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Improved Thermal Performance of Solar Air Heater Using V-Rib with Symmetrical Gap and Staggered Rib

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ABSTRACT
The most efficient technique to increase the performance of solar air heater is to enhance the heat transfer by using artificial roughness in form of repeatedly used ribs on the absorbing heated absorber plate. In order to analyse the thermal performance and flow pattern of rectangular duct with aspect ratio (W/H) of 8, the present experimental investigation is performed with V-rib with Symmetrical Gap and Staggered Rib. The experiment has covered a Reynolds number (Re) range of 3000-14000, rib height 2 mm, pitch (P) 24mm, relative roughness pitch (P/e) of 12, gap width (g) 8mm, relative gap width (g/e) as 4 and angle of attack (α) 60^o, number of gaps on each sides of V-rib (Ng) 3, relative roughness height (e/Dh) 0.045, staggered rib pitch (P') 15.6mm, relative staggered rib pitch (P'/P) 0.65, staggered rib size (w) 20mm and relative staggered rib size (w/g) 2.5. Results have been compared with the smooth plate under similar flow condition to determine the enhancement in heat transfer and improvement in efficiency. Relative staggered rib pitch was kept at 0.65 and staggered rib size was kept as 2.5 times gap width.
Keywords: Artificial roughness, Symmetrical gap, staggered rib, solar air heater, V-rib

I. INTRODUCTION

The coefficient of heat transfer of a solar air heater is low because of minimum utilization of solar energy by the absorber plate used in solar air heater. It is attributed to the formation of a very thin boundary layer at the absorber plate surface commonly known as viscous sub-layer which acts as an insulating layer for heat convection. The coefficient of heat transfer of a solar air heater duct can be increased by providing artificial roughness on the heated wall (i.e. the absorbing plate). Providing artificial roughness makes the flow turbulent and disturbs the viscous sub layer formed very near to the heated surface. In order to improve the thermal performance it is desirable to generate the turbulence very near to the heated surface to break the viscous sub layer and increase the convective heat transfer from the surface. Several investigations have carried out before by using different types of artificial roughness. Prasad and Mullick [1] have used the wire having small diameter to make transverse artificial roughness. Transverse rib was reinvestigated by Prasad and Saini [2]. Gupta et al. [3] again used transverse roughness. They reported the maximum enhancement in Stanton number for the Reynolds number of 12000. An outdoor experimental study on transverse roughness was performed by Verma and Prasad [4]. The optimum thermo hydraulic performance of 71 % was obtained corresponding to the roughness Reynolds number of 24. The rib was modified by breaking the continuous transverse rib by Sahu and Bhagoria [5]. The heat transfer coefficient increased by 1.25 to 1.4 times by comparing it with the smooth absorbing plate under similar parameters. Gupta et al. [6] provided circular artificial roughness having inclination. They enhanced the thermal efficiency by 1.16-1.25 by comparing it with the smooth plate. The inclined rib with gap was investigated by Aharwal et al. [7]. V-shaped rib results in better performance as compare to the inclined one because of the development of secondary turbulence. V-shaped roughness was investigated by Momin et al. [8]. V-shaped roughness was further discretized by Muluwork et al. [9]. They considered the effect of parameters as relative roughness length ratio (B/S), relative roughness segment ratio (S'/S) and angle of attack (α). It was observed that the Stanton number for V-down discrete roughness is more than V-up as well as transverse discrete rib. The comparative and experimental study using V-discrete and V-continuous was carried out by Karwa et al. [10]. It was reported that the inclination of 60^oon rib gives better performance as compare to the transverse roughness. Another experimental study using V-discrete and V-discontinuous roughness was done by Karwa et al. [11]. Singh et al. [12] did his investigation with discrete V-down roughness. Thermal performance of solar air heater was investigated by Patil et al [13] using broken V-rib roughness combined with staggered ribs. V-rib having multi gap with the combination of staggered rib was investigated by Deo et al. [14]. Maithani et al. [15] investigated the performance of roughened solar air heater with V-shaped roughness having symmetrical gap. The value of number of gap and angle of attack ranges from 1 to 5 and 30 to 75 degree respectively.

Literature shows, no study has been conducted to investigate the performance of V rib with symmetrical gap and influence of staggered rib element on its performance. Therefore the present experimental investigation performed on the rectangular duct of solar air heater with three perfectly insulated sides and a side having heated surface with V shaped with symmetrical gap and staggered rib. The variation in the value of heat transfer coefficient, Nusselt number and efficiency against Reynolds number are studied.

