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Performance Assessment of Double Pass Solar Air Heater with Inclined Discrete Ribs on Absorber Plate

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Abstract: - *In double pass solar air heater thermal performance can be obtained by augment the rate of heat transfer and minimizes the top losses and bottom losses.Efficiency of solar air heater is low because of low value of heat transfer coefficient between absorber plate and flowing air. This is due to the presence of laminar sub layer that need to be broken to increase heat transfer. Therefore for enhancement of heat transfer turbulence is created by providing inclined discrete ribs on absorber plate. This paper presents an experimental investigation carried out to study the effect Inclined continues rib roughness on heat transfer coefficient and friction factor in an artificially roughened solar air heater duct. The experiment encompassed Reynolds number (Re) from 3000 to 15,000. Relative roughness height (e/D) values 0.027, relative roughness pitch (P/e) range of 6–12, angle of attack (α) - 60^0.Extensive experimentation has been conducted to collect data on heat transfer and fluid flow characteristics of a rectangular duct roughened with inclined discrete ribs. Using these experimental data, correlations for Nusselt number and friction factor in terms of roughness geometry and flow parameters have been developed. The roughened wall has roughness with pitch (P), ranging from 9-18 mm, height of the rib of 1.5 mm and duct aspect ratio of 6.67. The air flow rate corresponds to Reynolds number between 3000–15000. The heat transfer results have been compared with those for smooth ducts under similar flow and thermal boundary condition to determine the thermal efficiency of solar air heater.*

Keywords: Double pass solar air heater, inclined discrete ribs, Reynolds number, Nusselt number, Friction factor.

Introduction

Solar air heater occupies an important place among solar heating systems because of nominal use of materials. Additionally, direct use of air as the working substance reduces the number of components required in the system.The air to be heated is passed through a rectangular cross-section duct below a metal absorber plate with the sun-facing side blackened to facilitate absorption of solar radiation incident on the absorber plate. Transparent covers are placed over the absorber plate to reduce the thermal losses from the heated absorber plate. The thermal efficiency of a solar air heater is significantly low because of the low value of the convective heat transfer coefficient between the absorber plate and the air, leading to high absorber plate temperature and greater heat losses to the surroundings. It has been found that the main thermal resistance to the heat transfer is due to the formation of a laminar sub layer on the absorber plate heat-transferring surface [1]. Efforts for improving the heat transfer rate have been directed towards artificially destroying or disturbing this sub-layer. An artificial roughness on the heat transfer surface in the form of projections mainly creates turbulence near the wall or breaks the laminar sub-layer and thus enhances the heat transfer coefficient with a minimum pressure loss penalty. Artificial roughness in the form of wires [2] or ribs of various geometrical shapes have been employed for the enhancement of the heat transfer coefficient in heat exchangers Several investigations have been carried out to study the effect of artificial roughness on heat transfer and friction factor for two opposite roughened surfaces by Han et al. [3-4], Lau et al. [5], Liou and Hwang [6], Han and Park [7] and the correlations were developed by different investigators. Prasad and Mullick [8], Gupta [9], Saini and Saini [10] and Karwa [11] have carried out investigations on rib roughened absorber plates of solar air heaters that form a system with only one roughened wall and three smooth walls.From the experimental study of the effect of artificial roughness, various investigators have shown that the geometry of the roughness (Roughness shape, pitch, height, etc.) Has a marked influence on the heat transfer and friction characteristics of the surface. The heat transfer coefficient enhancement is also accompanied with an enhancement in the friction factor. Thus, an appropriate way to evaluate performance, in the case of solar air heaters with a roughened absorber plate, is to take both heat collection rate and pumping power requirement into account, i.e. to carry out a thermo-hydraulic performance evaluation. The object of the present investigation was to study, both theoretically and experimentally, the effect of Continues Inclined rib-roughness on the absorber plate transverse to the flow of air as shown in Fig. 1(a) on the thermo-hydraulic performance of solar air heaters. A direct experimental comparison of the performance of the roughened solar air heater has been made with the performance of a solar air heater with a smooth absorber plate and also with single pass solar air heater Performance. This has been accomplished by using two identical passes ducts, one with the two glass cover pass (air flow in between them and pre-heated) and the other with a roughened absorber plate (top) and smooth one (bottom).

