu = 0.000824\left(e/D\right)^{-0.718}\left(W/H\right)^{0.284}Re^{1.062}$ For $e^{+} \leq 35$ $Nu = 0.00307\left(e/D\right)^{-0.469}\left(W/H\right)^{0.245}Re^{0.812}$ For $e^{+} \geq 35$ $f = 0.06412\left(e/D\right)^{0.019}\left(W/H\right)^{0.237}Re^{-0.185}$
3	Gupta et al., (1997)	Inclined ribs  (i) Inclined ribs	e/D=0.020-0.053 p/e=7.5 & 10 α =30-90 Re=5000-30000	$\overline{Nu} = 0.0024\left(e/D\right)^{0.001}\left(W/H\right)^{-0.06}Re^{1.084} \times \exp\left[-0.04\left(1-\alpha/60\right)^{2}\right]$ For $e^{+} < 35$ $\overline{Nu} = 0.0071\left(e/D\right)^{-0.024}\left(W/H\right)^{-0.028}Re^{0.088} \times \exp\left[-0.475\left(1-\alpha/60\right)^{2}\right]$ For $e^{+} > 35$ $\bar{f} = 0.06412\left(e/D\right)^{0.019}\left(W/H\right)^{0.237}(Re)^{-0.185}$
4	Saini and Saini (1997)	Wire mesh roughness 	e/D=0.012-0.039 S/e=15.62-46.87 L/e=25.00-71.87 Re=1900-13000	$\overline{Nu} = 4.0 \times 10^{-4}Re^{1.22}\left(e/D\right)^{0.625}\left(S/10e\right)^{2.22} \times \exp\left[-1.25\{\ln(S/10e)\}^{2}\right]\left(l/10e\right)^{2.66} \times \exp\left[-0.824\{\ln(l/10e)\}^{2}\right]$ $\bar{f} = 0.815(Re)^{0.261}\left(S/10e\right)^{0.19}\left(10e/d\right)^{0.591}$
5	Mulluwork et al., (1998)	V shaped staggered discrete wire ribs 	e/D=0.02 B/s=3-9 α =60^{0} Re=2000-15500	$Nu = 0.00534\left(B/s\right)^{1.2496}Re^{1.2991}$ $f = 0.7117\left(B/s\right)^{0.0626}Re^{-2.991}$
6	Verma and Prasad (2000)	Transverse Protrusion wire 	e/D=0.01-0.03 p/e=10-40 Re=5000-20000 e^{+}=8-42 P_r=0.7	$Nu = 0.08596\left(p/e\right)^{-0.054}\left(e/D\right)^{0.072}Re^{0.723}$ For, $e^{+} \leq 24$ $Nu = 0.0245\left(p/e\right)^{-0.016}\left(e/D\right)^{0.021}Re^{0.802}$ For, $e^{+} \geq 24$ $f = 0.0245\left(p/e\right)^{-0.0206}\left(e/D\right)^{0.021}Re^{-1.25}$
7	Bhagoria et al., (2002)	Wedge shape ribs 	e/D=0.015-0.033 $\dfrac{p}{e} = 60.17\Phi^{-1.0264}$ Φ =8,10,12,15 Re=3000-8000	$\overline{Nu} = 1.89 \times 10^{-4}Re^{1.21}\left(e/D\right)^{0.426}\left(p/e\right)^{2.94} \times exp\left[-0.71\{\ln(p/e)\}^{2}\right]\left(\varphi/10\right)^{-0.018} \times exp\left[-1.5\{\ln(\varphi/10)\}^{2}\right]$ $\bar{f} = 12.44\left(e/D\right)^{0.99}\left(\varphi/10\right)^{0.49}Re^{-0.18}\left(p/e\right)^{-0.52}$
8	Momin et al., (2002)	V-shape ribs	e/D=0.02-0.034 p/e=10	$\overline{Nu} = 0.067Re^{0.888}\left(e/D\right)^{0.424}\left(\alpha/60\right)^{-0.077} \times exp\left[-0.782\{\ln(\alpha/60)\}^{2}\right]$

