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Thermo-hydraulic Performance Analysis due to Relative Roughness Pitch in V-rib with Symmetrical Gap Roughened Duct of Solar Air Heater

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ABSTRACT: The present work investigates the thermohydraulic performance of a solar air heater ducts artificially roughened with V-rib with symmetrical gaps is determined. Symmetrical gaps are attached on each limb of the V-ribs. For simulating experimental indoor investigation of solar air heater duct, the test surface of duct is heated by means of an electric heater which supplies constant heat flux of 1000 W/m^2. Both the roughness parameters number of gaps (N_g) and relative gap width (g/e) values varies from 1-5, angle of attack (α) ranges from 30° to 75° and relative roughness pitch (P/e) is selected in the range of 6-12. The thermohydraulic performance parameter was maximum for P/e=10, α=60°, N_g=3 and g/e=4. A set of parameter showing maximum Thermohydraulic parameter was compared with the previous researchers.

Keywords: Nusselt number; friction factor; relative roughness pitch; thermohydraulic performance parameter.

INTRODUCTION: Solar air heaters find a wide application in the present energy scenario but due to low heat transfer between the absorber plate and air and formation of a boundary layer the efficiency is low. The heat transfer rate between the absorber plate and air can be enhanced by use of turbulence promoters in various forms of artificial roughness on absorber surface. Chamoli et. al. [1] reviewed various types of roughness geometry used for heat transfer enhancement. Numbers of investigations are carried out to analyse various roughness geometry to make the solar air heating system efficient. The artificial roughness that increases the heat transfer also enhances the pressure drop due to the increased friction which increases the pumping power requirement.

Prasad and Saini [2] experimentally studied the heat transfer and friction characteristics of a solar air heater duct roughened with transverse ribs on the underside of the absorber plate. Han et. al. [3] and Han and Zang [4] experimentally investigated and reported the enhancement of heat transfer for various angle of attack and rib configurations such as parallel ribs, crossed ribs, V-shaped ribs (upstream). Lau et al. [9] investigated the inclined and continuous transverse ribs and observed that the inclined ribs performs better than continuous transverse ribs for higher heat transfer rate because of higher turbulence created by primary and secondary flows interaction. Heat transfer and friction factor of angled and V-shaped ribs roughened absorber plate were investigated by Taslim et al. [10] and reported that that downstream V-shaped ribs shows higher heat-transfer as compared to upstream. Goa and Sunden [11] concluded that V-shaped ribs pointing downward are better for higher heat transfer than the rib pointing upward. Cho et al. [12] found that creating a gap in the inclined rib increases the turbulence intensity, which results in an enhancement in the heat transfer and performs better as compared to the continuous inclined rib roughness. Momin et al. [14] investigated the nusselt number and friction factor of v-shaped rib roughness. Varun et al. [15] reported the effect of transverse and inclined discrete ribs on heat transfer and friction factor.

In the present experimentation, performance of solar air heater ducts roughened with V-ribs with symmetrical gaps has been investigated. The Reynolds number was varied for experiment between 4000 and 18000. The effect of angle of attack on the Nusselt number, friction factor and thermohydraulic performance for a fixed value of the other geometrical parameters was evaluated and examined.

MATERIAL NAD METHODS: An experimental setup has been designed and fabricated as per guidelines suggested by ASHRAE standard [17] as shown in Fig. 1. The rectangular duct of 2395 mm length with a flow cross section of 300 mm×25 mm was fabricated. The duct consist of an inlet, test and exit section of 740 mm, 1100 mm and 555 mm length, respectively, equivalent to 16D, 24D and 12D, respectively. Constant flux of 1000 W/m^2 was provided by an electric heater. A 2 hp centrifugal air blower circulated the air through the test section. The mass flow rate of air was calculated by a U-tube manometer. 21 calibrated J-type copper constantan thermocouples were employed to measure the temperature of heated plate. The exit temperature of the air was measured at three different locations by placing thermocouples in the transverse direction to get the mean air temperature at the outlet of the duct with reasonably good accuracy. Inlet air temperature was measured by placing a thermocouple at the inlet section.

Roughness geometry and roughness parameters: For the experimental determination of the thermohydraulic performance the test plates were prepared with certain parameters as listed in table 1. Fig. 2 shows the picture of the V-ribs with symmetrical gaps roughness.

Table1: Range of parameters.

Sr. No.	Parameters	Value(s)	Level(s)
1	Relative roughness pitch (P/e)	10	4
2	Relative roughness height (e/D)	0.0433	1
3	Relative gap width (g/e)	4	1
4	Number of gaps (N_g)	3	1
5	Angle of attack (α)	30°- 75°	1
6	Reynolds number (Re)	4000-18000	8

Data reduction: In the experimentation under steady state conditions the temperature and pressure for plate and air at inlet, test and exit section were collected. Using the collected data the heat transfer rate from test plate to air in the duct was calculated. The heat-transfer coefficient 'h', Nusselt number 'Nu', Reynolds number 'Re' and friction factor 'f' were determined by the following equations:.

$$h = {Q_u}\Big/{A_p\left(T_p - T_f\right)} \tag{1}$$

where 'Q_u' rate of heat gain by the air is,

$$Q_u = mC_p\left(T_0 - T_i\right) \tag{2}$$

Mass flow rate, ṁ, is determined by the pressure drop across the orifice plate:

$$\dot{m} = C_d \times A_o\left[{2 \times \rho\left(\Delta P_a\right)}\Big/{\left(1 - \beta^4\right)}\right]^{0.5} \tag{3}$$

where C_d is the discharge coefficient and is evaluated as 0.62 by calibration.