II. EXPERIMENTAL INVESTIGATION

The schematic diagram of experimental setup is shown below in Fig.1. The dimensions of inner cross section of the wooden duct are 200mm x 25mm.The set up contains entry section, test section and exit section. Test section has length of 1500mm. Entry section of duct has a thermocouple at the mid section for the measurement of inlet temperature. At the test section of the duct an electric heater is placed over the test surface to obtain the required heat flux which may be varied from 0 to 1000 W/m^2. The average temperature of plate is taken by using thermocouples placed over the plate. Exit section contains thermocouples to get the bulk mean temperature of air at the outlet section. Control valve is provided to control the flow. Absorbing plate is made by GI sheet (galvanized iron sheet) having dimensions of 1500 mm x 216 mm X 1.2mm and the roughness is provided by using copper wire of roughness height (e) 2mm. The entire set up is covered with thermocol insulation to reduce the heat losses. Fig.2 shows the photograph of rib geometry used on absorber plates.

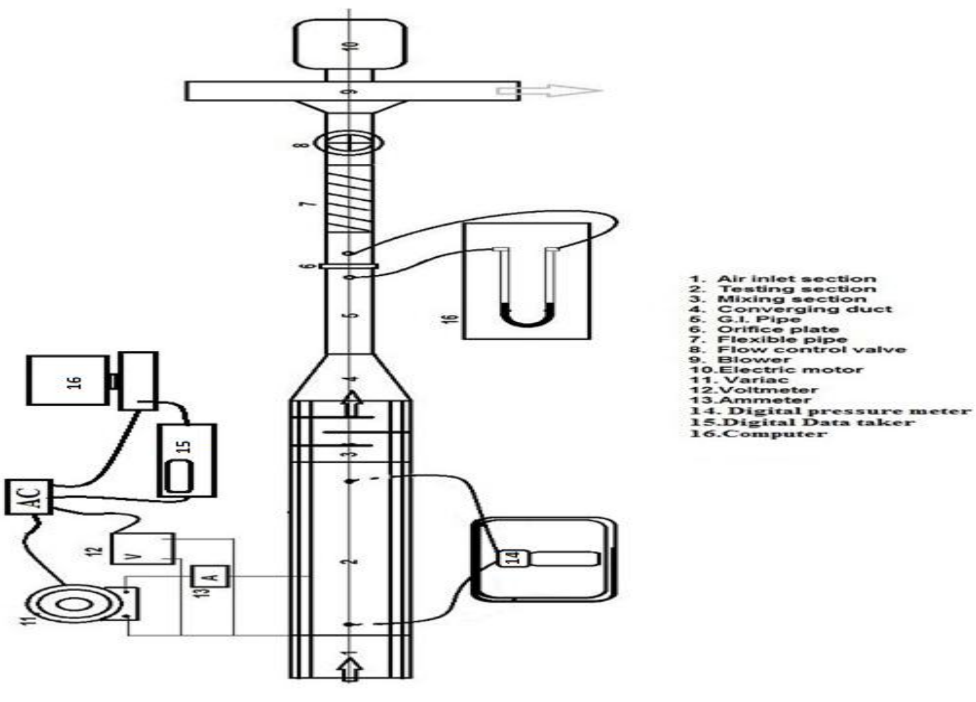

Fig.1. Schematic diagram of experimental set up

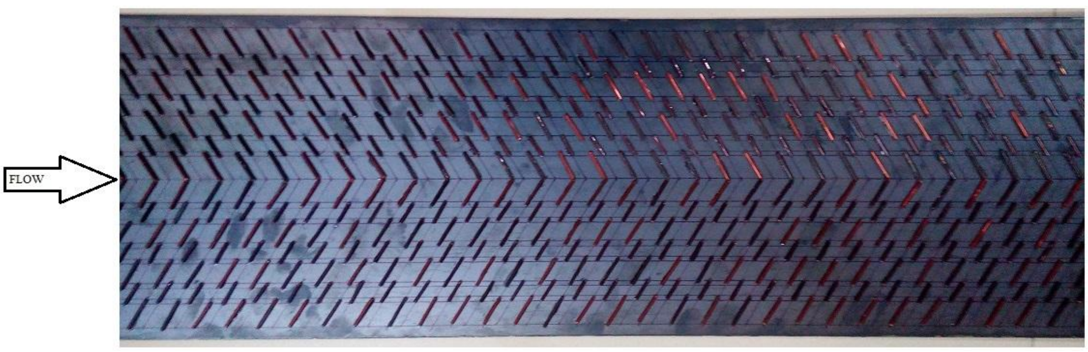

Fig.2. Photograph of plate having V rib with Symmetrical gap and staggered rib.

