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Performance evaluations of two pass solar air heater using 60^0 inclinedv-shaped ribs on absorber plate

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ABSTRACT : *Heat transfer enhancement in two pass solar air heater with V- shaped rib, have been investigated. Ribs were attached on absorber plate, having angle of attack (α= 60 ^{0}). The ratio of rib height to hydraulic diameter (e/D) was 0.0287, while the range of rib pitch-to-height ratio was (P/e = 6-12). Experiments were carried out using air as the convective fluid, with the Reynolds number rang (Re = 3000 -15000). Air enters the upper channel of the air heater and subsequentlyflows to the lower channel in the opposite direction. Roughened wall of the duct is uniformly heated with constant heat flux electric heater while the remaining three walls are insulated. The heat transfer results have been compared with those for smooth ducts under similar flow and thermal boundary condition. Enhancement of heat transfercoefficient and thermal efficiency by providing 60^0 inclined V–shaped roughnesses on absorber plate is 3.96and 1.82 times respectively as compere to smooth plate.*

KEY WORD: *Enhancement of heat transfer, Two- Pass solar air heater, Thermal efficiency, Top loss, Mass flow rate,Reynolds number, angle of attack.*

I. INTRODUCTION

Solar energy is a very large, inexhaustible source of energy. The power from the sun intercepted by the earth surface is approximately 1.8×1011 MW, which are many thousands of timeslarger than the present consumption rate on the earth of all commercial energy sources. Thus, in principle solar energy could supply all the present and future energy needs of the world on a continuing basis. The main difficulty is how to capture more and more energy from the available source and utilize for some useful work. Solar air heaters is the major component of solarenergy utilization system which absorbs the incoming solar radiation, converting it into thermal energy at the absorbing surface, and transferring the energy to a fluid flowing through the collector. Solar air heaters because of their inherent simplicity are cheap and most widely used in solar energy collection devices. The main applications of solar air heaters are space heating, drying of agricultural products and paint spraying operations. The efficiency of flat plate solar air heater has been found to be low because of low convective heat transfer coefficient between absorber plate and the flowing air. Several methods, including the use of fins, artificial roughness and packed beds in the ducts, have been proposed for the enhancement ofthermal performance. Use of artificial roughness in the form of repeated ribs of various shapes and orientations has been found to be a convenientmethod to investigate the performance of such systems.

The use of artificial roughness in solarair heaters owes its origin to several investigations carried out in connection with the enhancement of heat transfer in nuclear reactors and turbine blades. Several investigations have been carried out by different investigators as Prasad and Mullick [1],used artificial roughness in the form of fine wires in a solar air heater duct to improve the thermal performance of collector and they have obtained the enhancement (ratio of the values for roughened duct to that for the smooth duct) in Nusselt number of the order of 1.385. Gupta [2], found that the heat transfer coefficient of roughened duct using wires as artificial roughness can be improved by a factor up to 1.8 and the friction factor has beenfound to increase by a factor up to 2.7 times of smooth duct.Saini and Saini [3] reported that a maximum enhancement in Nusselt number and friction factor for a duct roughened with expanded metal mesh is of the order of 4 and 5 respectively in the range of parameters investigated.Karwa [4]Investigated the effect on heat transfer and friction factor by using the Transverse, inclined, v-continuous and v-discrete rib on absorber plate in solar air heater, with Reynolds number range of 2800-15000, relative roughness height (e/D) range of 0.0467 - 0.050 duct aspect ratio range of 7.19- 7.75 at the fixed value of relative roughness pitch (p/e) of 10 and also developed the correlation for Stanton number and friction factor. The rib in the v-pattern were tested for both pointing upstream (V-up) and (V-down) to the flow. Enhancement in Stanton number and friction factor over that of the smooth duct was observed of the order of 65 -90% and 2.68 - 2.94 times, respectively. It is observed that 60^0 inclined rectangular ribs produces better results than transvers rib.

