Comparative study of Different Geometry of Ribs for roughness on absorber plate of Solar Air Heater -A Review

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Abstract - *This paper includes study of Different Design of Solar Air Heater used for application in the field of agriculture and many other drying processes. In this paper a brief overview of various technique used for artificial roughness is taken into the consideration. Turbulent flow plays very important role in the field of heat transfer because it increase heat transfer and to generate this turbulent flow various types of artificial roughness such as ribs, grooves, dimple, metal mesh etc. is used by many researchers in previous research.*

***Key Words*: Solar Air Heater, Artificial roughness, Solar Energy, Turbulent flow etc.**

1.INTRODUCTION

Increasing demand of energy and limited resources of conventional energy is biggest threat for human being in future because our generation life style is wholly depends on machinery and electronics which are operated by power (Electricity) and in present scenario the main fuel for production of electricity is fossils fuel which is in verge of ending after 50 years from today according to some surveys. This threat can be thrashed by development in the field of solar energy because this is known to everybody solar energy is a biggest form of energy and free of cost available to us for millions of year. Solar Energy is the most abundant Energy and freely available to us. It emits Energy at a rate of 3.8×10^{23} kW but approximately 1.8×10^{14} kW is intercepted by earth. Heat and Light are the form of solar Energy which are directly available to us in free of cost and conversion of these form to other form of Energy is required for various applications. For conversion of Solar Energy to Thermal Energy for heating application Solar collectors are used costs very efficient and easy to manufacture.

2. NOMENCLATURE

A_p	area of absorber plate, m^2
A_o	cross Section area of orifice,
C_d	coefficient of discharge of orifice
C_p	Specific heat of air at constant pressure, J/kg K
D	hydraulic diameter of duct, m
G_d/L_v	Relative Gap Distance
g/e	relative gap width
h	convective heat transfer coefficient, W/m^2 K
h_m	average convective heat transfer coefficient, W/m^2K
m	mass flow rate, kg/s
Nu	Nusselt number of roughened duct
(ΔP_o)	difference of manometric fluid level in U-tube manometer, m
$(\Delta P)_d$	difference of water column level in micro-manometer, m
Q_u	useful heat gain rate, W
T_f	mean temperature of air, K
T_i	inlet temperature of air, K
T_o	outlet temperature of air, K
T_p	average plate temperature, K
W/w	relative roughness width ratio
β	ratio of orifice diameter to pipe diameter

(e/D) relative roughness height

ρ ensity of air, kg/m^3

3. CLASSIFICATION OF SOLAR AIR HEATER ON THE BASIS OF ABSORBING SURFACE

3.1 Simple flat-plate collector: This is the simplest and most commonly used type of collector. In its simplest form it is composed of one or two glazings over a flat plate backed by insulation. The path of air flow may be either above or below or the absorber plate.

3.2 Finned-plate collector: This is a modified version of the type (3.1) collector, where the heat transfer co-efficient is increased by using fins on the flat plate absorber, and in certain design the surface is made directionally selective. The fins are usually located in the air-flow passage.

3.3 Corrugated- plate collector: This is another variation of the simple flat-plate design, in which the absorber is corrugated either in rounded troughs or V-troughs. This increases the heat transfer area and may make the surface directionally selective.

3.4 Matrix type collector: In this design an absorbing matrix is plate, cotton gauze or loosely packed porous material. This type of collector offers a high heat-transfer to volume ratio; it may also offer low friction losses depending on the design.

3.5 Overlapped transparent plate type collector: This type of collector is composed of a staggered array of transparent plates which are partially blacked. The air flow paths are between the overlapped plates.

3.6 Transpiration collector: The transpiration or porous bed design is a variation of type (3.4), in which the matrix material is closely packed and the backed and the back absorber-plate is eliminated. The air flow usually enters just under the innermost cover and flows downward through the porous bed and into the distribution ducting.

4. ARTIFICIAL ROUGHNESS

The thermal efficiency of Solar Air Heater is poor due to low heat transfer coefficient between absorber plate and air flowing over absorber plate because laminar sub layer is formed over absorber plate and Laminar layer is responsible for low heat transfer so to break this Laminar sub layer various types of roughness surface is created on the absorber plate such as ribs, Rib-grooved, metal mesh, Dimple etc. Following are the dimensionless geometrical parameter used to characterized roughness.