The operating parameters during investigation are listed in table 1.

Table 1: Operating Parameters

PARAMETERS	VALUE
Rib height (e)	2mm
Number of gaps on each side of V rib (Ng)	3
Relative gap width (g/e)	4
Gap width (g)	8mm
Angle of attack (α)	60^o
Relative roughness pitch (P/e)	12
Pitch (P)	24mm
Relative staggered rib size (w/g)	2.5
Staggered rib size (w)	20mm
Relative staggered rib pitch (P'/P)	0.65
Staggered rib pitch (P')	15.6mm
Reynolds number (Re)	3000-14000
Relative roughness height (e/Dh)	0.045
Channel aspect ratio (W/H)	8
Test length	1500mm
Hydraulic Diameter	44.44

III. DATA REDUCTION

The parameters under steady state condition like mean bulk temperature of air, plate temperature at various locations, heat transfer coefficient, Nusselt number of solar air heater can be calculated by using the relations as mentioned below.

1. Mean bulk air temperature (T_{fav})

Simple arithmetic mean of measured inlet and exit value of air under testing section

$$T_{fav} = (T_i + T_o)/2 \tag{1}$$

Where,
Ti = Inlet temperature of air in ˚C
To = outlet temperature of air in ˚C

2. Mean plate temperature (T_{pav})

Thermocouple wires are arranged at equidistance on the entire length of plate therefore it is the average reading of six points located at the distance of 215mm on the plate.

$$Tp_{av} = (T_{p1} + T_{p2} + T_{p3} + T_{p4} + T_{p5} + T_{p6}) / 6 \tag{2}$$

Where,
Tp_{1-6} = temperature of plate at different locations of plate.

3. Pressure drop across the orifice plate (□Po).

$$\Delta P_o = \Delta h \times 9.81 \times \rho. \tag{3}$$

Where,
Δh = Difference of water level in U-tube manometer
ρ = Density of water

4. Mass flow rate measurement (m).

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

Where,
β = Ratio of orifice diameter to pipe diameter (d_2/d_1)
C_d=Coefficient of discharge of orifice i.e. 0.62
A_o= Area of orifice plate, m^2
ρ = Air density at mean bulk air temperature

5. Reynolds Number

$$Re = \frac{VD_h}{\nu} \quad (5)$$

Where,
ν = Kinematic viscosity of air at t_{fav} in m^2/sec
D_h= Hydraulic diameter 4WH/2(W+H)

6. Heat Transfer Rate

$$Qa= mC_p(t_o-t_i) \quad (6)$$

Where,
C_p=Specific heat of air at constant pressure in kJ/kgK.

7. Convective heat Transfer Coefficient

$$h= Q_a/A_p(t_{pav}-t_{fav}) \quad (7)$$

8. Nusselt Number

$$Nu= hD_h/k \quad (8)$$

Where,
k= thermal conductivity

9. Efficiency

$$\eta = \frac{Q_u}{IA_p} \quad (9)$$

IV. EXPERIMENTAL PROCEDURE

The components and instruments are connected properly with the experimental set up for proper operation, after checking all the connections blower is then switched on. All the joints of GI pipe are made leak proof to eliminate the errors. The flow control valve is then open to adjust predetermined rate of flow of air for the testing section. Experiment is conducted to collect the data regarding heat transfer coefficient and frictional flow under quasi-steady state condition. Each change in rate of flow of air the system should attain a steady state before the data were recorded. The plate temperature at different flowing range of air is taken only after obtaining the steady state condition which is assumed to be reached when the plate temperature and air outlet temperature did not deviate over a 15 min. of time period. After reaching the quasi-steady state condition the inlet, outlet air temperature and plate temperature by using thermocouple wires, voltage and current of heater assembly by using voltmeter and ammeter, pressure drop across the orifice plate by using U tube manometer are recorded. For experimentation, seven different rate of air flow have been taken.

V. VALIDATION OF EXPERIMENTAL SET UP

The value of friction factor and nusselt number obtained from the experiments were compared with the values obtained from correlation of the Modified Dittus-Boelter equation for the nusselt number given by eq.(10) and modified Blasius equation for the friction factor given by eq. (11).