The enhancement in the Stanton number over the smooth duct was up to 137%, 147%, 134% and 142% for the V-up continuous, V-down continuous, V-up discrete and V-down discrete rib arrangement respectively. The friction factor ratio for these arrangements was up to 3.92, 3.65, 2.47 and 2.58 respectively. Based on the equal pumping power, V-down discrete roughness provides the best heat transfer performance. However, the increase in heat transfer is accompanied by an increase in the resistance of fluid flow.Prasad and Saini [5] reported that a maximum enhancement in Nusselt number and friction factor which are 2.38 and 4.25 times of smooth duct has been obtained by using artificialroughness. Although the heat transfer problems can be investigated by analytical means too, but due to the complex nature of governing equations and the difficulty in obtaining analytical/numerical solutions, the researchershave focused greater attention on the experimental investigation. From the literature survey it was found that the Nusselt number on the ribbed side wall having transverse ribs is about two or three times higher than the four sided smooth channel values [6]. Han et al. [7] reported that ribs inclined at an angle of attack of 45° were found to have superior heat transfer performance when compared to transverse ribs. Han et al. [8], Lau et al. [9] and Taslim et al. [10] carried out investigation on rib roughened walls having V-shaped ribs and have reported that V-shaped ribs result in better enhancement compared to inclined ribs and transverse ribs. Lau et al. [9] reported that for the range of Reynolds numbers studied, the values of Stanton number in the 45°and 60°V-shaped ribs cases are 38–46% and 47–66% higher than those in the 90° full ribs case respectively and the pressure drop in 45° and 60° V-shaped ribs is 55–72% and 68–79% over the 90°full ribs respectively. They have also reported that the 60°V-shaped ribs have the higher ribbed wall heat transfer and thermal performance. Taslim et al. [10] have shown that the enhancement in heat transfer coefficient for air flow in a channel roughened with V-shaped ribs is on the average higher than that roughened with angled ribs as well as 90°ribs of the same geometry. The secondary flows generated and the favourable direction of vortices has been cited as the reasons for better performance of V-shaped ribs over that of others. The V-shaped ribs are tested for both pointing upstream and downstream of main flow. It has been shown that those pointing downstream are slightly better in performance.

In recent times there has been renewed interest in non-conventional solar air heaters because of its applications for achieving higher collector performance. Out of various non-conventional solar energy collectors, a Two-Pass solar air heater may play an important role for getting higher heat transfer coefficient of air. The first ever reported work on the design modification dates back to morethan five decades, when Pery [11] presented experimental results on high velocity jets impingement perpendicular to the heat transfer surface. First of all, the useful heat transfer correlation of the jet plate solar air heater has been reported by Kercher and Tabakoff [12]. Later Chaudhary and Garg[13] reported the detailed studies on the jet plate solar air heater. Kuzay et al. [14] have reported to get substantial improvement in thermal efficiency when they did experiment on a finned solar airheater. Thombre and Sukhatme [15] have conducted extensive experiments for turbulent heat transfer in a shrouded fin arrays solar air heater and suggested Dittus-Boelter equation for finding heat transfer coefficient. Singh [16] also Satcunanathan and Deonarine [17] suggested the use of a Two-Pass solar air heater in order to reduce the losses from the top. It was observed from the experiment that the outer glass cover was lowered by 2 to 5°C and as a result, the losses were reduced and the efficiency of the collector was measuredto be 10 to 15% higher than the conventional single pass solar air heater. Subsequently, Wijeysundera et al. [18] studied Two -Pass concept in greater detail both analytically and experimentally. However, a search of technical literature indicates that only a limited amount of work has been done in Two -Pass solar air heater.

The present investigation is therefore, taken up with the objective of extensive experimentation on 60°inclined V-shaped ribs as artificial roughness attached to the underside of one broad wall of the two -pass solar air heater duct, to collect data on heat transfer and fluid flow characteristics. The data has been presented in the form of Nusselt number plots as a function of geometrical parameters of artificial roughness. Plots to bring out clearly the effect of these parameters and the enhancement in heat transfer achieved as a result of providing artificial roughness. The experimental data will be used to develop correlations for Nusselt number for duct flow with one V-shaped rib roughened broad wall. These correlations can be employed by the designer for the selection of suitable roughness parameters for optimal enhancement of performance of solar air heater consistent with its system and operating parameters.