Experimental program and roughness geometry

It is necessary to design, develop and fabricate the solar air heater duct with using artificially roughened absorber plate for obtaining the heat transfer performance. An experimental setup has been designed and fabricated [l] to18mm thick wooden ply with 12mm thick insulation around rectangular outlet. The enhancement of heat transfer having continuous inclined shape ribs made from G. I. wires of 16 gauges, on the G.I absorber Rectangular plate of 22 gauges. The flow system consist of an entry section (400mm×200mm), first pass section (between two transparent 5mm thick glass covers(1500mm×200mm), test section(1500mm×200mm), exit section(200mm×200mm),and flow measuring orifice plate(25mm) and a centrifugal blower (3-phase, 440 V, 3.0 kW and 2880 r.p.m.AC motor) with a control valve. The air enters at entry section and first pass section where the atmospheric air slightly pre heated by trapped heat between two transparent glass cover and then enter in to test section where that preheated air gets heated by taking heat from absorber G.I. plate of 22 ASWG of (1500mm×200mm) which gets it heat from electric heater of uniform heat flux up to a maximum of 1059 W/m^2. Eight Calibrated thermocouples (K-type) are attached with upper surface of the absorber plate for measurement of plate average temperature. Two thermocouple is connected at entry section to find inlet temperature and two is used for measuring exit temperature of the air. A schematic diagram of the experimental set-up and view of the roughness plate is shown in Fig. 1(a-b-c-d) respectively. Pressure drop across the orifice meter was measured by an inclined U-tube manometer with water as manometric fluid. A digital Calibrated temperature indicator with Copper Constantan (28 ASWG) thermocouples have been used for the measurements of temperatures.

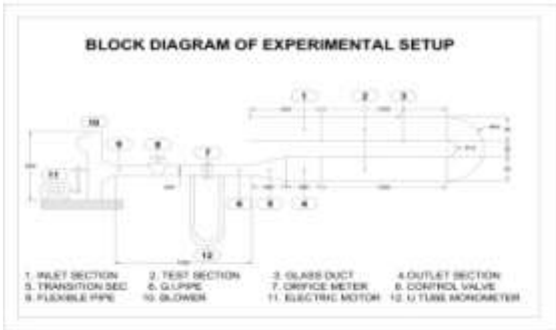

Fig. 1(a) Front view

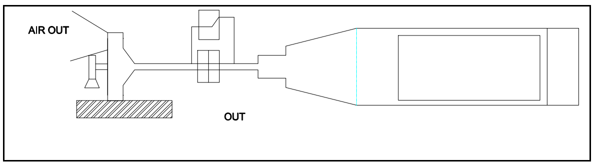

Fig. 1(b) Top view

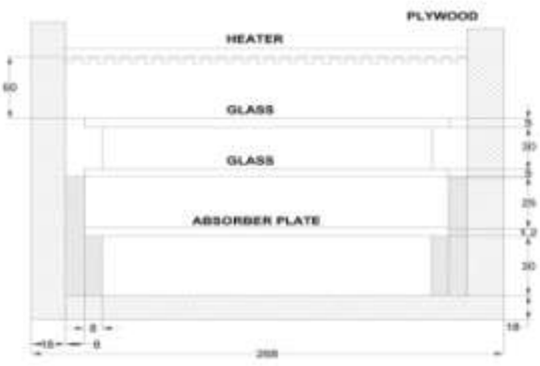

Fig. 1(c) Section at X-X

Absorber plate

The roughness parameters are determined by rib height (e), rib pitch (p),angle of attack (α) These parameters have been expressed in the form of the following dimensionless roughness parameters:-

(1)

Velocity measurement-

$$V = \frac{\dot{m}}{\rho \times w \times h} \tag{2}$$

Hydraulic diameter -

$$D_h = \frac{4WH}{2(W+H)} \tag{3}$$

$$\dot{m} = cd\, A_o \sqrt{\frac{2\rho(\delta P)}{1-\beta^4}}$$

Reynolds number -

$$Re = \frac{\rho V D_h}{\mu}$$

(4)