Modified Dittus-Boelter equation for nusselt number

$$Nu_s = 0.023\, Re^{0.8}\, Pr^{0.4}\, (2R_{av}/D_e)^{-0.2} \quad (10)$$

Where,

$2R_{av}/D_e$ = [(1.156+H/W-1)/ (H/W)]

Modified Blasius equation

$$f_s= 0.085(Re)^{-0.25} \quad (11)$$

The experimental set up is validated by comparing the experimental data by keeping the surfaces smooth with the theoretical results of smooth surface. The change in the value of Nusselt number with respect to Reynolds number for smooth duct is shown in Fig.3. and small deviation in the experimental and theoretical value is observed. Similar results have been found for the variation in friction factor with respect to the Reynolds number as shown in Fig.4. and shows better compliance between theoretical and experimental results.

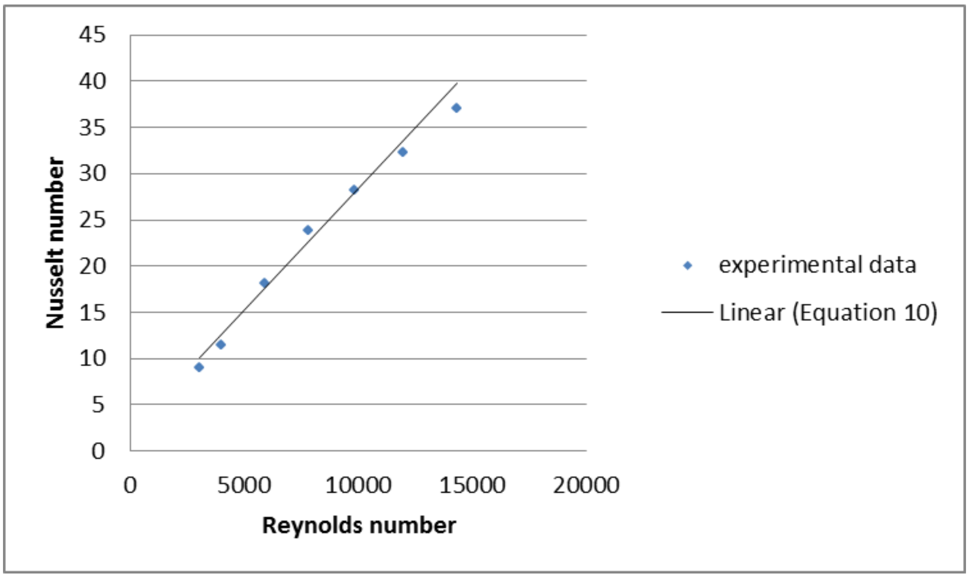

Fig.3. Comparison of experimental and formulated value of Nusselt no. Vs Reynolds no. for smooth plate

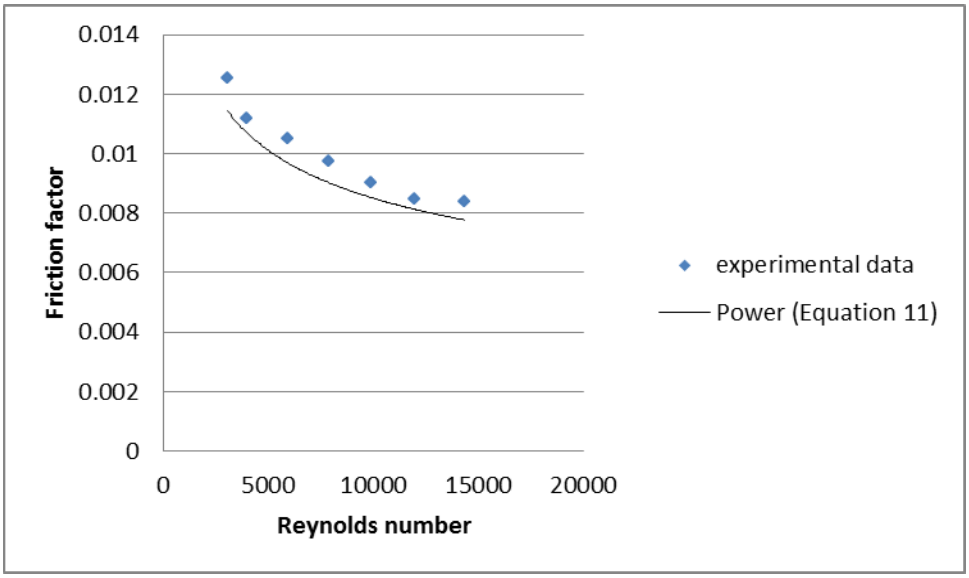

Fig.4. Comparison of experimental and formulated value of Friction factor Vs Reynolds no. for smooth plate