II. EXPERIMENTAL PROGRAM AND PROCEDURE

An experimental set-up has been designed and fabricatedto study the effect of V-shaped ribs on heattransfer and fluid flow characteristics of flow in rectangular duct and to develop correlations for Nusselt numberfor the range of parametersdecided on the basis of practical considerations of the system and operating conditions. The experimental duct consists of a wooden channel of 1700 mm long and 200mm wide which includes fivesections, namely, smooth entrance section, smooth first pass glass entrance section, second pass test section, outlet section and transition section.

A G.I. plate of length 1500mm and width 200mm is used as an absorber plate. The lower surface of the plate is provided with artificial roughness in the form of V-shaped 60^O inclinedrib. An electric heater is fabricated by nichrome wire of 25 SWG of size 1500 X 200 was used to provide a uniform heat flux up to a maximum of 1067 $W/m^2$to the absorber plate. The power supply to the heater plate assembly was controlled through an ACvariac.A schematic diagram of the experimental set-up, cross sectional view of the duct and view of plate with roughness geometry of 60^oinclined V-shaped ribs is shown in Figs. 1(a)–(c)respectively. The roughness elements used in the roughened plate areG. I. wires of 16 gauges. The V-shaped roughness elements were fixed below the absorbing plate and a fast drying epoxy applied for fixing the roughness elementsand allowed to dry to ensure that the roughness elements were fixed properly with the surface of thePlate.

The broad specifications and the range of roughnessand flow parameters for the investigation are given in Table 1.

Work Parameter	Values
Reynolds number, (Re)	3000- 15000
Relative roughness pitch, (p/e)	6 - 12
Relative roughness height, $(e/D_{h)}$	0.027
Angle of attack, (α)	60^0
Aspect ratio, (W/H)	6.67

Table 1.Range of Roughnessand Flow Parameters

Four set of roughened plates, having relative roughness pitch (p/e) 6, 8, 10and 12 as detailed in Table above have been tested. Each set consisting of 7 runs with different flow rates covering the Reynolds number range3000–15000.

The following parameter has been measured.

- Temperature of air at inlet and outlet of test section of the duct.
- Temperature of heated plate.
- Pressure difference across orifice meter.

Validity tests have also been conducted on a conventional smooth absorber plate under similar operating conditions overall duct geometrical and flow conditions to serve as the basis of comparison of results with the values for heat transfer from the correlations available for smooth duct in the literature.

