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EFFECT OF ARTIFICIAL ROUGHNESS ON HEAT TRANSFER AND FRICTION CHARACTERSTICS OF DOUBLE PASS SOLAR AIR HEATER

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Abstract: Double pass solar air heater (DPSAH) consisted of rectangular duct provided with artificial roughness on both side of the absorber plate has been experimentally investigated. Circular ribs of aluminium wire is used to provide roughness to increase heat transfer coefficient between absorber plate and air. Ribs are attached to absorber plate at four different angle of attackbetween 30° to 75° . Experiment is carried out over the range of Reynolds Number from 4900 to 12000, and relative roughness height (e/D_h) varies from 0.022 to 0.044. Experimentally different values of Nusselt number(Nu). and friction factor(f_r) have been determined for various parameters. The enhance-ment in heat transfer and increment in the friction factor values of Nusselt number and friction factor have also been compared with the smooth one.

Keywords: *Thermohydraulic performance; Artificial roughness; Solar air heater; Triangular duct.*

1. INTRODUCTION

Our conventional energy resources are limited and likely to be depleted in sooner than later. They produces the energy on burning and pose serious environmental hazards. So, its necessary to focus on the renewable energy sources and solar energy is one of the most promising of the alternative energy resources. Solar air heater is one of the method to utilize this solar energy. A flat plate solar air heater in its simplest form consist of blackened absorber plate to transfer the absorbed energy to the flowing medium(air), transparent cover plate to reduce the convection and radiation loss to atmosphere and back plate and side insulation to reduce conduction losses. The main application of solar air heater are space heating, drying of agriculture and paint spraying operation. The main drawback of solar air heater is the low heat transfer coefficient between the absorber plate and air stream because of low heat carrying capacity of air, which results in lower thermal efficiency. Satcunanathan and Deonarine[1] states that a solar air heater operated in two pass mode give a better performance to that when operated in single pass mode under same condition and is also cost effective.The efficiency of solar air heater can be improved by two ways.(a) By the development of turbulence in air stream inside the channel to increase convective heat transfer rate between absorber plate and air.(b) By increasing the heat transfer area by using fins or corrugated surface.

In field of double pass solar air heater a number of studies have been carried out on the performance analysis with different heat transfer augmentation techniques like, (a) using extended surface (b) packed bed (c) corrugated absorber plate (d) using recycle process. But so far no experimental study has been reported on double pass solar air heater with artificially roughened absorber plate. In this study experimental evaluation is done for Nusselt number and friction factor for different parameters for double pass solar air heater (DPSAH) with artificial roughness on both side of absorber plate. So far many investigator works on DPSAH with following different flow arrangements also shown in Fig. 1 as-

i. Parellel pass double duct solar air heater.
ii. Counter flow DPSAH.
iii. Counter flow DPSAH.

El-Sebaii et al.[2] investegated the double pass finned plate solar air heater theoritically and experimentally and an analytical model for air heater was also presented. He also showed that double pass V-corrugated solar air heater is 9.3 to 11.9 % more efficient compared to double pass finned solar air heater.Paisarn Naphson[3] did a numerical study on the performance and entropy generation of double pass solar air heater with longitudinal fins. He also develop the mathematical model describimg the heat transfer characterstics for mass flow rate between 0.02 to 0.1 Kg/s. El-Sebaii et al. [4] also investigated the thermal performance of DPSAH with packed bed by both, experimentally and theoritically. Furthermore he investigated the effect of mass flow rate of air and effect of mass and porosity of packed bed material on different output like outlet temperature of air, thermal output power , pressure drop and thermohydraulic efficiency. Prashant Dhiman et al. [5] investigated a parellel flow DPSAH with packed material in its upper channel and also gave an analytical model describing the various temperature and heat transfer charactersticsand employd this model to study the effect of mass flow rate and varying porosities of packed material on its thermal performance. Ho et al. [6] found a considerable improvement in collector efficiency when operation of DPSAH was carried out with external recycle. He also found that desirable effect i.e. increased fluid velocity compensate the undesirable effect of decreased driving force (i.e. temperature difference).Ho et al. [7] also investigate theoritically and experimentally a device for inserting an absorber plate in DPSAH with recycle and compared the results graphically with downward type single pass solar air heater. The effect of absorbing plate location was also discussed. Aldabbagh[8] investigated both single and double pass solar air heater experimentally with fins and used steel wire mesh as absorber plate. He found that for same mass flow rate the efficiency of double pass solar air heater is more than single pass solar air heater. Ho et al.[9] investigated theoritically the collector efficiency with fins and external recycle of upward type double pass solar air heater. Esen et al.[10] experimentally investigates the performance of double pass solar air heater having aluminium cans. Momin et al.[11] studied the heat transfer and friction solar air heater duct with V-shaped ribs on absorber plate.