Heat transfer coefficient for the test section is -

$$h = \frac{Q_{air}}{Ap\,(T_p - T_f)} \tag{5}$$

$$Q_{air} = \dot{m} c_p (T_o - T_i) \tag{6}$$

The Nusselt number as -

$$N_u = \frac{h D_h}{k} \tag{7}$$

Thermal efficiency -

(a) Reynolds number, (Re) 3000–15000
(b) Relative roughness height, (e/D_h) 0.028
(c) Relative roughness pitch, (p/e) 6–12
(d) Angle of attack (α)-60^0

Data reduction

Steady state values of plate and air temperature have been obtained for the given heat flux and mass flow rate of air. Mass flow rate of air is-

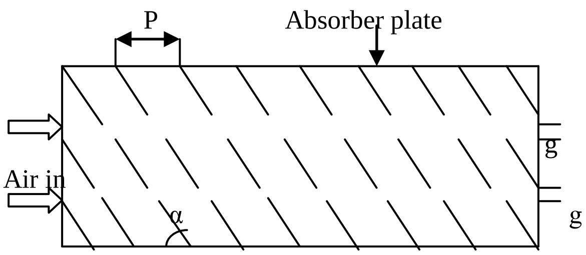

Fig. 1(d) Absorber plate

$$ή = \frac{G\, Cp(T_o - T_i)}{I} \tag{8}$$

$$G = \frac{\dot{m}}{A_p}$$

Where,

$\dot{m}$	= mass flow rate	(kg/s)
C_d	= coefficient of discharge	(m^3/s)
A_o	= area of orifice	(m^2)
ρ	= density of air	(kg/m^3)
β	= beta factor, ratio of diameter of orifice to diameter of pipe (dimensionless)	
W	= width of the duct	(m)
H	= height of the duct	(m)
V	= velocity of air	(m/s)
D_h	= hydraulic diameter	(m)
μ	= viscosity of air	(m^2/s)
Q_{air}	= heat input to air	(KJ)
C_p	= specific heat of air	(KJ/kg-K)

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T_p = ave. temperature of plate (^0C)

T_f = ave. temperature of fluid (^0C)

T_o = temperature at exit (^0C)

T_i = temperature at entry (^0C)

G = mass velocity ($kg/s\text{-}m^2$)

A_p = area of the plate (m^2)

I = heat flux (W/m^2)

h = heat transfer coefficient (W/m^2K)

K = thermal conductivity of air (W/mK)

Thermo-hydraulic performance

Performance of the suggested counter-flow solar air heater is studied and compared with the performance of single and double pass conventional solar air heaters.

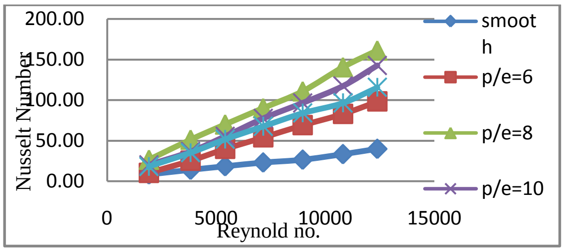

Fig. 2(a) Variation of Nusselt Number with Reynolds Number

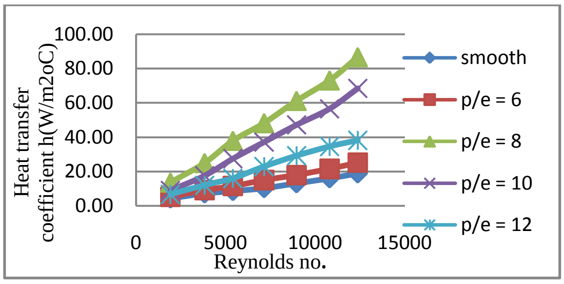

Fig. 2(b).Variation of Heat transfer coefficient with Reynolds number

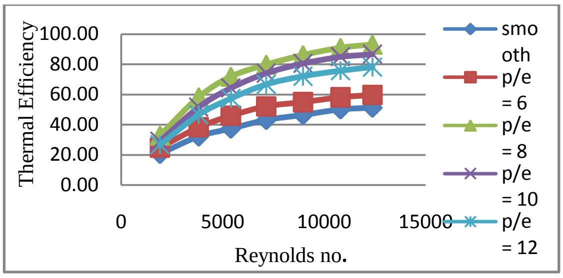

Fig. 2(c). Variation of Thermal Efficiency with Reynolds number

Validity Test

The Nusselt number and friction factor determined from experimental data for smooth duct have been compared with the values obtained from Dittus Boelter Equation and Modified Blasius Equation. Shown in fig.4