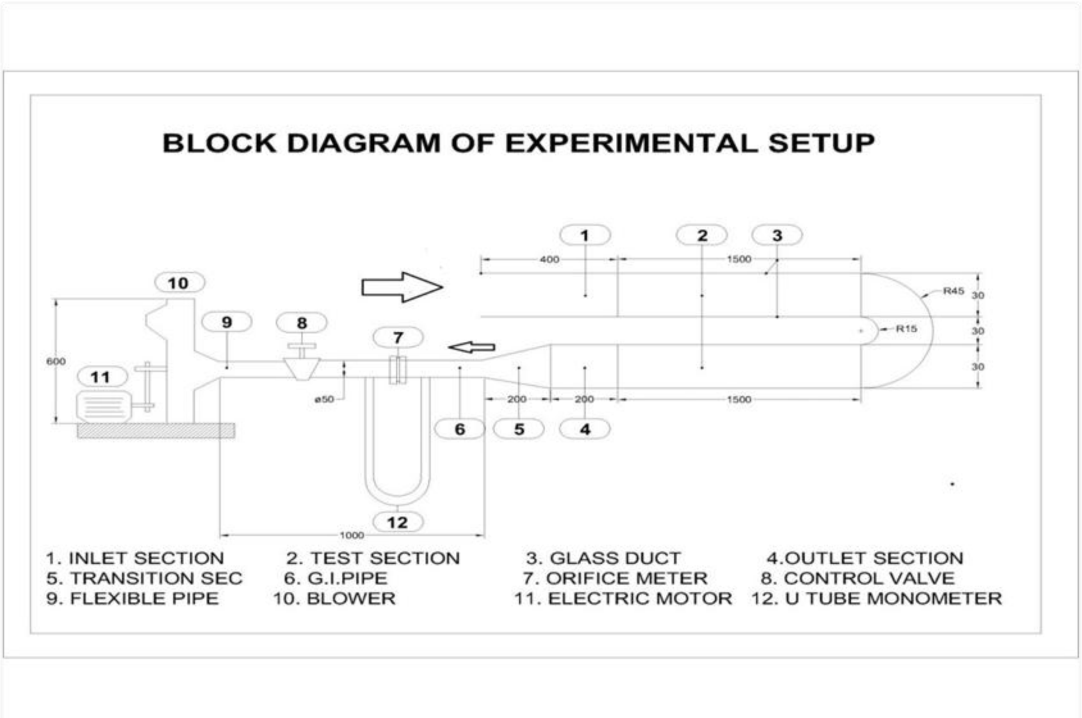

Fig: 1(a) schematic diagram of the experimental set-up

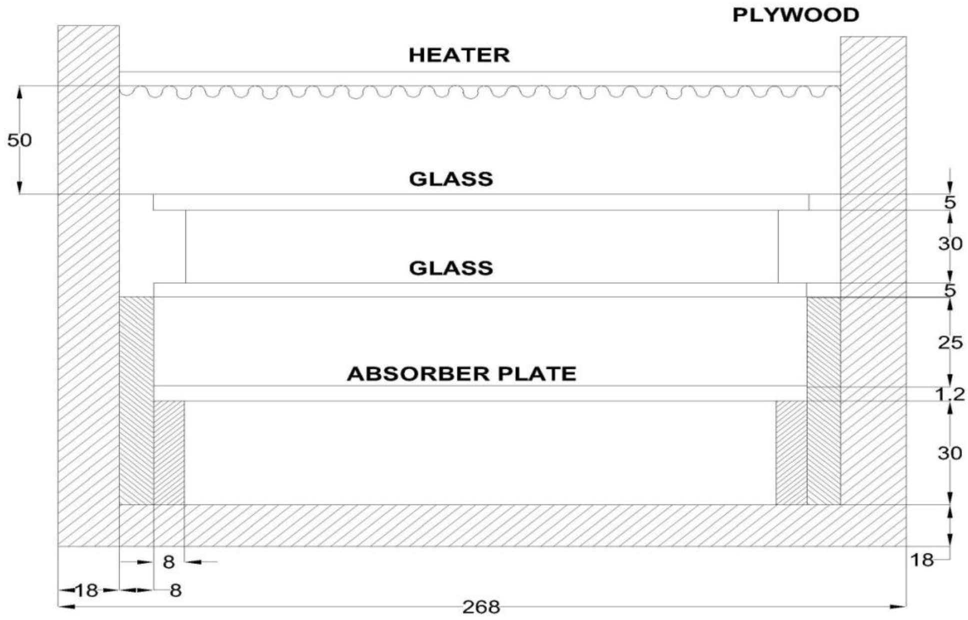

Fig: 1(b) cross sectional view of duct

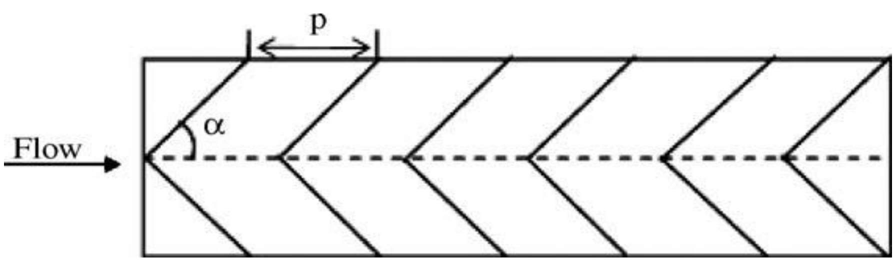

Fig. 1(c) Schematic Diagram of absorber plate with 60^{o}inclined V-shaped ribs

The air is sucked through the rectangular duct by means of a blower driven by a 3-phase, 440 V, 2.3 kW and 1420 r.p.m. AC motor. It sucked the air through the duct and a gate valve has been used to control the amount of air in the duct. The duct was covered with thermocole from the three sides and upper side of the duct was covered with the thermocole and black insulating material, to ensure that all the heat flux which is supplied from the heater plate is transferred to the duct and also to minimize the losses to the surroundings. The other end of the duct is connected to a circular pipe via a rectangular to circular transition section. The flow rate of air in the duct was measured by means of a flange type orifice meter calibrated by using a pitot tube, and the values of the coefficient of the discharge were obtained and used for calculating the flow rate of the air. Pressure drop across the orifice meter was measured by an inclined U-tube manometer with Spirit (water) as manometric fluid. A digital temperature indicator with Calibrated Copper Constantan (28 ASWG) thermocouples have been used for the measurements of inlet and outlet air temperatures; average plate temperatures; and average fluid temperatures in the duct. Fig. 2. shows the position of the thermocouples on the absorbing plate.

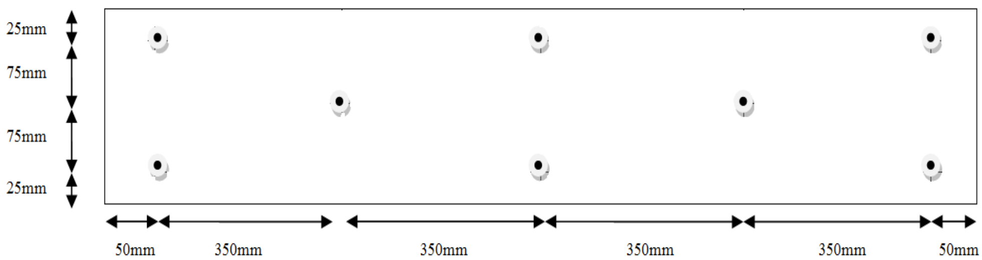

Fig. 2 Position of the Thermocouples on the absorber plate

Before starting the experiment all the thermocoupleswere checked carefully that they give the room temperature and all the pressure tappings were checked for the leakage problem.After the ribs are installed and the test section is assembled,the energy for heating is supplied for one hour to the roughened entrance section and the test section. After 1 hr. the blower is switched on to let a predetermined rate of air flow through the duct. The steady state is attained in about 2 hr. when all the temperatures and pressures were recorded. The barometric pressure is assumed to be constant during the day.

