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COMPUTATIONAL INVESTIGATION OF THERMAL PERFORMANCE OF SOLAR AIR HEATER HAVING ROUGHNESS ELEMENTS AS TRANSVERSE WIRE ON THE TWO SIDE ABSORBER PLATE

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ABSTRACT
An Analytical investigation has been carried out to study the heat transfer and friction characteristics by using a transverse wire on the absorber plate of a solar air heater. The Analytical investigation consist the Reynolds number (Re) ranges from 3000 to 14 000, relative roughness pitch (p/e) 10–30 and relative roughness height (e/D) 0.0135-0.0225. The effect of these parameters on the heat transfer coefficient and friction factor has been discussed in the present paper and correlations for Nusselt number and friction factor has been developed within above limits. A procedure to compute the thermal efficiency based on heat transfer processes in the system is also given and the effect of these parameters on thermal efficiency has been discussed .transverse wire have been used to enhance heat transfer coefficient. A Matlab code has been encoded for the analysis of thermal efficiency through iteration method. Results shows that the accuracy of thermal efficiency are 78% for this roughened solar air heater. By providing artificial roughness it has been concluded that there is an improvement of heat transfer coefficients which results increase in pumping power, pressure drop and high value of thermal performance.

KEYWORDS: Solar air heater; Artificial Roughness; Performance Analysis; Two side; thermal efficienc .

I. INTRODUCTION
Solar collector is a basic device which captures the solar radiation and convers into useful heat energy. Solar air heat collector due to its simple design, construction and low cost, it is widely used to collect solar energy. It has applications from seasoning of timber to drying of seeds tor preservation.
Solar collector consists of an absorber plate, wooden box, back plate, glass cover and insulator.

Major drawback of solar air heater is low efficiency due to low heat transfer coefficient which leads to poor performance.Convective heat transfer is low between air and absorber plate due to formation of laminar sub-layer. Heat transfer coefficient can be enhanced by breaking the sub-layer by creating turbulence in air flow. Turbulence can be created by introducing artificial roughness on collector surface however it increases the friction losses and therefore required more power for pumping air through collector. To keep frictional losses as low as possible it is required to create turbulence only near the surface without disturbing already existing turbulent flow.

The concept of surface roughness in solar air heater was first introduced by Prasad and Mullic Saini (1988) they studied the effect of transverse rib roughness in solar collector on heat transfer it was based on the approach considered by Han 1984, the maximum nusselt number was reported to be 2.38 times over the conventional duct.

The correlation used to predict the average nusselt number is given as:
Nu=f21+(f2)[4.5(e+)0.28pr0.57-0.95(pe)0.53]RePr

(1)

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Top side artificially roughened solar heater was studied by Prasad and saini 1991 for optimizing Thermo-Hydraulic performance, they uses various values of relative roughness pitch (pe) ,relative roughness height (eD) and Reynoldsnumber (Re) they arrived at the conclusion that the value of roughness Reynolds number , e+≈24 gives the optimal value of Thermo-Hydraulic performance (i.e. minimum pumping power and maximum heat transfer).

Gupta et al 1993 investigated the performance of solar air heater using transverse wire rib on the top surface. They kept relative roughness pitch value constant (pe=10) and studied it for different aspect ratio and relative roughness height, flow Reynolds number used was 3000 - 18000. Based on their study they concluded following correlations:

For $e+ < 35$

$$Nur = 0.000824(eD)^{-0.178}(WH)^{0.288}(Re)^{1.62} \quad (2)$$

For $e+ \geq 35$

$$Nur = 0.00307(eD)^{0.469}(WH)^{0.245}(Re)^{0.812} \quad (3)$$

Karwa et al. developed correlations for friction factor and heat transfer coefficient in transition flow for top roughened solar collector duct. Verma and Prasad 2000 developed correlations for heat transfer coefficient for top side artificially roughened solar heater duct in fully developed turbulent flow which is given as:

For $e+ \leq 24$

$$Nur = 0.08596(eD)^{0.072}(pe)^{-0.054}(Re)^{0.728} \quad (4)$$

For $e+ > 24$

$$Nur = 0.02954(eD)^{0.021}(pe)^{-0.016}(Re)^{0.802} \quad (5)$$

Various investigators with their roughness geometry and dimensional parameters are tabulated below:

Investigator	Roughness geometry	parameter
Prasad and Mallic	Transverse wire rib	eD=0.0190 ; Pe=12.7
Gupta	Inclined wire rib	eD=0.0230 ; Pe=10
Momin	v- shaped rin	eD=0.0230 ; Pe=10
Karwa	Chamfered rib	eD=0.0441 ; Pe=4.85
Jaurker	Rib – grooved	eD=0.0363 ; Pe=6
Bhagoria	Transverse wedge	eD=0.0330 ; Pe=7.57
Saini and Saini	Arc shaped rib	eD=0.0422 ; Pe=10
Karmare and Tikekar	Metal grit rib	eD=0.0440 ; Pe=17.5

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Nomenclature

Ac 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
e^+ roughness Reynolds number $= {}^{e}/_{D}\sqrt{\frac{f_r}{2}}\,ReSt_r$ (referred case)
e^+ roughness Reynolds number $= {}^{e}/_{D}\sqrt{\frac{f_r}{2}}\,Re$ (present case)
$\frac{e}{D}$ relative roughness height
F plate efficiency factor
$\bar{f}$ average friction factor $= (f_s + f_r)/2$ in Rough-ended collector (referred cases)
f_s friction factor in four sided smooth duct
f_r friction factor in four sided rough duct
$\bar{f_r}$ average friction factor in two sided rough Duct (present case)
$\bar{\tau}$ average wall shear stress in two sided rough and two sided smooth duct (N/m^2)
G mass velocity ($Kg/s.m^2$)
H solar air heater duct height, m (referred cases)
h convective heat transfer coefficient, W/m^2-K
m mass flow rate (Kg/s)
L collector length, (m)
q_u rate of heat transfer to air (W)
t_a ambient air temperature
t_i inlet air temperature
t_a Average absorber plate temperature.
F_O Heat removal factor referred to the outlet temp.
F' Collector-efficiency factor.
K thermal conductivity of air (W/m-K)
Nu_r Nusselt number for top side roughened duct (referred cases)
μ dynamic viscosity (kg/m-s)
ϵ_p emittance of absorber plate
Re Reynolds number
N number of glass cover
$\overline{Nu_r}$ average nusselt number for two sided roughened Duct (present case)
p/e relative roughness pitch
pr prandtal number
$\overline{St_r}$ average Stanton number for two sided roughened duct (present case)
$\bar{St}$ Stanton number for top side roughened duct (referred case)
St_r stsnton number in four sided rough duct
W solar air heater duct width , m
v_s fluid flow velocity in four sided smooth duct m/s
v_r fluid flow velocity in four sided rough duct m/s
$\bar{v_r}$ fluid flow velocity in two sided rough and two sided smooth duct m/s
τ wall shear stress (N/m^2)
τ_r wall shear stress on four sided roughened duct (N/m^2)
τ_s wall shear stress on four sided smooth duct N/m^2
ρ fluid density (Kg/m^3)
h_w wind convection co-efficient (W/m^2-K)
U_L loss co-efficient (W/m^2-K)
U_t top loss co-efficient (W/m^2-K)
t_o output air temperature
Nu_s nusselt number for four sided smooth duct
I insolation (W/m^2)
K_i thermal conductivity of insulating material(W/m-K)
p roughness pitch, m
$\tau\alpha$ transmittance-absorptance produc
σ Stefan-Boltzmann constant
ϵ_g emittance of glass cover
η thermal efficiency

subscripts

s four sided smooth duct
r four sided rough duct
2s two sided smooth
2r two sided rough

Sharp edge roughness element increases heat transfer coefficient more than smooth or roundness shaped roughness but it increases friction losses even more than the roundness shaped roughness. The net effect of

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sharpness of roughening element is investigated by Sparrow and Hossfeld, 1984 and it was reported that round shaped roughness geometry is more suitable than sharp edge roughness.

Studies of sharma and varun, 2010, while comparing the performance of different types of geometry of roughness element in solar air heater duct, shows that small diameter protrusion wire are better for flow Reynolds number up to 10000.

The value of collector efficiency factor, F' given by equation (Bliss, 1959), is given as:

F'=hh+UL (6)

F'Can be enhanced by increasing the value of heat transfer coefficient, h, between the absorber plate and flowing air over it . the increase in the value of h further decrease the value of heat loss coefficient which also increases the value of F' .It is general practice to provide roughness only on one surface (top surface) of solar air heater duct so, only the top surface forms the absorber plate and the side plates are insulated which are not the part of absorber. Glass cover is also provided on the top side to receive solar radiation. The side wall of the solar air heater duct may form the part of absorber plate having artificial roughness and side glass to receive solar radiation. Considering that the rectangular solar air heater duct has two sides absorber plate and two sides (bottom plate and one side wall) insulated, the present work has been focused at to analyze for fluid flow and heat transfer for fully developed turbulent flow in solar duct roughened artificially using small diameter wires on two sides (top side and one side wall). The analytical values of the roughness and the heat transfer parameter has been found out with the reference to the results of Prasad and saini 1998; Prasad , 2013;B.N. Prasad et al. 2014 available for one side and three side roughened solar air heater , to see the effect of roughness on heat transfer enhancement in the present solar air heater.