			α=30-90 Re=2000-15500	$\bar{f} = 6.266\left({}^{e}/_{D}\right)^{0.565}\left({}^{\alpha}/_{60}\right)^{-0.093}Re^{-0.425}\exp[-0.719\ln\left({}^{\alpha}/_{60}\right)^{2}]$
9	Saini and Saini (2008)	Arc shape roughness 	e/D=0.0213-0.0422 p/e=10 Re=2000-17000 α/90=0.333-0.666 W/H=12	$\overline{Nu} = 0.001047Re^{1.3186}\left({}^{e}/_{D}\right)^{0.3772}\left({}^{\alpha}/_{90}\right)^{-0.1198}$ $\bar{f} = 0.14408\left({}^{e}/_{D}\right)^{0.1765}\left({}^{\alpha}/_{90}\right)^{0.1185}Re^{-0.17103}$
10	Saini et al., (2008)	Dimple shape roughness 	e/D=0.0189-0.038 p/e=8-12 Re=2000-12000	$\overline{Nu} = 5.2\times10^{-4}Re^{1.27}\left({}^{p}/_{e}\right)^{3.15}\times\left[\exp(-2.21)\{\log\left({}^{p}/_{e}\right)\}^{2}\right]\left({}^{e}/_{D}\right)^{0.033}\times\left[\exp(-1.30)\{\log\left({}^{e}/_{D}\right)\}^{2}\right]$ $\bar{f} = 0.0642(Re)^{-0.423}\left({}^{e}/_{D}\right)^{-0.0214}\left({}^{p}/_{e}\right)^{-0.465}\exp\left[0.054\{\ln {}^{p}/_{e}\}^{2}\right]$ $\exp\left[-0.840\{\ln {}^{e}/_{D}\}^{2}\right]$
11	Varun et al., (2008)	Combined inclined and transverse ribs 	e/D=0.030 p/e=8 W/H=10 Re=2000-14000	$\bar{f} = 1.0858(Re)^{-0.3685}\left({}^{p}/_{e}\right)^{0.0114}$ $\overline{Nu} = 0.0006(Re)^{1.213}\left({}^{p}/_{e}\right)^{0.0104}$
12	Promvonge, (2010)	Multiple 60^{0} V shaped baffles 	e/H=0.1, 0.2 and 0.3 P/H=1, 2 and 3 Re=5000-25000	$Nu = 0.147Re^{0.762}\left({}^{P}/_{H}\right)^{0.4}(1-e/H)^{-1.793}\left(1+{}^{P}/_{H}\right)^{-0.42}$ $f = 0.48Re^{0.038}(1-e/H)^{-5.428}(1+P/H)^{-0.833}$
13	Hans et al., (2010)	Multi V-rib roughness 	e/D=0.019-0.043 p/e=6-12 Re=2000-20000 α=30-75 W/w=1-10	$\overline{Nu} = 3.35\times10^{-5}Re^{0.92}\left({}^{e}/_{D}\right)^{0.77}\left({}^{W}/_{w}\right)^{0.43}\left({}^{\alpha}/_{90}\right)^{-0.49}\times\exp\left[-0.1177\{\ln\left({}^{W}/_{w}\right)\}^{2}\right]\times\exp\left[-0.61\{\ln\left({}^{\alpha}/_{90}\right)\}^{2}\right]\left({}^{p}/_{e}\right)^{8.54}\times\exp\left[-2.0407\{\ln\left({}^{p}/_{e}\right)\}^{2}\right]$ $\bar{f} = 4.47\times10^{-4}Re^{-0.3188}\left({}^{e}/_{D}\right)^{0.73}\left({}^{W}/_{w}\right)^{0.22}\left({}^{\alpha}/_{90}\right)^{-0.39}\times\exp\left[-0.52\left(\ln\left(\frac{\alpha}{90}\right)\right)^{2}\right]\left({}^{p}/_{e}\right)^{8.9}\exp\left[-2.133\left(\ln\left({}^{p}/_{e}\right)\right)^{2}\right]$
14	Kumar et al., (2011)	60^{0} angle inclined continuous discrete rib 	Re=4105.2-20526.2 e/D=0.0249, 0.0374 and 0.0498 p/e=8, 12 and 16 d/W=0.15, 0.25 and 0.35 g/e=1	$Nu = 3\times10^{-5}Re^{0.947}\left({}^{e}/_{D}\right)^{0.29}\left({}^{p}/_{e}\right)^{5.885}\left({}^{d}/_{W}\right)^{0.115}\times\exp\left[-1.237\left\{\ln\left({}^{p}/_{e}\right)^{2}\right\}\right]$ $f = 0.0014Re^{-0.22}\left({}^{e}/_{D}\right)^{0.804}\left({}^{p}/_{e}\right)^{4.518}\left({}^{d}/_{W}\right)^{0.097}\times\exp\left[-0.944\left\{\ln\left({}^{p}/_{e}\right)^{2}\right\}\right]$
15	Lanjewar et al., (2011)	W-shaped ribs 	p/e=10 e=1.5 e/D=0.03375 W/H=8 Re=2300-14000 α =30^{0},45^{0},60^{0}-75^{0} e^{+}=8-44	$R = \sqrt{\left({}^{2}/_{f}\right)} + 2.5\ln\left({}^{2e}/_{D}\right) + 3.75$ $e^{+} = \sqrt{\left({}^{f}/_{2}\right)}\,Re\left({}^{e}/_{D}\right)$ $g = \left[\left({}^{f}/_{2St}\right) - 1\right]\sqrt{\left({}^{2}/_{f}\right)} + R$