III. DATA REDUCTION

The experimental data for plate and air temperatures at various location in the duct was recorded under steady state conditions for given heat flux and mass flow rate of air. The data includes thermocouple reading and mass flow rates. This data have been reduced to obtain the average plate temperature, average air temperature, velocity of air flow in the duct and the value of heat transfercoefficient.

Average Temperature: The mean air temperature or average flow temperature T_f is the measure value at the inlet and outlet of the test section by using the following equation:

$$T_f = (T_i + T_o)/2$$

The mean plate temperature, t_p is the average of the reading of eight points located on the absorber plate.

Mass Flow Rate: In order to calculate mass flow rate, m, heat gain by the air, $Q_{air,}$ and heat transfer coefficient, h, the following equation were used:

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

Calibration of orifice plate against a standard pitot tube yielding a value of 0.624 for coefficient of discharge, C_d. where $\Delta P_o = 9.81 \rho_m \Delta h_o \sin\theta$:

Heat Gain by the Air : Heat gain by the air can be calculated by the following equation:

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

Heat Transfer Coefficient: The value ofheat transfer coefficient between the absorber plate and fluid is given by the equation:

$$m\ C_p[t_o - t_i] = hA_p\ [t_p - t_f]$$

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

Where,

Q_{air} = heat input to air , KJ
C_p= specific heat of air, KJ/kg-K
T_p= temperature of plate, ^{0}C
T_f= temperature of fluid, ^{0}C
T_o= temperature at exit, ^{0}C
T_i= temperature at entry, ^{0}C

Velocity Measurement: Velocity measurement by the following equation:

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

Where,

$\dot{m}$ =mass flow rate, kg/s
w =width of the duct, m
h =height of the duct, m
ρ = density of air, kg/m^3

Reynolds Number: The value of Reynoldsnumbermeasurement by the following equation:

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

Where
V = velocity of air (m/s)
D_h= hydraulic diameter (m)
P = density of air (kg/m^3)
μ = viscosity of air (m^2/s)

