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Enhancement of Heat Transfer in Solar Air Heater with Parallelogrm Protrusions as Roughness Elements

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Abstract:** This paper presents the results of an experimental investigation of heat transfer rate in solar air heater duct having parallelogram protrusions as artificial roughness elements with different apex angles (30^0, 45^0 and 60^0). The parallelogram protrusions of base side 10mm, thickness 3mm and with a pitch 25mm are made as zig-zag pattern throughout the length of the plate. The range of parameters for this study has been decided on the basis of practical considerations of the system and operating condition of the solar air heater. The results shows that the ducts with parallelogram protrusions have higher heat transfer rate than the smooth plate, with Reynolds number ranging from 10000-12500 , nusselt number from 50-112 and friction factor is from $4.5\text{-}6.7*10^{-3}$. Among the protrusions heat transfer rate is high for 45^0plate.

***Keywords:** solar air heater, heat transfer, Reynolds number, Nusselt number, Friction factor.*

I. INTRODUCTION

In all our daily activities energy is required for transportation, industrial, domestic, agricultural purposes etc… in different forms (heating, electricity etc..). For all these purposes major extent of energy comes from conventional sources like coal , petroleum and natural gas. As a result of their limited availability we need to replace them with renewable energy sources like solar, wind, hydel energy etc…. Among these renewable sources utilization of solar energy is very easy and if we even collects 1% of solar radiation that comes from the earth and convert it into different forms like heat or electricity it will be sufficient for the whole world to survive for one year. The most simplest and economical way to utilize solar energy is to convert it into thermal energy for heating applications by using solar collectors. Solar air heaters because of their inherent simplicity are cheap and most widely used solar collectors. Due to low convective heat transfer co-efficient between the air and the absorber plate solar air heaters have low efficiency which leads to higher temperature of the absorber plate leading to high heat losses to the environment. Hence in order to increase the heat transfer co-efficient of solar air heater various methods have been proposed. One of the methods is creating the turbulence on the absorber plate in the form of artificial roughness. Providing the roughness over the existing surface is easy way to disturb the flow and also increase the heat transfer rate from surface to the air flowing through the solar air heater duct. For many years researchers have studied the enhancement of heat transfer co-efficinets of solar air heaters by having artificial roughness on the air flow side of the absorber plate in various ways to improve the thermal efficiency of solar air heater (Karwa et al.,1999; Muluwork et al.,2000; Prasad et al.,1988; Bhagoria et al.,2002; Momin et al.,2002; Karwa et al.,2003) by applying different types of rib roughness. The present investigation is taken up with the of experimentation on parallelogram protrusions as artificial roughness to the underside of one broad wall of the duct to evaluate enhancement of heat transfer co-efficient of solar air collector subjected to uniform heat flux. The apex angle of the protrusions on plates is varied to find out how they affect heat transfer rate.

II. NOMENCLATURE

A_p	Absorber plate area, m^2
A_0	Cross-Section area of the orifice, m^2
AR	Aspect ratio of duct
A_1	Pipe Diameter
C_d	Coefficient of discharge of orifice
C_{pair}	Specific heat of air at constant

D_h	Hydraulic diameter of the duct, m
f	friction factor of roughened duct
f_s	friction factor of smooth duct
G_{air}	Mass flow velocity of air, kg/s m^2
L_f	Test Section length for pressure drop
m	mass flow rate of air, Kg/s
Nu	Nusselt number
Nu_R	Nusselt number of roughened duct
Nu_s	Nusselt number of smooth duct
P/e	Relative roughness pitch
Pr	Prandtl number
Re	Reynolds number
St	Stanton number
T_i	Inlet air temperature, ^oC
T_0	Outlet air temperature, ^oC
ΔP	Pressure drop across orifice plate
δp	Pressure drop across test-section
P/e	Relative roughness pitch
e/D_h	Relative roughness height

III. EXPERIMENTAL SETUP

The experimental setup was designed in order to carry out experimental investigation on heat transfer of smooth as well as roughened duct used in solar air heaters. The setup is an indoor open flow loop that mainly consists of a long duct having dimensions 2400x150x15 mm^3which is divided into three sections namely