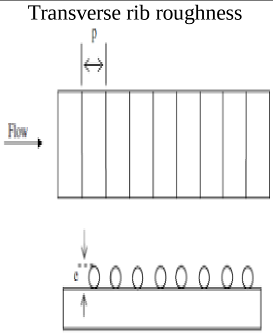
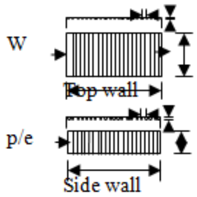
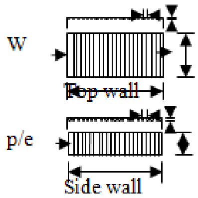
16	Prasad (2013)	Transverse rib roughness 	e/D=0.0092-0.0279 p/e=10-40 Re=2959-12631	$\overline{Nu} = \left(\bar{f}/_2\right) RePr/\left[1 + \sqrt{\left(\bar{f}/_2\right)}\left\{4.5(e^+)^{0.28}Pr^{0.57} - 0.95\left(^p/_e\right)^{0.53}\right\}\right]$ $\bar{f} = \frac{2}{[0.95\left(^p/_e\right)^{0.53} + 2.5\,ln\left(^D/_{2e}\right) - 3.75]^2}$
17	Prasad et al., (2014)	Small diameter protrusion wire in three sides e/D 	e/D=0.020-0.033 p/e=10-20 Re=3000-12000	$\overline{Nu_r} = \frac{\bar{f}/2}{1 + \left(\sqrt{\bar{f}}/2\right)\left[4.5(e^+)^{0.28}Pr^{0.57} - 0.95\left(^p/_e\right)^{0.53}\right]} Re\, Pr$ $\overline{f_r} = \frac{(W+2B)\left[\frac{2}{\left\{0.95\left(^p/_e\right)^{0.53} + 2.5\ln\left(^D/_{2e}\right) - 3.75\right\}^2}\right] + Wf_s}{2(W+B)}$
18	Prasad et al., (2015)	Small diameter protrusion wire in three sides e/D 	e/D=0.01126-0.0279 p/e=10-40 Re=3000-20000	$\bar{S}_{t_r} = \frac{\bar{f}_r/2}{1 + \sqrt{\left(\frac{\bar{f}_r}{2}\right)}\left[4.5(e^+)^{0.28}p_r^{0.57} - 0.95\left(^p/_e\right)^{0.53}\right]}$ $e^+ = \frac{^e/_D\sqrt{\bar{f}_r}}{2}Re$ For optimal condition, $e^+_{opt} = \frac{^e/_D\sqrt{\bar{f}_r}}{2}Re = 23$

TABLE II. Comparison of enhancement in heat transfer data ($\overline{Nu}$ & $\bar{f}, \bar{f}_r$)