II. SOLAR AIR HEATER DUCT DESIGN

Fig. 1a represents the two side roughened solar air heater which is considered for the analysis. It has two side roughened wall and two side smooth wall, rough wall is shown in dark area. Absorber plate constitute of these two roughened wall. Its dimension is W ×B

Fig. 1b represents four side smooth air heater duct of dimensions identical as roughened duct. It is used for comparison purpose Glass cover plates are used on the top and one side of the absorber plate in roughened solar collector duct and three side glass cover is used smooth solar collector duct while the bottom surface is provided with wood of adequate thickness to reduce bottom heat loss.

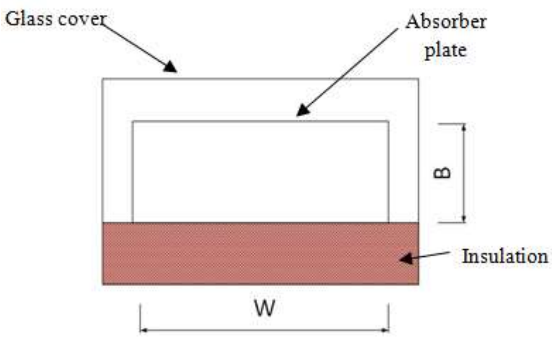

Fig 1a

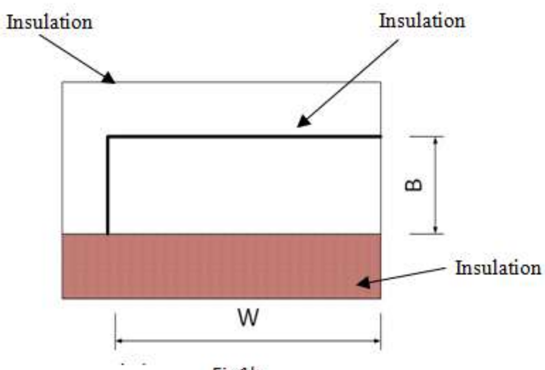

Fig 1b

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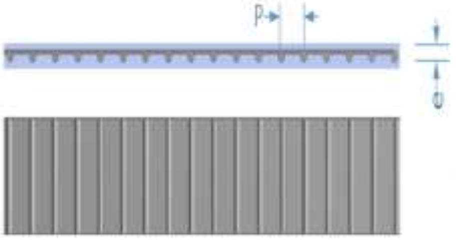

Fig.2a side absorber plate

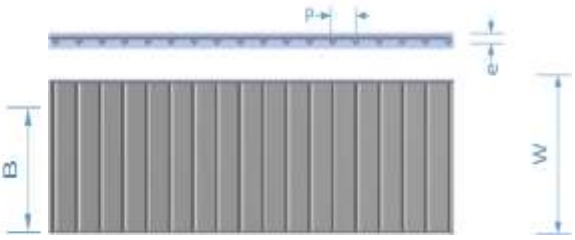

Fig.2b top absorber plate

Relative roughness and roughness pitch are same in side plate and top plate.
Top and side plate of the solar air heater duct collects solar radiation and acts as absorber plate. Fig 2a top view of side absorber plate and fig 2b shows the top view of top absorber plate.

III. ANALYSIS

1. Fluid flow analysis

The following analysis is purely based on the approach used by (Prasad ET. Al, 1984) who analyzed for the fluid flow in rectangular duct having three side artificial roughness and (Prasad and saini 1988) who analyzed for one side roughened rectangular solar air heater duct. Fig 1a and 1b shows four side smooth solar air heater duct and two side artificially roughened solar air heater duct respectively. Both of these ducts have identical dimensions and cross sectional area of W × B with the assumption of W >> B.

Friction factor for fully developed turbulent flow in a four side smooth duct, is given by the relation

$$f_s = \frac{\tau_s}{\frac{1}{2}\rho v_s^2} \tag{7}$$

Similarly, the friction factor for fully developed turbulent flow in a four side rough duct, is given by the relation

$$f_r = \frac{\tau_r}{\frac{1}{2}\rho v_r^2} \tag{8}$$

Average friction factor for fully developed turbulent flow for a duct having two side smooth and two side rough wall is given by the relation

$$\bar{f}_r = \frac{\bar{\tau}_r}{\frac{1}{2}\rho \bar{v}_r^2} \tag{9}$$

If τ_{2r} is the shear stress on the rough surfaces of four sided duct having two roughened walls and two smooth wall , and τ_{2s} is shear stress on two smooth wall surfaces .τ_s is the shear stress in the duct having all four wall rough and $\tau_r$is the shear stress in the duct having all four wall smooth.