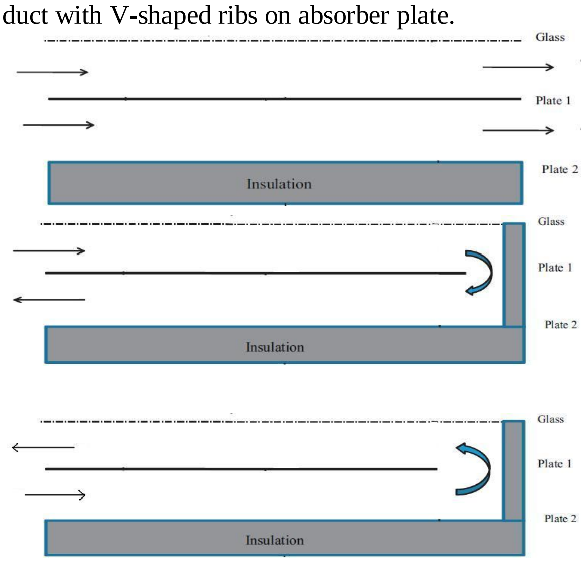

Figure 1. Different flow arrangements in DPSAH

2. ROUGHNESS GEOMETRY AND RANGE OF PARAMETERS

Aluminium wie of different diameter is used to provide artificial roughness. Fig. 2 shows the V-shape pattern in which circular ribs are gluedon both side of the surface of the absorber plate.These ribs are glued on surface in downstream i.e. in direction of flow on both side. The cross-section of artificial roughness has been described by the values of rib height (e), rib pitch (P) and angle of attack (α). These parameters have been expressed in the form of the following dimensionless roughness parameters:-

i. Relative roughness pitch (P/e).
ii. Relative roughness height (e/D_h).
iii. Angle of attack (α).

The range of parameters for this experimental study has been decided on the basis of practical considerations of the system and operating conditions of the solar air heater and is shown in Table I.

Table I. Operating parameters range

Operating parameters	Range
Reynolds Number (Re)	4900-12000
Relative roughness pitch (P/e)	10
Relative roughness height (e/D_h)	0.022-0.044
Angle of attack (α)	60°

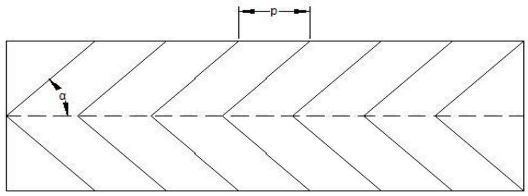

Figure. 2. Orientation of roughness geometry

3. EXPERIMENTAL PROGRAM

3.1 Experimental setup

The schmetic diagram of experimental setup and cross section of rectangular duct is shown in Fig. 3. The flow system consists of an entry section, an exit section, test section, plenum and centrifugal blower. A 250mm ×50mm rectangular duct having length of 2040mm was fabricated by wood. The length and width of test section are 1600mm and 250mm respectively and the length of entry and exit section are 400mm each on same side. A 40mm gap is provided at the end of test section for circulation of air from first pass to second pass. The aluminium circular ribs are glued to the surface of GI sheet having thickness 0.8mm and length 1600mm. The absorber plate is painted with black paint on upper side. A transparent glass of transmitivity about 0.88 is used to cover absorber plate and to reduce the convection and radiation loss. A solar simulator is also fabricated to heat the absorber plate. This solar simulator consist of halozen lamps and placed at such a height that an average intensity of 900W/m^2 falls constantly on the absorber plate. The mass flow rate of air through duct is measured by using a caliberated orifice meter(C_d=0.61) in the flow line connected with a U-tube manometer. A control valve is also provided to controll the mass of air flow through the duct. The calibrated copper-constantan (T) thermocouples have been used to measure the average air and absorber plate temperatures at different locations as shown in Fig. 4. The caliberation curve for the thermocouple has been shown in Fig. 5.The thermocouples output is fed to a digital milli-voltmeter through a selector switch and is used to indicate the output of the thermocouples in °C. The temperature measurement system is calibrated to yield temperature value within ±0.1 °C.