4.1 Relative Roughness Pitch (p/e):- It is the ratio of Distance Between two consecutive ribs and height of ribs.

4.2 Relative Roughness Height (e/D):- It is the ratio of rib height to equivalent diameter of the air passage.

4.3 Angle of Attack (α):- It is inclination of rib with direction of air flow in the duct.

4.4 Aspect Ratio:- It is ratio of duct width to duct height.

5. LITERATURE REVIEW

5.1 1 Investigation By **Anil Kumar et al.[5]** for **V Shaped rib with gap roughness** the effect on heat transfer and fluid flow encompassed Reynolds no range 2000 to 20000, relative width ratio (W/w) of 6, relative gap distance G_d/L_v of 0.24-0.80, relative gap width (g/e) 0f 0.5-1.5, relative roughness height (e/D) of 0.043, relative roughness pitch (P/e) of 10, angle of attack (α) of 60. In mentioned condition thermo-hydraulic parameter is found to be best for the relative gap distance of 0.69 and the relative gap width of 1.0.

Equation used for calculation of heat transfer coefficient h

$$h = \frac{Q_u}{A_p.(T_p - T_f)}$$

$$Q = mC_p(T_o - T_i);$$

$$m = C_d \times A_o[2 \times \rho(\Delta P_o)/(1 - \beta^4)]^{0.5}$$

$$Nu = \frac{h.D}{k}; \qquad f = \frac{2.(\Delta P)_d.D}{4\,L\,\rho\,V^2}$$

5.2 T.T.Wong et al.[7] experimentally and numerically investigated the force convection and flow friction of a turbulent flow friction of a turbulent airflow in a horizontal air-cooled rectangular duct, with square sectioned cross ribs mounted on its bottom surface. Dimensions of the cross ribs were 6.37mmx6.37mm. Reynolds number of the fully turbulent flow was maintained constant at 12380. Heat was supplied uniformly to the airflow via bottom surface only. Experiment done on effects of varying the angle formed by the cross ribs between 30° to 120° on the force convection and results was forced convection can be enhanced by mounting cross-ribs on the heated surface of a rectangular duct and average heat transfer coefficient would be obtained at an angle between 60°and 70° and this range is optimum corresponding to the maximum force convection from the ribbed duct to the airflow.

$$Nu_e = \frac{h_m D_e}{k}$$

$$f = \frac{(\frac{\Delta p}{L})D_e}{\frac{1}{2}\rho u_m^2}$$

$$h_m = \frac{Q}{A(T_{wm} - T_{fm})}$$

5.3 S.Y. Won et al[6] compared spatially resolved local flow structure, spatially averaged flow structure and Nusselt number for stationary channels with aspect ratio of 4 and rib turbulators inclined at 45°. Two different rib arrangements with parallel and perpendicular orientation on two opposite surface are investigated at Reynolds number from 480 to 18300. Comparison showed important local Nusselt number difference for the two rib arrangements, especially just upstream of the ribs, which are due to significant difference in global and local primary and secondary flow characteristics.

5.4 A. R. Jaurker et al.[4] experimentally investigated heat transfer can be more improved by introducing the groove between the ribs. Experimentally investigation encompassed the Reynolds number range from 2000 to 21000; relative roughness height 0.0363; relative roughness pitch 6 and relative groove position of 0.3 to 0.7. Result showed the heat transfer coefficient for rib-grooved arrangement is higher than that for the transverse ribs, whereas the friction factor is slightly higher for rib-grooved arrangement as compared to that of rectangular transverse ribs of similar rib height and rib spacing.

5.5 Y. M. Zhang et al.[8] studied on the effects of compound turbulators on friction factors and heat transfer coefficients in rectangular channel with two opposite ribbed-groove walls, Reynolds number ranging from 10,000 to 50,000 in channel having width to height ratio 10. They measured fully developed heat transfer coefficient and friction factors for spacings (p/e = 8, 10, 15, 20, 25 and 30). The result showed that the heat transfer performance of the rib-groove roughened duct is much better than the rib roughened duct. The rib-groove roughened wall enhance the heat transfer 3.4 times and pays 6 time the penalty, whereas the rib roughened wall with similar rib height and rib spacing, enhance the heat transfer 2.4 times and pays about the same pressure drop penalty.