The following equivalence between the total shear forces of the duct system can be established

$$[(W+B)\tau_{2r}+ (W+B)\tau_{2S}]L \cong [(W+B)\tau_r+ (W+B)\tau_S]L \tag{10}$$

$$\text{Or, } \tau_{2r} + \tau_{2S} \cong \tau_r + \tau_S \tag{11}$$

Here τ_{2r} is slightly less than τ_r due to two rough walls and τ_s will be slightly less than τ_{2s} due two rough walls i.e. $\tau_{2r} = \tau_r - \epsilon_1$(12) { ϵ_1 *and* ϵ_2 >0 and is a very small quantity }

$$\tau_{2s} = \tau_s + \epsilon_2 \tag{13}$$

Combining above two equations we will get,

$$\tau_{2r} + \tau_{2s} = \tau_r - \epsilon_1 + \tau_s + \epsilon_2 \tag{14}$$

Or, $\tau_{2r} + \tau_{2s} = \tau_r + \tau_s + (\epsilon_2 - \epsilon_1)$

$$\text{Or, } \tau_{2r} + \tau_{2s} \cong \tau_r + \tau_s \tag{15}$$

Since, $\epsilon_2 - \epsilon_1$ very small and can be neglected.
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$\bar{\tau}$(2W+2B)L = $[\tau_{2s}(W+B) + \tau_{2r}(W+B)]$L (16)

Or, $\bar{\tau} = \frac{(W+B)\{\tau_{2r}+\tau_{2s}\}}{2(W+B)}$

Or, $\bar{\tau} = \frac{\tau_{2r}+\tau_{2s}}{2}$ (17)

Or, $\bar{\tau} = \frac{\frac{1}{2}f_s\rho v_s^2+\frac{1}{2}f_r\rho v_r^2}{2}$ (18)

Or, $\bar{f_r} = \frac{\frac{1}{2}f_s\rho v_s^2+\frac{1}{2}f_r\rho v_r^2}{2\times\frac{1}{2}\rho\bar{v}_r^2}$ (19)

Since, $\frac{1}{2}\rho v_s^2 \approx \frac{1}{2}\rho v_r^2 \approx \frac{1}{2}\rho\bar{v}_r^2$ (20)

So, $\bar{f_r} = \frac{f_s+f_r}{2}$ (21)

Friction factor in four side smooth duct can be obtained from moody chart for fully developed turbulent flow whereas friction factor in four side rough duct can be determined using friction similarity law. Webb et al. 1971; sheriff and Gumley. 1966; Han, 1984; Prasad and Saini, 1988 had used friction similarity law to correlate friction data in tubes and ducts with different roughness element for fully developed turbulent flow.
According to Webb et. al 1971 the expression for f_r is given by

$$f_r = \frac{2}{\left[0.95\left(\frac{p}{e}\right)^{0.53}+2.5ln\left(\frac{D}{2e}\right)-3.75\right]^2} \quad (22)$$

The expression for average friction factor $\bar{f_r}$ for the present case of solar air heater duct can be best predicted by

$$\bar{f_r} = \frac{\frac{2}{\left[0.95\left(\frac{p}{e}\right)^{0.53}+2.5ln\left(\frac{D}{2e}\right)-3.75\right]^2}+f_s}{2} \quad (23)$$

2. *Heat transfer analysis*

Wall similarity law is used for temperature profile to correlate the heat transfer analysis with the assumption that the heat transfer roughness function, $G_H(e^+,\ pr)$, is only depends on roughness geometry and it is independent of duct dimensions. According to Dalle Donne and Mayer, 1977 the equation of Webb. Etal 1971 can be used for similar roughness geometry which is given by

$$St_r = \frac{f_r/_2}{1+\sqrt{\left(\frac{f_r}{2}\right)}\left[4.5(e^+)^{0.28}pr^{0.57}-0.95\left(\frac{p}{e}\right)^{0.53}\right]} \quad (24)$$