References	Reynolds Number	Average Nusselt Number		Average Friction Factor	
		Values	Remarks	Values	Remarks
Prasad et al (2014)	$(3\text{-}14)X10^3$	40-140	Analytical	0.047-0.035	Analytical
Bhagoria (2002)	$(3\text{-}14)X10^3$	18-82	Analytical	0.03-0.0264	Analytical
Prasad & Saini (1998)	$(3\text{-}14)X10^3$	20-65	Analytical	0.03-0.025	Analytical
Gupta et al (1993)	$(3\text{-}14)X10^3$	10-30	Analytical	0.037-0.019	Analytical
Saini & Saini (1997)	$(3\text{-}14)X10^3$	20-120	Analytical	0.039-0.027	Analytical
Hans et al (2010)	$(3\text{-}14)X10^3$	50-185	Experimental	0.049-0.027	Experimental
Momin et al (2002)	$(3\text{-}14)X10^3$	20-70	Experimental	0.013-0.007	Experimental
Prasad (2013)	$(3\text{-}14)X10^3$	20-65	Experimental	-------------	-----------
Saini & Saini (2008)	$(3\text{-}14)X10^3$	10-65	Experimental	0.018-0.014	Experimental
Saini et al (2008)	$(3\text{-}14)X10^3$	8-50	Experimental	0.07-0.045	Experimental
Varun et al (2008)	$(3\text{-}14)X10^3$	9-32	Experimental	0.03-0.019	Experimental

TABLE III. Value of e^+_{opt} for different roughness geometries for fully developed turbulent flow

S. No.	References	Roughness Geometry Type	p/e	e/D	Re	e^+_{opt}
1.	Sheriff and Gumley (1966)	Annulus with wires	10	———	$10^4\text{-}2\times10^3$	35
2.	Webb and Eckert (1972)	Rectangular	10-40	0.01-0.04	$6\text{-}100\times10^3$	20
3.	Lewis (1975a, 1975b)	Circular tubes with ribs	2-60	0.02-0.1	———	20
4.	Prasad and saini (1991)	Rectangular duct with thin wires on one side	10-40	0.020-0.033	$3\text{-}20\times10^3$	24
5.	Prasad et al., (2015)	Rectangular duct with thin wires on three sides	10-40	0.01126-0.0279	$3\text{–}20\times10^3$	23

Table III shows the different values of the optimal roughness Reynolds number, e^+_{opt}, found in literature for different roughness geometries and range of values of roughness and flow parameters for fully developed turbulent flow.

II. RESULTS AND DISCUSSION

Studies of artificially roughened solar air heater duct predominantly concerned with the effect of shape and arrangement of roughness elements on heat transfer requirement comparative assessment. It is therefore, reasonable to compare the performance of some distinct roughness geometries separately with respect to heat transfer.

Figure 1 & 2 have been drawn as such, with the help of worked out values of the heat transfer data, $\overline{Nu}$ and $\bar{f}, \bar{f}_r$ respectively, by substituting the values of roughness and flow parameters in the equations developed by the respective

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authors and the variation in average Nusselt number and average friction factor as a function of Reynolds number for different roughness geometries, analytically developed by different investigators. Considerable increase in Nusselt number and friction factor can be seen in all the cases. Figure 1 reveals that the value of Nusselt number is highest in case of transverse rib in three sided (the two side walls and the top side) artificially roughened solar air heater duct and lowest in case of inclined ribs for all the values of Reynolds number. Transverse rib roughness in three sides shows discrete performance among all the roughness geometries and provides almost double increment in heat transfer rates in comparison to other roughness geometries.