A. Entry section (800 mm length)
B. Exit section (600 mm length)
C. Test section (1000 mm length)

The recommended minimum entry and exit lengths are $5\sqrt{WH}$ and $2.5\sqrt{WH}$. The flow can be assumed to be fully developed in the entire test section. The top side of the test section consists roughened aluminum absorber plate. The other dimensions of design are determined by initially fixing the length of the test section. An aspect ratio of 8-15 is observed to be optimum for maximum heat transfer rate of solar air heater as per previous researchers. So the aspect ratio of the duct is chosen to be 10.The electric heater is made by fixing a Nichrome of wattage 1000 W/m^2in zigzag pattern across the length of asbestos sheet.28-SWG copper-constantan thermocouples are fixed across the length of the absorber plate as well as in the duct to measure the temperature of plate across is length and air flowing through the duct respectively. The baffles are provided to mix the hot air coming out from the duct to obtain a uniform temperature of air at the outlet. A transition has been made that connects the wooden duct with the G.I. pipe to which centrifugal blower and bypass valve are attached. Another transition is used to join the other end of G.I. pipe to the centrifugal blower. The centrifugal blower is coupled with a single phase induction motor which runs at 2800 rpm. The blower creates vacuum in the duct to suck the atmospheric air and flow through it. A bypass valve is connected across the length of the G.I. pipe to change the mass flow rate of air in the duct. The mass flow rate of air is measured by means of an orifice meter connected with a U-tube manometer and the flow is controlled by the control valve. The entire duct is insulated by using thermocol. The schematic diagram is show in below.

Fig.1 Actual setup of experiment

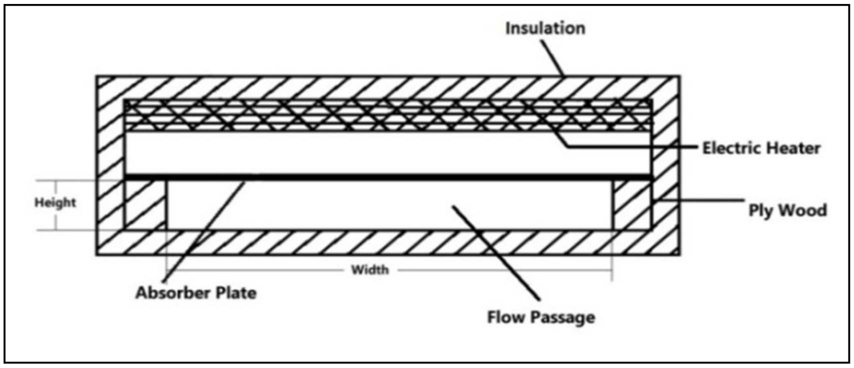

Fig.2 Cross -sectional view of the solar air heater

A. Absorber plates

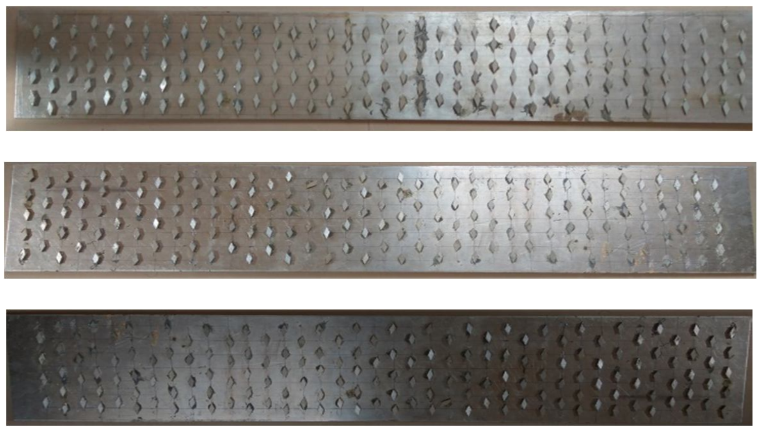

Fig.3 Rough plates of different apex angles

Absorber plate is made up of aluminum plate of 6 mm thickness. Roughness is created by parallelogram protrusions shape of 3mm rib height on the absorber plate. Different plates with parallelogram protrusions having different apex angles (30^0,45^0 and 60^0) and the base side of the protrusions (10mm) are same for all plates

IV. EXPERIMENTATION

Before starting the setup is checked for proper operation. The blower is then switched on and joints have been checked for leakage. The absorber plate is placed on the test section and the heater is placed above the absorber plate. The duct is then fully covered with thick glass wool insulation. The electric heater is connected to the variac. Copper-constantan thermocouples of gauge (28SWG) are used to measure the temperature of absorber plate and the air. Flow of air is controlled with the help of control valve. Initially the control valve is closed and switch on the heater. The plate is allowed to get heated and the blower is switch on the temperature of the plate and air are maintained to assume steady state. Under steady state conditions the test runs to collect relevant heat transfer data were conducted. Later the absorber plate is replaced by another absorber plate with protrusions of different configurations and the experimentation is repeated.