The above equation is for a rough tube, but in our present situation, there is roughness and heat flux in two sides of the duct. Therefore, the average value of friction factor $\bar{f_r}$ and average value of Stanton number, $\overline{St_r}$ are used in place of f_r and St_r.
So equation becomes

$$\overline{St_r} = \frac{\bar{f_r}/_2}{1+\sqrt{\left(\frac{\bar{f_r}}{2}\right)}\left[4.5(e^+)^{0.28}pr^{0.57}-0.95\left(\frac{p}{e}\right)^{0.53}\right]} \quad (25)$$

The values of average Nusselt number can be written as

$\overline{Nu_r} = \overline{St_r}$RePr

$$\overline{Nu_r} = \frac{\bar{f_r}/_2}{1+\sqrt{\left(\frac{\bar{f_r}}{2}\right)}\left[4.5(e^+)^{0.28}pr^{0.57}-0.95\left(\frac{p}{e}\right)^{0.53}\right]}\text{RePr} \quad (26)$$

h $= \frac{\overline{Nu_r}ka}{D_h}$ (27)

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3. *Thermal performance*

According to ASHARE recommendation the thermal efficiency of a solar collector can be expressed by the equation

$$\eta = F_R[(\tau\alpha) - U_L(\frac{t_i - t_a}{I})] \quad (28)$$

Where F_R is known as collector heat removal factor which is ratio between actual heat transfer rate to maximum possible heat transfer rate. F_R is related to collector efficiency factor F' by the following relation

$$F_R = \frac{mc_p}{A_c U_L}\left[1 - \exp\left(\frac{A_c U_L F'}{mc_p}\right)\right] \quad (29)$$

Generally the inlet temperature for single flow solar collector inlet temperature to collector is the ambient temperature i.e. $t_i = t_a$ which reduces (28) into $\eta = F_R(\tau\alpha)$ which is a function of temperature therefor cannot be used for temperature estimation.
Biondi et al. introduce the new equation for efficiency calculation

$$\eta = F_o[(\tau\alpha) - U_L(\frac{t_o - t_i}{I})] \quad (30)$$

Where $F_o$is collector heat removal factor refereed to outlet temperature which is calculated by the relation:

$$F_O = \frac{Gc_p}{U_L}\left[\exp\left(\frac{U_L F'}{Gc_p}\right) - 1\right]. \quad (31)$$

4. *Thermal performance prediction*

The thermal performance of solar air heater can be analyzed based on the heat transfer process in the collector. By utilizing the correlation developed for heat transfer co-efficient (Nusselt number) and friction factor, various loss co-efficient and gain factors are calculated which is used for estimating the efficiency of the solar collector.

Tabor proposed top plate loss co-efficient can be calculated b

$$U_T = \left[\frac{N}{\left(\frac{c}{t_p}\right)\left[\frac{t_p - t_a}{N+f}\right]^{0.33}} + \frac{1}{h_w}\right]^{-1} + \frac{\sigma(t_p - t_a)(t_p^2 - t_a^2)}{[\varepsilon_p + 0.05N(1-\varepsilon_p)]^{-1} + \left[\frac{2N+f-1}{\varepsilon_g}\right] - N} \quad (32)$$

Where:

f=$(1 - 0.04h_w + 0.005h_w^2)(1 + 0.091N)$ *and* $c = 365.9(1 - 0.00883\beta + 0.0001298\beta^2)$

Here h_w is a convection coefficient and its value, as suggested by McAdams is

$$h_w = 5.7 + 3.8V_w \quad (33)$$

Here V_w is air flowing velocity.
The above correlation is valid for top side absorber plate which is losing heat from top side only but in our design losses heat from one side wall also, but still it can predict the loss factor because height of the collector duct is very less as compared to the breath of the collector.

Overall heat loss co-efficient U_L is determined from:

$$U_L = U_T + \frac{k_i}{t} \quad (34)$$

A computer program code has been generated for the calculation of efficiency of the collector through iteration.

Following steps has been taken for the calculation.

1. Initial inlet temperature t_i is assumed equal to the ambient temperature t_a =300 k and final temperature rise in the collector is assumed $30°c$. Mean plate temperature is assumed $10\ °c$
2. Top loss coefficient, overall loss coefficient is evaluated using (6), (27), (32) and (34).
3. Useful energy gain rate Q_u is estimated using

$$Q_u = A_p[I(\tau\alpha) - U_L(t_p - t_a)] \quad (35)$$

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4. Heat removal factor based on outlet temperature from (31) and actual heat gain rete Q_{uc} is evaluated using relation

$$Q_{uc} = F_o[I(\tau\alpha) - U_L(t_o - t_i)] \quad (36)$$

5. Difference between Q_u and Q_{uc} is compared with adopted tolerance limit, if it is below acceptance limit then the above steps are repeated until desire level of tolerance is reached.