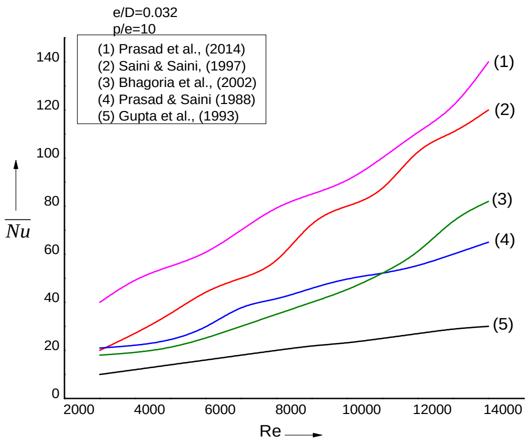

Fig.1. Comparision of average values of Nusselt number

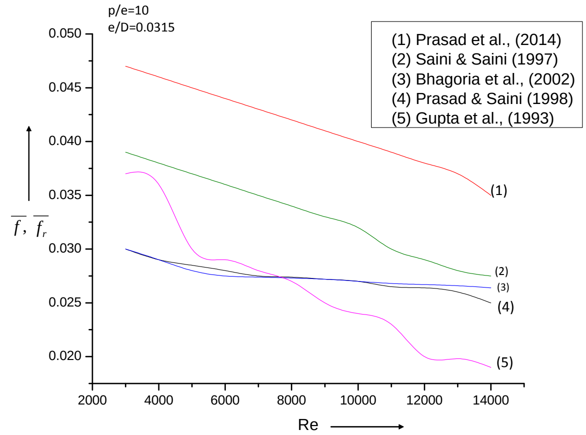

Fig.2 Comparison of average values of friction factor

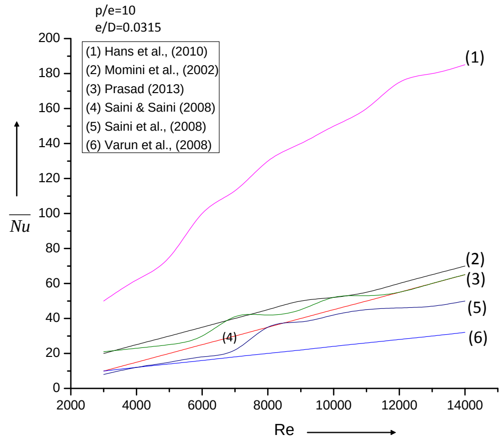

Fig.3 Comparison of experimental values of average Nusselt number

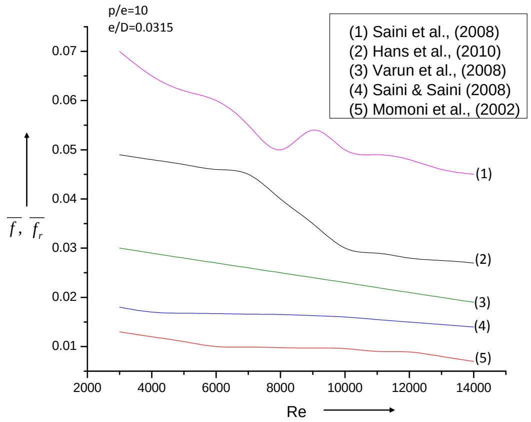

Fig.4 Comparison of experimental values of average friction factor

Figure 3 & 4 have been drawn for the respective experimented values of heat transfer data, $\overline{Nu}$ and $\bar{f}, \bar{f_r}$ respectively and the variation of average Nusselt number and average friction factor as a function of Reynolds number for roughness geometries tested experimentally by different authors. It can be seen that the value of Nusselt number is highest in case of multi V shape roughness and lowest in case of combination of transverse and inclined ribs for all the values of Reynolds number. Multi V shape roughness shows the maximum value of Nusselt number as compared to other roughness geometries. It can be seen that in case of V shape rib, multi V shape rib, inclined rib and transverse rib roughness, a step rise in Nusselt 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 rate to achieve higher rate of heat transfer. But the system with low and moderate flow rates, multi V shape roughness is the best option as far as heat transfer enhancement is concerned. Therefore the geometries like V shape rib and multi V shape rib with gap at suitable location could exhibit considerable enhancement in heat transfer rate 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.

III. CONCLUSION

- Use of artificial 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.
- Multi V rib roughness is one of the most suitable form of roughness tested experimentally showed distinct performance in case of solar air heaters.
- Use of artificial roughness in three sides of the absorber plate(top side and two side walls) is one of the best form of roughness geometry developed analytically shows the higher rate of heat transfer in case of 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 increase time duration.

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