$$Nu = 0.023Re^{0.8}\,Pr^{0.4}$$

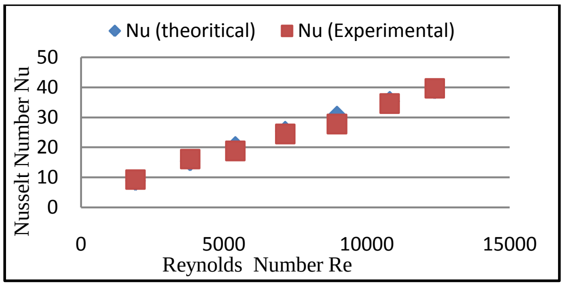

Fig. 2(d).Nusselt Number vs Reynolds Number with for Smooth Duct

Conclusions

Detailed Experiments were conducted to study the performance of the double pass solar air heater with continuous inclined ribs.

Experiments were conducted on rectangular duct having one broad wall roughened with continuous inclined ribs and subjected to constant heat flux. Heat and flow characteristics were determined. Based on the results obtained, the following conclusions are drawn:-

(i) Nusselt number increases and friction factor decreases with increases of Reynolds number. Roughness causes flow separation, reattachment and generation of secondary flow, so performance higher as compare to smooth absorber plate.

(ii) The maximum enhancement of Nusselt number for inclined continuous roughened duct in comparison to that for smooth duct has been found to be in range of (2.5-3.0) eon, for an angle of attack 60^0 the investigated range of parameters.

(iii) The maximum value of heat transfer coefficient (h) occur at (P/e) of 8.0, and these decrease on the both sides of this pitch. Heat transfer coefficient increases 3.0-4.0 times of smooth plate.

(iv) Thermo-hydraulic performance improves with angle of attack 60^0 of flow and relative roughness height and maximum.

(v) The maximum value of thermal efficiency occurs at Reynolds number range of 13000-14000, for (P/e) of 8.0 is 95.12%.

Nomenclature

A_p	surface area of absorber plate, m^2
C_p	specific heat of air, J/kg K
D_h	hydraulic diameter of duct, m
e	rib height, m
h	heat transfer coefficient, W/m^2 K
H	depth of air duct, m
I	intensity of solar radiation, W/m^2
K	thermal conductivity of air, W/mK
L	length of test section of duct or long way length of plate, m
m	mass flow rate, kg/s
P	pitch, m
D_P	pressure drop, Pa
q_u	useful heat flux, W/m^2
Q_u	useful heat gain, W
Q_l	heat loss from collector, W
Q_t	heat loss from top of collector, W
T_o	fluid outlet temperature, K
T_i	fluid inlet temperature, K
T_a	ambient temperature, K
T_{pm}	mean plate temperature, K
T_{pm}	mean air temperature, K
v	velocity of air in the duct, m/s
W	width of duct, m

Dimensionless parameters

(p/e)	Relative roughness pitch
e^+	roughness Reynolds number
e/D_h	relative roughness height
e/H	rib to channel height ratio
f_r	average friction factor
F_R	heat removal factor
G	momentum heat transfer function
Nu	Nusselt number
N_{us}	Nusselt number for smooth channel
N_{ur}	Nusselt number for rough channel
N_{uav}	area-averaged Nusselt number
N_{uo}	Nusselt number for fully developed flow smooth Channel
p/e	relative roughness pitch
Pr	Prandtl number
R	roughness function
Re	Reynolds number
W/H	duct aspect ratio

Greek symbols

η_{th}	thermal efficiency
μ	dynamic viscosity, Ns/m^2
ρ	density of air, kg/m^3
α	angle of attack, degree
β	beta factor,

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