The pressure drops across a 1600 mm length of test section is measured by a micro-manometer.

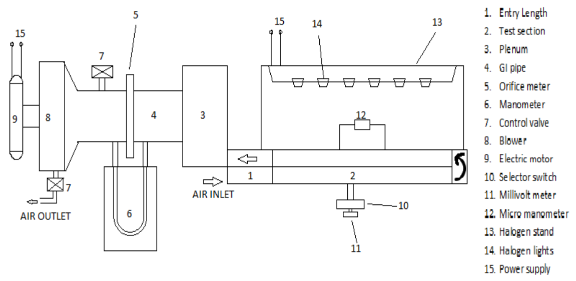

Figure 3. Schematic diagram of the set-up

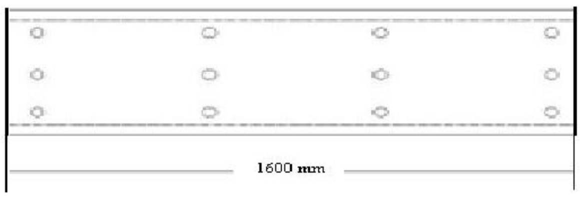

Figure 4. Location of thermocouples at the test section

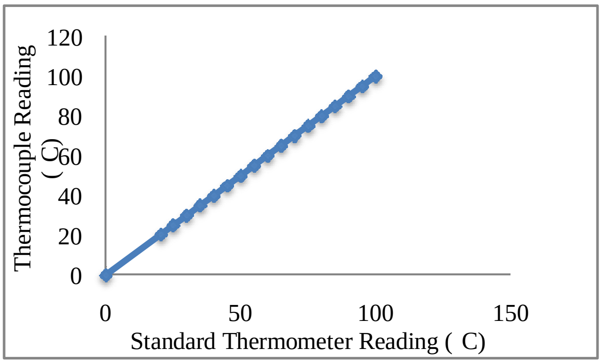

Figure 5. Calibration of Thermocouples reading with standard Thermometer reading.

3.2 *Experimental procedure*

Before starting the experiments, all joints of the duct, inlet section, and pipe fittings were carefully examined against air leakage. The test runs were conducted to collect relevant heat transfer and flow friction data under quasi steady state conditions. The system was allowed to attain the quasi steady sate condition for different mass flow rate before data were recorded. It takes about 2 hrs to attain steady state. The following parameters were measured:

i. Temperature of the absorber plate and air at inlet and outlet of the test section.
ii. Pressure drop across the test section.
iii. Pressure difference across the orifice meter.

4. DATA REDUCTION

To study the performnance of DPSAH, values of different useful parameters were required. These values are determined with the help of data obtained from experiment, like quasi steady-state value of plate and air temperature in the duct. Pressure drop across test section and orifice meter were also recorded. Average plate temperature was determined by taking average of steady state values of 12 thermocouple attached on absorber plate on different location as shown in Fig. 4. It was found that the temperature of the absorber plate increase linearly in flow directionand no variation occour in direction normal to the flow. The fluid temperature was determined as an average of the temperature at the exit and entrance location of the duct section. Mass flow rate *m*, velocity of air *v*, heat supplied to the air *q* and heat transfer coefficient *h*, were calculated by using following expressions:

$$m = C_d A_T \left[\frac{2\rho\Delta P}{1-\beta^4}\right]^{1/2} \qquad (1)$$

$$v = \frac{m}{\rho A_c} \qquad (2)$$

$$q = mC_p(t_o - t_i) \qquad (3)$$

$$h = \frac{mC_p(t_o - t_i)}{A_p(t_p - t_a)} \qquad (4)$$

$$D_h = \frac{4A_c}{p} \qquad (5)$$

where t_p and t_a are the average values of the temperature of absorber plate and fluid (air), respectively.