5.6 Firth and Meyer[1] investigated the heat transfer and friction factor performance based on equal pumping power, of four different type of artificially roughened surface common namely.

- Square transverse ribbed surface.
- Helically ribbed surface.
- Trapezoidal ribbed surface.
- Three dimensional ribs.

The ribbed geometries of above cases are studied by them and found that:

- The square transverse ribbed surface has an overall performance, which compares well with the other surfaces considered.
- The helically ribbed surface has a thermal performance, which compares as closely with squares transverse ribbed surface.
- The performance of trapezoidal ribbed surface is substantially poorer then the other entire surface. The best overall thermal performance is given by the three dimensional ribs, which shows an improvement of over 15% compared with the trapezoidal transverse ribbed surface.

5.7 Han et al.[3] investigated the effect of rib shape, angle of attack and pitch to height ratio on friction factor and heat transfer coefficient. The main conclusions of the investigation are;

- At a given value of relative roughness height, e/D, the friction factor approaches a constant value as the Reynolds number increases, whereas the Stanton number continues to decreases with Reynolds number.
- The influence of rib shape on the Stanton number disappears at higher Reynolds number when the flow is in the completely rough region.
- For small value of relative roughness pitch (≈5), the flow which separates after each rib does not reattach before the succeeding rib while for a relative roughness pitch value of about 10, the flow does reattach close to the next rib. For larger rib spacing the reattachment point is reached and a boundary layer begins to grow before the succeeding rib is encountered, reducing both the average shear stress and heat transfer.
- The optimum thermo-hydraulic performance occurred at an angle of attack of about 45°.

5.8 Han et al.[2] investigated the effect of rib angle of attack and channel aspect ratio on the heat transfer and friction for developing flow in short rectangular channel (L/D=10 and 15) with a pair of opposite rib roughened walls for Reynolds number range of 10000 to 60000. The important conclusions drawn from the study are;

- The highest heat transfer and accompanying highest pressure drop is obtained at an angle of attack of 60°in the Square channel while the highest heat transfer and highest pressure drop occur at the angle of attack of 90° in the rectangular channel with an aspect ratio of 4.
- The best heat transfer performance in the square channel with rib angle between range from 30° to 45° is about 30% higher than with the transverse ribs for a constant pumping power . The best heat transfer performance in the rectangular channel (with W/H= 2 and 4) with angle of attack between 30°to 45° is only about 5% higher than with the transverse ribs on a constant pumping power basis.

5. CONCLUSION

As mentioned above by studying various research paper it is clear that heat transfer coefficient can be increased by using different types of artificial roughness and it can be further increased by using new types of roughness like new structure, geometry for roughness and conclusion is for heat transfer is summarized according to various researchers in following table.

Author	Parameter	Heat transfer	Friction	Remark
Kumar Anil et al[5]	Re 2000 to 20000 W/w – 6 G_d/L_v – 0.24 to 0.80 g/e – 0.5 to 1.5 p/e – 10 and angle of attack is 60°	Sufficient increase in Heat transfer	6.32 times of smooth surface	
Wong T T et al.[7]	Re 12380 e/D – 0.12 ϴ – 30° to 120°	Sufficient increase in Heat transfer	2 – 11 %	Highest heat transfer at 60° and 70°
Won S Y and Ligrani P M[6]	Re 480 to 18300 p/e – 10 AR 4	Sufficient increase in Heat transfer	Increased	
Zhang Y M et al.[8]	Re 10000to 50000 p/e – 8,10, 15,20,25 e/D – 0.028	3.4 times of smooth surface	3.2 times of smooth surface	at p/e =8 Highest heat transfer and largest p drop
Jaurker A R et al.[4]	Re 2000 to 21000 e/D - 0.0363	2.75 times	3.48 times of smooth surface	

	p/e - 6 g/p – 0.30 to 0.70			
Han et al.[2]	Re 10000 to 60000 L/D = 10 and 15 AR 4	30% Heat transfer increased	Friction increased	Highest at rib angle 30° to 45°

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