V. DATA REDUCTION

A. Average Temperature

The mean air temperature of Average flow temperature T_f is the measure values at the inlet and outlet of the test section. Thus,
$T_f = (Ti+T_{oav})/2$

The mean plate temperature $T_{p_{av}}$ is the average of the reading of all points located on the absorber plate.

B. Mass Flow Measurement

Mass flow rate is calculated using the equation.

$$m = C_d \times A_0 \times \left[2\rho\Delta P_0 / \left(1-\beta^4\right)\right]^{0.5}$$

Where,

M, Mass flow rate, kg/s

C_d, Coefficient of discharge orifice

A_0, Area of orifice in m^2

ρ, Density of air kg/m3

β, Ratio of diameter of orifice and pipe

ΔP_0, Pressure difference

C. Velocity Measurement

$$V = \frac{m}{\rho\ W\ H}$$

Where,

ρ= density of air, Kg/m^3

W = width of duct, m

H = height of duct, m

D. Reynolds Number

$$\mathrm{Re} = \frac{VD_h}{v}$$

Where,

D_h = Hydraulic diameter $= \dfrac{4WH}{2(W+H)}$

v = kinematic viscosity, m^2/s

E. Heat Transfer Coefficient:

Value of heat transfer coefficient between the absorber plate and fluid is given by the equation.

$mC_p\ [t_0 - t_i] = h\ A_p\ [t_p - t_f]$

Where,

C_p= specific heat of air, J/Kg K

A_p = area of absorber plate, m^2

F. Nusselt Number

$$Nu = \frac{hD_h}{K}$$

Where,

K = thermal conductivity of air, W/m K.

V.RESULTS AND DISCUSSION

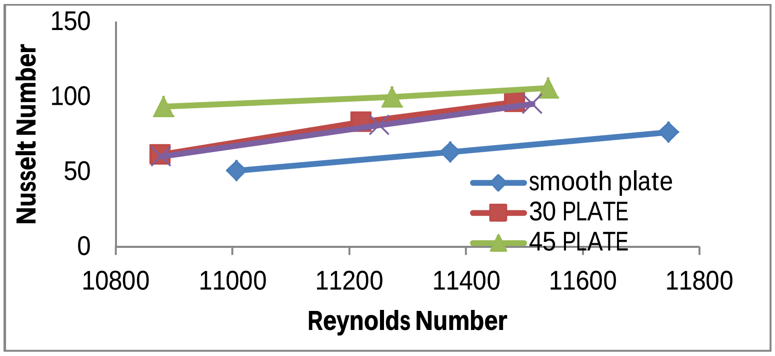

Fig.4 Nusselt number Vs Reynolds number

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The graph shows the variation of Nusselt number with respect to Reynolds number for smooth plate and for the plates of different apex angle (30^0,$45^0$and 60^0) protrusions. As the Nusselt number increases Reynolds number also increases. For 45^0 apex angle the Nusselt number is high and it is less for smooth plate.

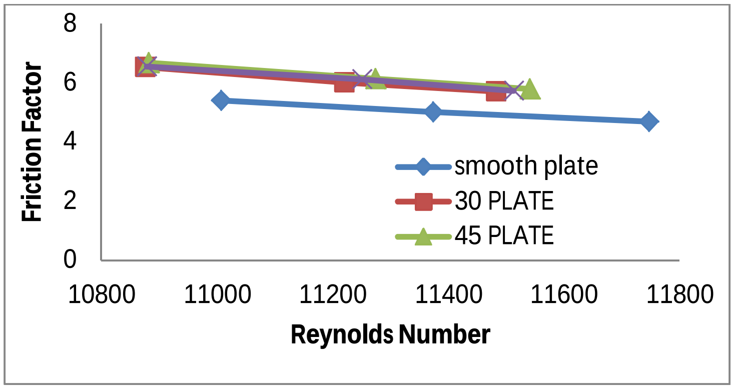

Fig.5Friction factor Vs Reynolds number

The graph was made between friction factor and Reynolds number for smooth plate and for plates having 30^0, $45^0$and 60^0 apex angle protrusions. From this it is clear that, the friction factor is highest at lower Reynolds number and it is gradually decreases with increase of Reynolds number.