The average Nusselt number was obtained by the use of convective heat transfer using the following expression:

$$Nu = \frac{hD_h}{K} \qquad (6)$$

The friction factor was obtained using the value of pressure drop across the test section by the following expression:

$$f_r = \frac{2\Delta P D_h}{4\rho L v^2} \qquad (7)$$

5. VALIDITY TEST

The basic aim of validity test is to compare experimental heat transfer result through the smooth plate with the correlations available for the heat transfer and friction factor (f_s) through smooth plate. But these correlation available, are for single pass solar air heater with smooth plate so for double pass, these correlations are multiplied with '2' and found to be in good agreement. The Nusselt number (Nu_s) and friction factor determined from these experimental data are compared with the values obtained from the correlations i.e. Dittus–Boelter correlation[10] and the Modified Blasius equation[11]. The comparision is shown in Fig. 6 (a) and (b).

Dittus-Boelter equation

$$Nu_s = 2 \times 0.024 \, \mathrm{Re}^{0.8} \, \mathrm{Pr}^{0.4} \tag{8}$$

Modified Blasius equation :

$$f_s = 2 \times 0.085 \, \mathrm{Re}^{-0.25} \tag{9}$$

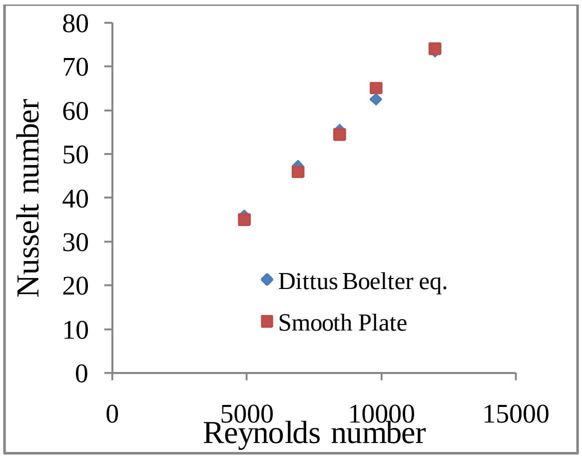

Figure 6.(a) Comparision of experimental and predicted values of Nusselt Number for smooth plate.

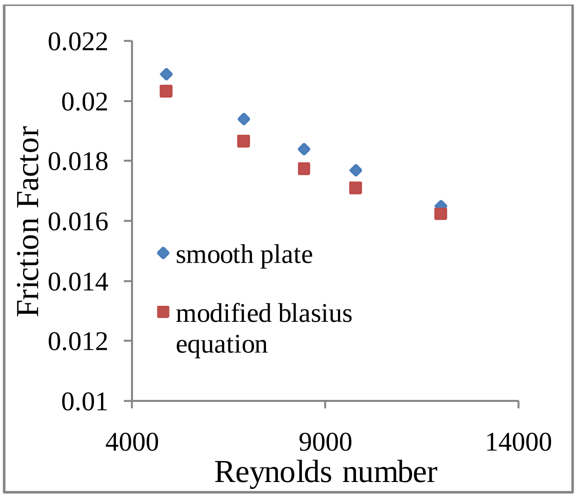

Figure.6.(b) Comparision of experimental and predicted values of friction factor for smooth plate

6. RESULTS AND DISCUSSION

In this section of paper the effect of relative roughness height (e/D_h) at fixed relative roughness pitch (p/e) on heat transfer and friction factor in double pass solar air heater. Fig. 7(a) and 7(b) shows that Nusselt number increases with increase in Reynolds number and decrease in friction factor with increase in Reynolds number. The Fig. 7(a) shows the relation between Nusselt number and reynolds number as a function of e/D_h it also shows the increament in Nusselt number with increase in value of relative roughness height from 0.022 to 0.044 at fixed p/e as 10, because the laminar sublayer breaks as relative roughness height increases and heat transfer enhanced. Similarly Fig. 7(b) shows that value of friction factor decreases as value of Reynolds number increases and increses as value of relative roughness height increases from 0.022 to 0.044 at fixed value of relative roughness pitch as 10.