Nusselt number is calculated by heat transfer coefficient using the equation:

$$Nu = \frac{h.D}{k} \tag{4}$$

Darcy–Wiesbach equation is used to determined the friction factor from flow velocity 'V' and pressure drop, $(\Delta P)_d$ across the test section using the expression:

$$f = \frac{2\left(\Delta P\right)_d D}{4\rho L V^2} \tag{5}$$

Performance evaluation: The Nusselt number ratio, Nu/Nu_s is defined as a ratio of Nusselt number of roughened duct to the Nusselt number of smooth duct. This ratio indicates the enhancement in the nusselt number by using artificial roughness as compared to smooth duct.

Thermohydraulic Performance Parameter

Thermohydraulic Performance Parameter makes possible the consideration of thermal and hydraulic performance and is given by Webb and Eckert [19] as thermohydraulic performance parameter

$$\eta = \frac{\left[{Nu}\big/{Nu_s}\right]}{\left[{f}\big/{f_s}\right]^{1/3}} \tag{6}$$

RESULTS AND DISCUSSION: The present experimental investigation deals with analysis of both heat transfer and friction loss of a solar air heater duct roughened with V-ribs with symmetrical gaps. The results for different relative roughness pitch and flow Reynolds number are discussed below. The use of artificial roughness generates secondary flows that leads to higher heat transfer rate. The results are compared with that of the smooth duct under similar operating conditions to show the enhancement in the nusselt number and the friction factor. For fixed values of number of gap (N_g) of 3, relative gap width (g/e) of 4, angle of attack (α) of 60° and relative roughness height (D/e) of 0.0433 for different relative roughness pitch (P/e) the variation of nusselt number and nusselt number ratios is presented in Fig. 1 and 2, respectively. The figures reveals that the increase in the relative roughness pitch up to 10 increase the Nusselt number and Nusselt number ratio and by increasing the relative roughness pitch beyond this level shows a decrement in the nusselt number and nusselt number ratios. For an relative roughness pitch of 10 the Nusselt number and Nusselt number ratio was highest and lowest for relative roughness pitch of 6 (Fig. 3). The enhancement is caused by interaction of the secondary flow and the boundary layer at the point of reattachments as the relative roughness pitch is increased to a value of 6. Beyond this value the number of reattachment points is less in number because the pitch size increases. Below the relative roughness pitch value of 10 the reattachment does not takes place hence it is not able to break the thermal boundary layer.

The variation of friction factor and friction factor ratio for different relative roughness pitch on roughened ducts is shown in the Fig. 4 and 5, respectively. It is observed that by increasing the relative roughness pitch upto 10 the friction factor and friction factor ratio increase and thereafter an increase in relative roughness pitch shows a downfall. The value of the friction factor and friction factor ratio is highest for an relative roughness pitch of 10 and is lowest for 6 (Fig. 5).

It is seen from the study that enhancement in heat transfer increases friction power. Thus it is required to find out a geometry that results in enhanced heat transfer and minimum friction power requirement. Roughness geometry with a value of thermal as well as hydraulic parameter more than unity is considered to be prolific and therefore, this parameter is used to compare the performance of various roughness geometries. Fig. 6 shows that the maximum value of the thermohydraulic performance parameter is for a relative roughness pitch of 10 for all values of Reynolds number.

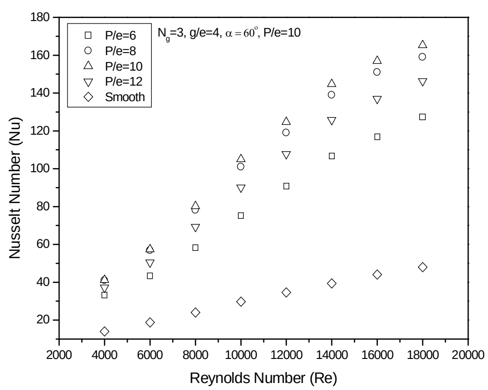

Fig. 1: Variation of Nusselt number with Reynolds number at different angle of attack.

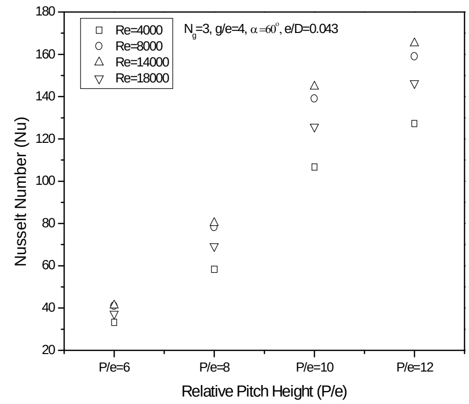

Fig. 2: Variation of Nusselt number ratios with Reynolds number at different angle of attack.