VI. RESULTS

Fig.5. shows variation of heat transfer coefficient against Reynolds number for both smooth and roughened plate. Fig.6. shows variation of Nusselt number against Reynolds number for both smooth and roughened plate. From both the figures, it is seen that heat transfer coefficient of roughened plate is better than that of smooth plate. Also as Reynolds number increases heat transfer coefficient also increases. Enhancement in heat transfer coefficient is due to the fact that the rib geometry increases the secondary flow and the gap provided accelerates the flow. The staggered rib arrangement scatters the flow to increase the heat transfer. Fig.7. shows the variation of efficiency for both smooth and the roughened plate. It is seen from figure that efficiency of roughened plate is more than that of smooth plate. It is due to improvement in rate of heat transfer of roughened plate as compare to the smooth plate.

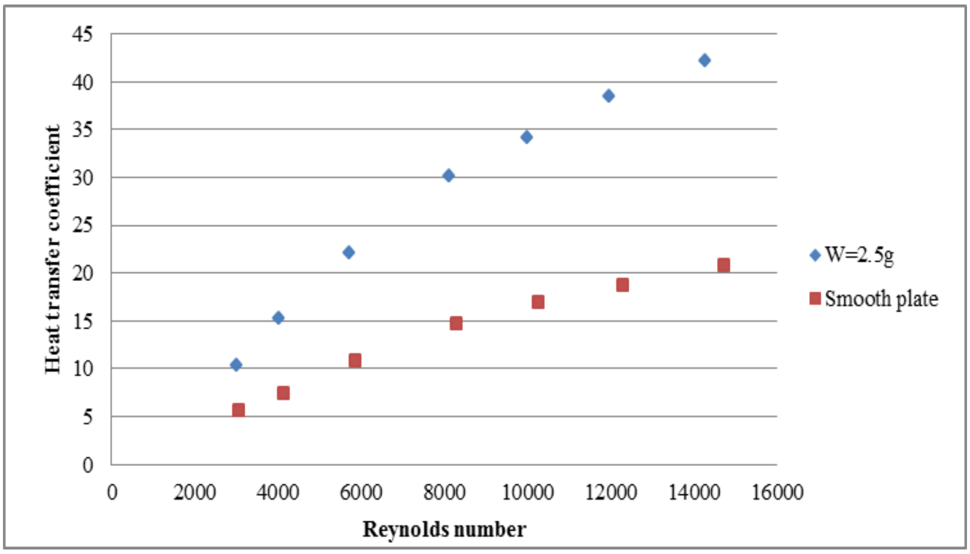

Fig.5. Variation of Heat transfer coefficient with Reynolds number

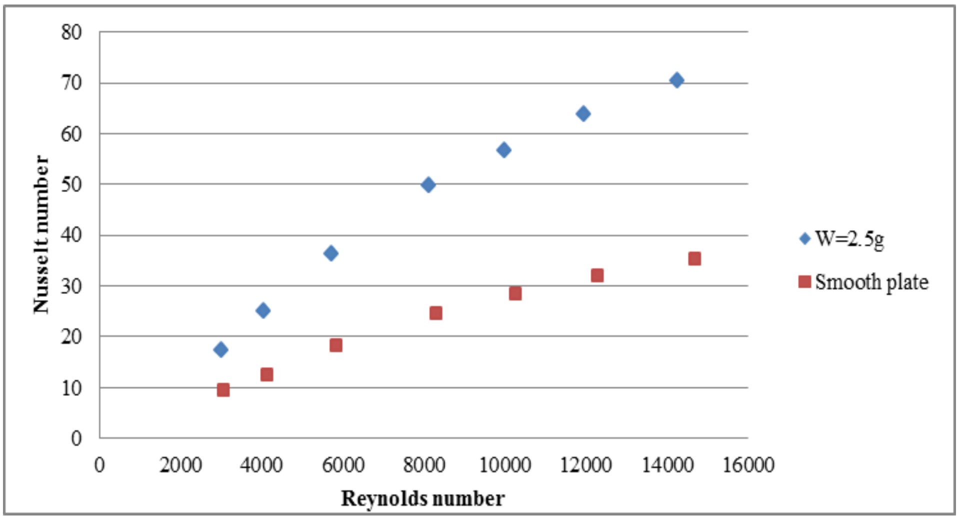

Fig.6. Variation of Nusselt number with Reynolds number

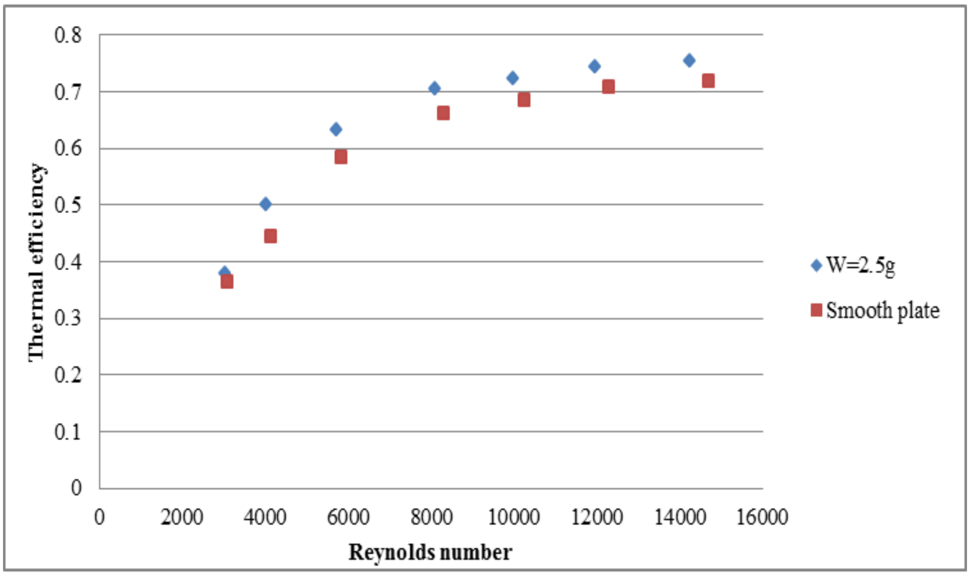

Fig.7. Variation of Thermal Efficiency with Reynolds number

VII. CONCLUSION

Use of V-rib with symmetrical gap with staggered rib leads to considerable enhancement in heat transfer. Also there is increase in efficiency as compared to smooth plate solar air heater due to improvement in heat transfer. The increase in heat transfer coefficient is due to release of flow through gap, reattachment of flow and scaterring of flow.