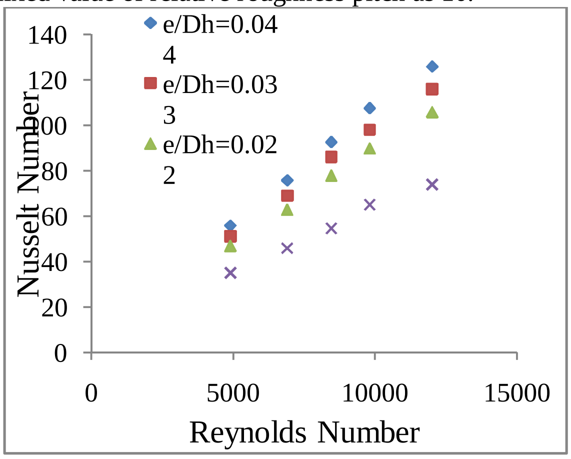

Figure. 7 (a) Enhancement in the value of Nusselt number for different *e/D_h* and for fixed *p/e=10*.

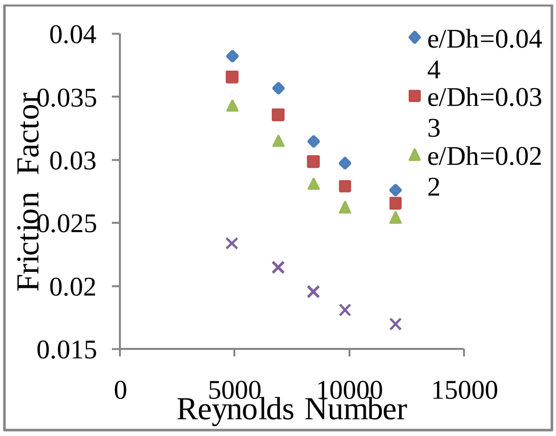

Figure 7(b) Enhancement in the value of friction factor for different *e/D_h* and for fixed *p/e=10*.

7. CONCLUSION

An experimental study has been performed on a arificially roughened double pass solar air heater at various Reynolds number subjected to uniform heat flux by a solar simulator. The effect of relative roughness height (e/D_h) has been studied on the heat transfer and friction factor. This study shows that the maximum heat transfer and friction factor occour at the relative roughness height of 0.044.

NOMENCLATURE

A_C Area of the flow (m^2)

$A_T A_T$ Throat area of the orifice (m^2)
$A_P A_p$ Area of the absorber plate (m^2)
C_d Coefficient of discharge for the orifice meter
$C_p C_p$ Specific heat of air (kJ/kg/K)
D_h Hydraulic diameter of the duct (m)
e Height of the roughness element (m)
f_r Friction factor for roughened absorber plates
f_s Friction factor for the smooth absorber plate
H Height of the duct (m)
h Average heat transfer coefficient ($W/m^2/K$)
L Length of the absorber plate (m)
m Mass flow rate (kg/s)
Nu Nusselt number for the roughened plates
Nu_s Nusselt number for the smooth plates
P Roughness pitch (m)
Pr Prandtl number
ΔP Pressure drop across the orifice meter (N/m^2)
Δp Pressure drop across the test section (N/m^2)
Re Reynolds number
W Width of the duct (m)
D_1 Diameter of orifice (m)
D_2 Diameter of pipe (m)
t_i Inlet temperature of air (°C)
t_o Outlet temperature of air (°C)
t_a Average temperature of air (°C)
t_p Average temperature of the absorbing plate (°C)
ρ Density of fluid (kg/m^3)
β Diameter ratio, D_2/D_1
α Angle of attack
e/D Relative roughness height
P/e Relative roughness pitch
P Perimeter of flow section

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