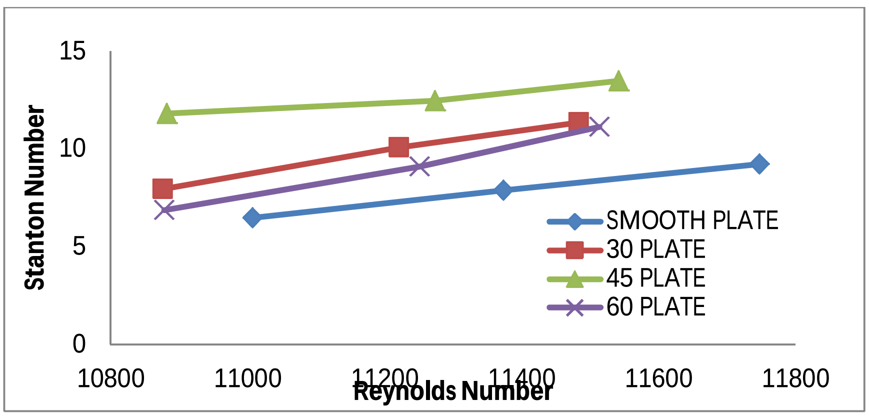

Fig.6 Stanton number Vs. Reynolds number

By taking Reynolds number on x-axis and Stanton number on y-axis, a graph is plotted for the available varieties of the absorber plates.

The graph shows that the Stanton number value rise with a rise in Reynolds number and also it is observed that the Stanton number is highest for the absorber plate with 45^0 apex angle protrusions while least for the smooth absorber plate.

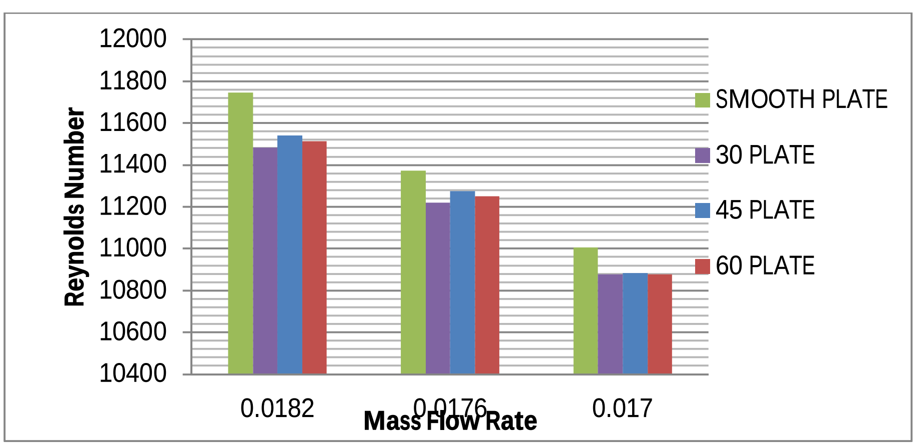

Fig.7Reynolds number VsMass flow rate

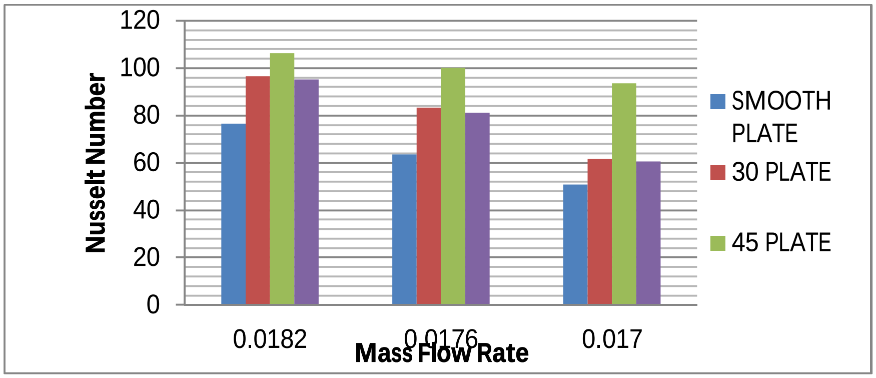

Fig.4 Nusselt number Vs Mass Flow rate

Owing to this the mass flow rate of the air increases with increase in Reynolds number and Nusselt Number and the convective heat transfer rate increase with increase in mass flow rate as observed from the previous graphs. This is because the quantity of the air moving through the duct in unit time, the air sub laminar layers are disturbed by the protrusions and thus all the air layers are mixed and the heat is uniformly distributed to all the air layers. But if the air is moving faster as the quantity of air moving through the duct is high and the heat transfer occurs rapidly. As Stanton number is a function of convective heat transfer to the thermal capacity of the air from the above discussion one can conclude that Stanton number rises with the rise in Reynolds number.