IV. RESULTS AND DISCUSSION

Using obtained co-relations different performance parameters are calculated against Reynolds number for different relative roughness height and relative roughness pitch

Fig 4a represents the effect of roughness pitch on the average friction factor for constant relative roughness value of e/D = 0.0135 at varying values of Reynolds number, while the fig. 4b shows the effect of relative roughness height on friction for constant value of roughness pitch p/e =10, for varying values of Reynolds number Both figure also shows the investigation made by B.N. Prasad et.al 2014 for three side artificially roughened solar air heater duct From Fig.4a and Fig 4b. It has concluded that average friction factor $\overline{f_r}$, increases with increasing relative roughness height as well as relative roughness pitch because of increase in turbulence.

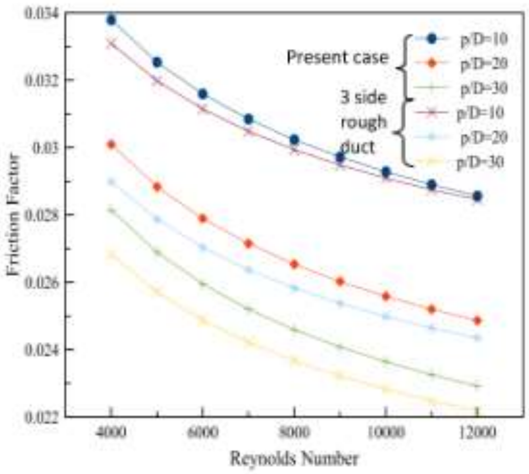

Fig.4a

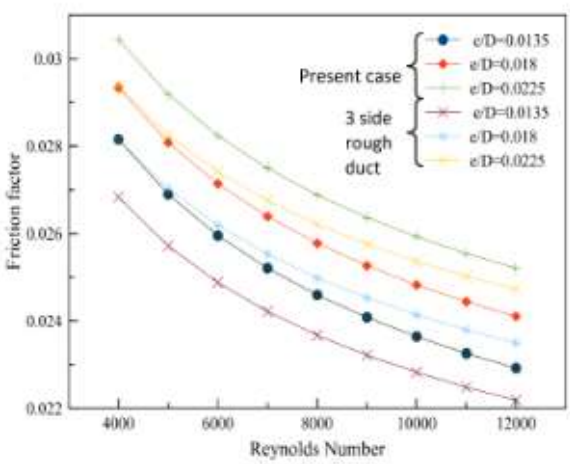

Fig.4b

Friction factor Decreases with increasing values of Reynolds number. Comparing the present study with B.N. Prasad work it is observed that friction factor for three side roughened duct is higher than the present study which is due to the extra roughened surface present on third side of the collector duct. From fig. 4c which depicts the relation between friction factor and Reynolds number for relative roughness e/D=0.018 for various relative roughness pitches it is observed that friction factor for higher values of relative roughness pitch (e/D) in both the studies are nearly equal whereas at lower values of relative roughness pitch, present model has friction factor about 4.7 % lower than the three side roughened collector duct, it is marginally small because of very small height of the duct which contribute for reduction in friction factorFig.4d shows the effect of relative roughness pitch on average Nusselt $\overline{Nu_r}$ number for constant values of relative roughness height of e/D=0.0135 at varying values of Reynolds number while the fig.4e shows the variation of average Nusselt number with relative roughness height for constant relative roughness pitch of p/e=10 at various Reynolds number. Nusselt number increases with increase in relative roughness pitch for constant values relative roughness height whereas for constant values of p/e=10 at various Reynolds number. Nusselt number increases with increase in relative roughness pitch for constant values relative roughness height whereas for constant values of relative roughness pitch, enhancement in Nusselt number is not recognizable at lower Reynolds number. In both of the cases Nusselt number increases with increase in Reynolds number. Nusselt number for two side roughened duct is slightly higher than three side.

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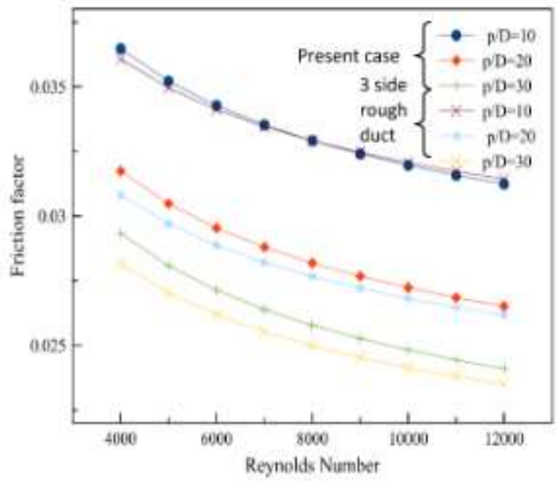