For proper understanding of thermo-hydraulic performance of the proposed rib geometry, experimental validation of various geometrical parameters and their optimisation is currently in the advanced stage at our research lab.

NOMENCLATURE

A_o	*Throat area of orifice plate (m^2)*
A_p	*Area of absorber plate (m^2)*
C_d	*Coefficient of discharge*
C_p	*Specific heat of air (KJ/kg K)*
D_h	*Hydraulic Diameter*
d/w	*relative gap position*
e	*Roughness height (mm)*
e/D_h	*Relative Roughness height*
g/e	*relative gap width*
H	*Height of air channel (m)*
h	*Convective heat transfer coefficient (W/m^2 ℃)*
I	*Intensity of solar radiation (W/m^2)*
k	*thermal conductivity of air (W/mK)*
L	*length of duct (m)*
m	*mass flow rate (kg/s)*
Ng	*Number of gaps*
P	*Roughness pitch*
P'	*Distance of staggered rib*
P/e	*Relative roughness pitch*
ΔP	*pressure drop*
Q_a	*Heat transfer rate*
Q_u	*useful heat gain (W)*
T_a	*Atmospheric temperature*
T_{fav}	*Average temperature of air*
T_i	*Initial temperature of air*
T_o	*Outlet temperature of air*
T_{pav}	*Average plate temperature*
V	*Velocity of air*
W	*Width of air duct*
W/H	*Channel aspect ratio*
w/g	*Staggering ratio*

Dimensionless parameters

f_s	*friction factor for smooth plate*
Nu	*Nusselt number*
Nu_s	*Nusselt number for smooth plate*
P_r	*Prandtle number*
Re	*Reynolds number*

Greek symbols

β	*Ratio of orifice diameter to pipe*
η	*Thermal efficiency*
α	*Angle of attack*
ρ	*Density of air*
ρ_m	*Density of manometer fluid*
ν	*Kinematic viscosity*

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