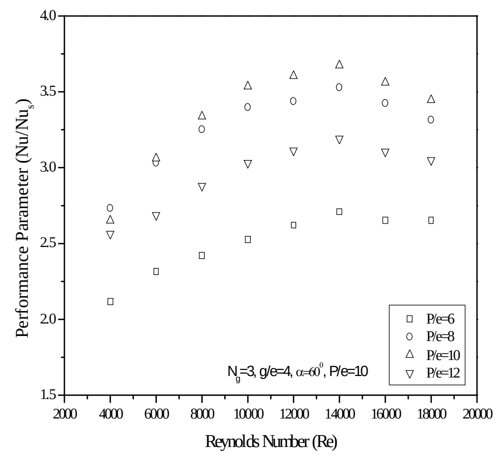

Fig. 3: Variation of Nusselt number with angle of attack at selected Reynolds number.

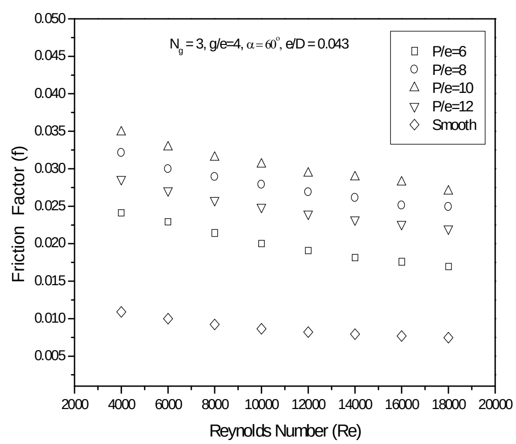

Fig. 4: Variation of friction factor with Reynolds number at different angle of attack.

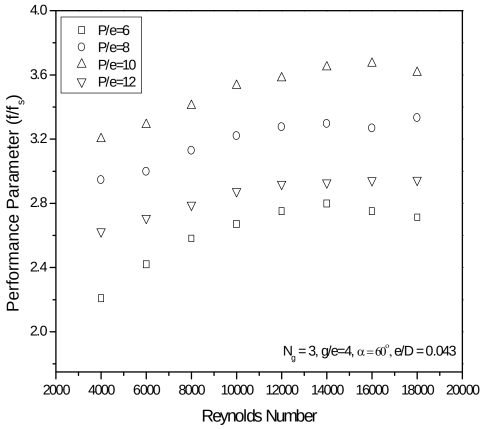

Fig. 5: Variation of friction factor ratios with Reynolds number at different angle of attack.

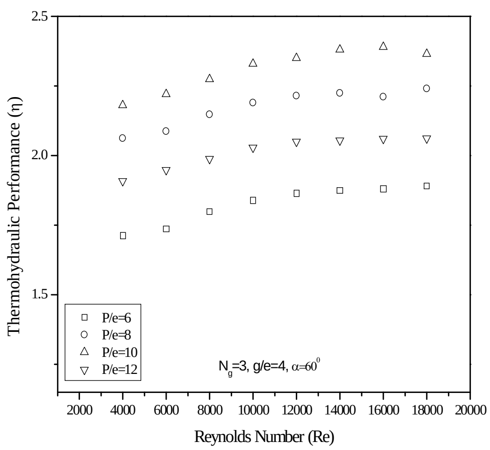

Fig. 6: Variation of thermo-hydraulic performance parameters with Reynolds number at different angle of attack.

Nomenclature: A_p *Absorber plate area(m^2)*
A_o *Cross-section area of the orifice (m^2)*
Pa Atmospheric pressure(N/m^2)
C_d *Coefficient of discharge of orifice*
f Friction factor of roughened duct
g Gap size (m)
Q_u *Heat-transfer rate to air (W)*
h Height of duct (m)
D Hydraulic diameter of duct (m)
$\dot{m}$ *Mass flow rate of air (kg/s)*
N_g *Number of gaps*
Nu Nusselt number of roughened duct
P Pitch of the rib (m)
W Width of duct (m)
e/D Relative roughness height
P/e Relative roughness pitch
g/e Relative gap width
Re Reynolds number
e Rib height (m)
Greek symbols:
α *Angle of attack*
η *Thermohydraulic performance parameter*

CONCLUSION: It can be concluded on the basis of the investigation carried out that the solar air heater ducts with V-ribs with symmetrical gaps roughness geometry that the use of present geometry enhances the performance of a solar air heater. The following conclusions can be drawn on this basis:

1. Angle of attack is a very important parameter in the study of Nusselt number and friction factor of the roughened duct.

2. The value of Nusselt number and friction factor increase with an increase of angle of attack upto 60° and then decreases.

3. 3 Symmetrical gaps on both limbs of V-rib results in a substantial improvement in the thermohydraulic performance.

4. V-ribs with symmetrical gaps with an angle of attack of 60° represent the best geometry thermo hydraulically.

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