VI.CONCLUSION

The heat transfer rate would be maximum for the plate having an apex angle of protrusions 45^0. So that it is concluded that 45^0 plates has given maximum amount of heat transfer which is higher than that of smooth plate by 42%.The smooth plate having very least heat transfer rate and it Nusselt number value is 50.92.The Stanton number is highest for the absorber plate with 45^0 apex angle protrusions while least for the smooth absorber plate.

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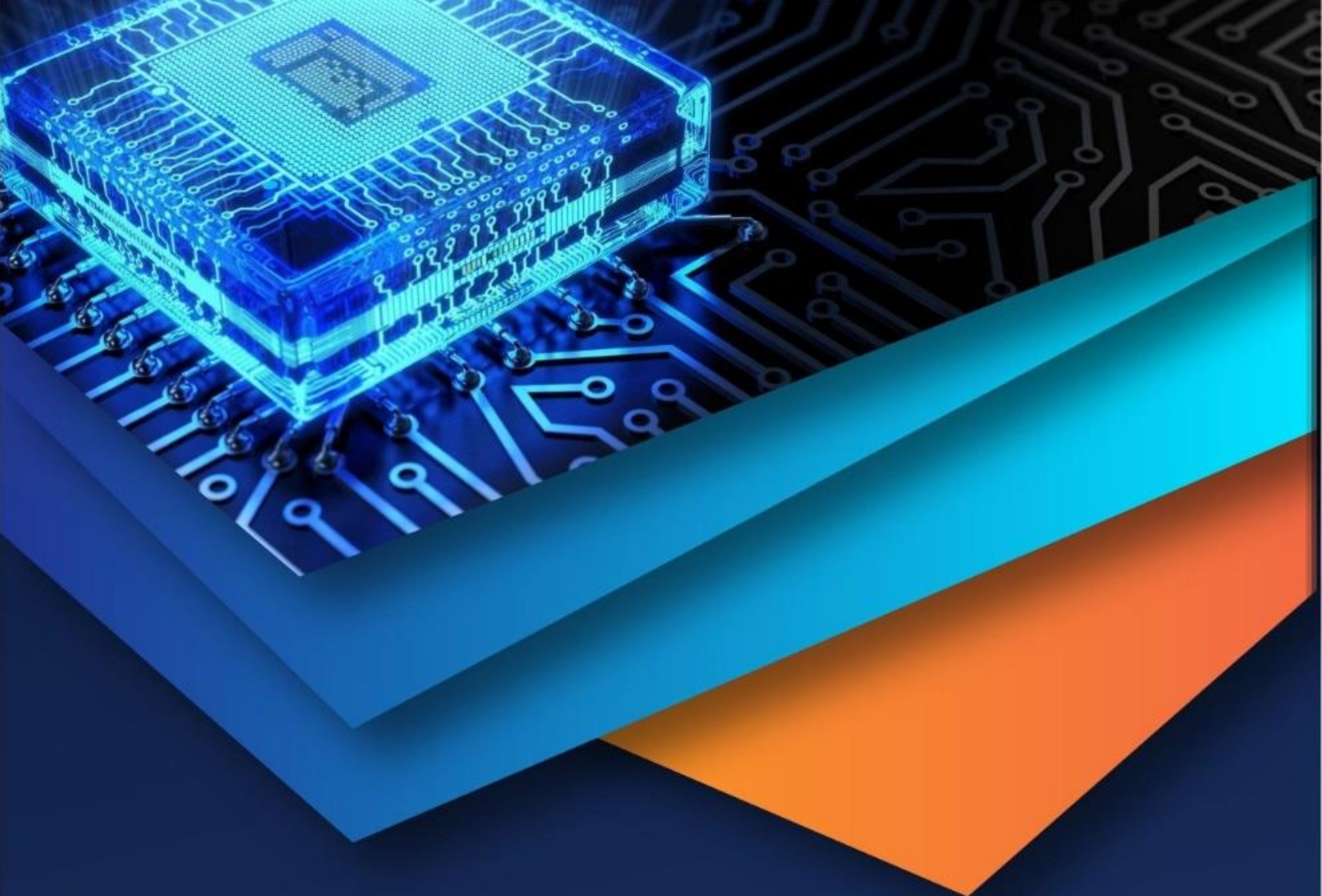

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