Fig.4c

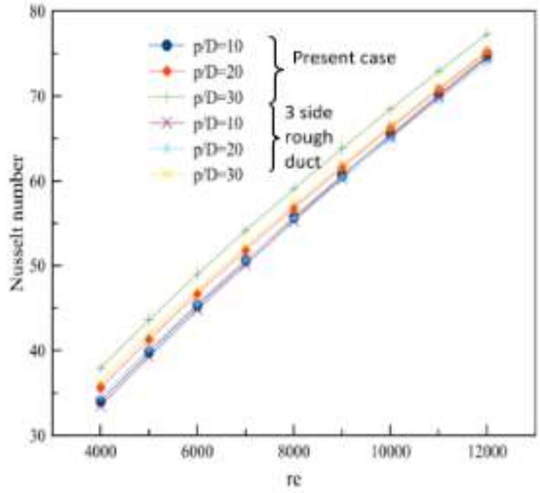

Fig.4d

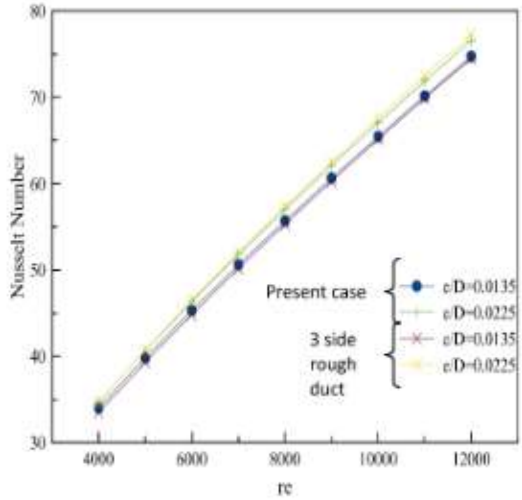

Fig.4e

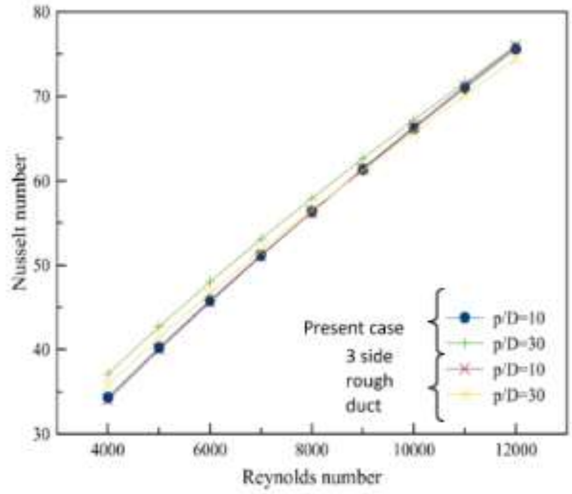

Fig.4f

Roughened duct for higher values of relative roughness pitch whereas Nusselt number is higher for three side roughened duct for higher values of relative roughness height.

Fig. 4f represents variation of nusselt number for relative roughness height e/D=0.018 for different relative roughness pitches (p/e) , it shows that nusselt number for three side roughened duct always higher than the two side roughened duct for all values of relative roughness pitch.

Stanton number is also calculated and plotted against flow Reynolds number for different relative roughness pitch and relative roughness height.

Fig 4g shows Stanton number and friction factor for constant relative roughness height of e/D =0.0135 at various relative roughness pitch. Graph shows there is decrease in Stanton number as flow Reynolds numberIncreases and Stanton number decreases with increase in relative roughness pitch. Stanton number is lower at lower relative roughness pitch for lower Reynolds number, and for higher Reynolds number it becomes larger than the higher relative roughness pitch

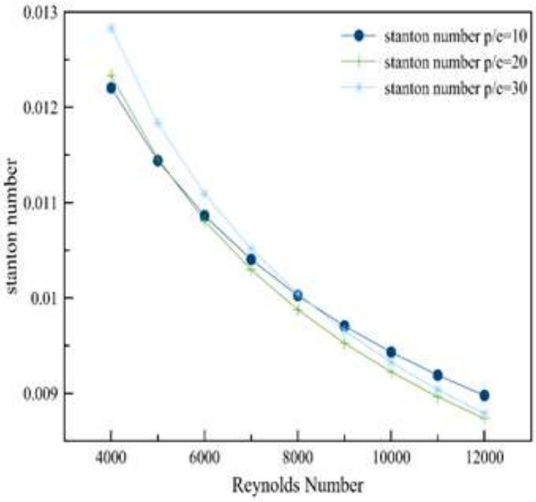

Fig.4g

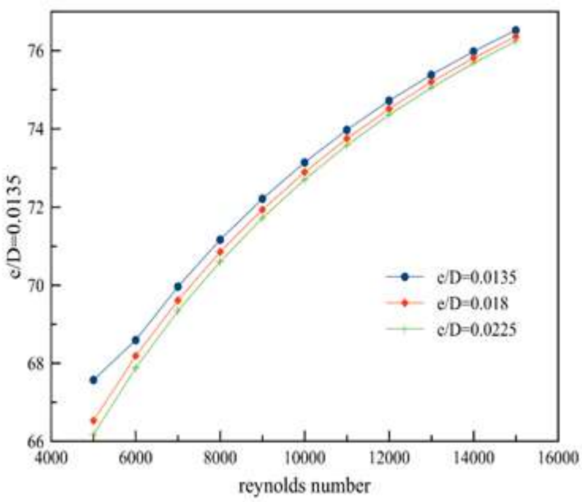

Fig.4h

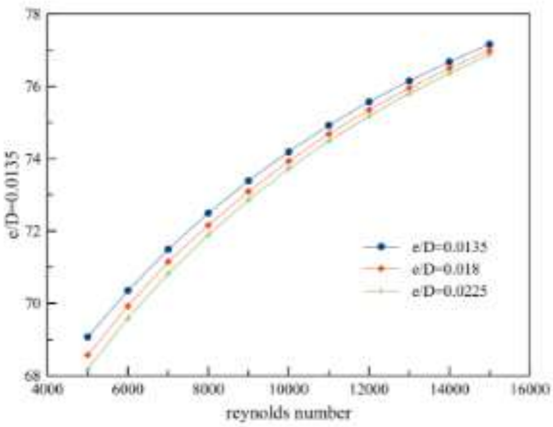

Fig.4i

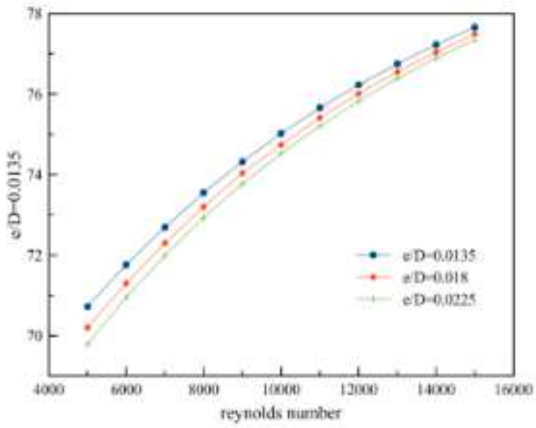

Fig.4j

Fig. 4h , fig 4i and fig. 4j represents the plots between thermal efficiency and Reynolds number for various relative roughness height at relative roughness pitch p/D=10 ,p/D=20 and p/D=30 respectively. Plot shows that efficiency increases with decrease in relative roughness height because of grater intermixing of high temperature bottom layer fluid with low temperature incoming air.

V. CONCLUSIONS

On the basis of the results obtained and discussion the following conclusion can be drown:-

1. Heat transfer and fluid flow analysis of rectangular solar air heater duct having two side (one top and one side wall) artificially roughened surface with two side glass cover.
2. Correlations for Average friction factor and heat transfer co-efficient have been developed in terms of geometrical parameters
3. Average Nusselt number and Average friction factor for different values of relative roughness height, relative roughness pitch and Reynolds number are calculated.
4. Maximum 7.4 % decrease in friction factor is recorded as compared to three side roughened duct.
5. In this study it is found out that substantial increase in thermal efficiency in solar air heater having roughness elements as a transverse wire on the two side absorber plate. The efficiency increases with increasing air flow rate though for higher air flow rate. In this solar air heater it is found out that solar irradiation is also a factor to effect the solar efficiency. The maximum Nusselt number found 77.32 for a relative pitch p/e=10, roughness pitch of e/D= 0.0225 at Reynolds number Re = 12000 and corresponding friction factor is 0.025207.

6. The maximum efficiency obtained is nearly 78% which is 4% higher than varun et.al
7. The higher efficiency is obtained because in matlab because of no leakage as well as environmental consideration is taken but in real situation there is always some weather situation which affect the efficiency.it also depends on the Reynolds number of the flow and dependent parameters on the Reynolds number

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