Hydraulic Diameter:

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

Nusselt Number:

$$N_u = \frac{hD_h}{k}$$

Where
h = heat transfer coefficient , W/m^{2}K
D_h= hydraulic diameter, m
K = thermal conductivity of air, W/Mk

Thermal efficiency ή

$$ή = \frac{G\,Cp(T_o - T_i)}{I}$$

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

Where
G = mass velocity, kg/s-m^2
A_p= area of the plate , m^2
I = heat flux, W/m^2

Validity Test

The Nusselt number determined from experimental data from smooth duct have been compared with the values obtained from dittus boelter equation (rosenhow and Hartnett, 1973).
Dittus boelter equation:$N_u = 0.023Re^{0.8}Pr^{0.4}$The comparison of experimental and predicted values from equation of Nusselt number has been shown in fig. 3.

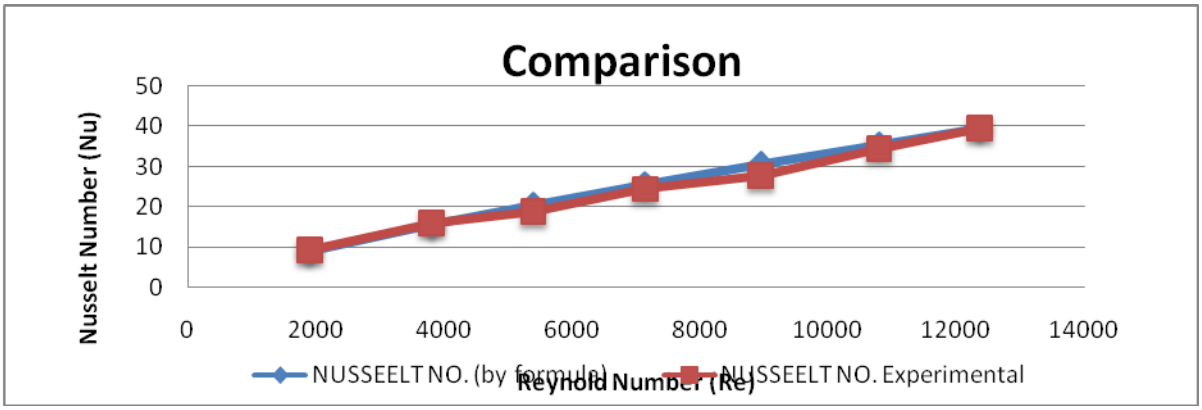

Fig. 3: Variation of Heat transfer coefficient with Relative Roughness pitch

IV. RESULTS AND OBSERVATION

The heat transfer characteristics of rectangular duct roughened with 60^0 inclined V- shaped ribs, computed on the basis of experimental data collected for various flow and roughness parameters, have been discussed below. The result has been computed with those obtained in case of smooth ducts under similar operating conditions to discuss the enhancement in Nusselt number. Fig.4Show the effect of Reynolds number on Nusselt number for fixed values of angle of attack (α), relative roughness height (e/D_h) and different values of relative roughness pitch (p/e). Nusselt number of V- shaped roughened duct is higher than that of smooth duct. It also observed that Nusselt number increases with increases of Reynolds number for both smooth and roughened duct. The Nusselt number was found maximum at relative roughness pitch (p/e) 10 because of possibility of occurrence of reattachment point in this parameter. Literature also support this result because of maximum heat transfer occurs in the vicinity of reattachment point.

Similar result is plotted in the form of variation of heat transfer coefficient (h) with relative roughness pitch (p/e) for different flow rates as shown in fig.5. it shows that the value of heat transfer coefficient (h) increases with increasing the value of relative roughness pitch, it attains maximum value of relative roughness pitch (p/e) 10, than it decreases with increasing relative roughness pitch. This pattern occurs is same for all the flow rates. The maximum experimental thermal efficiency of two pass solar air heater has been found to be 93% at the relative roughness pitch of 10 as shown in fig.6.Hence the use of artificial roughness in the form of 60^0 inclined V- shaped ribis recommended in all practical purposes in the field of heat transfer enhancement.

The variation of Nusselt number with the Reynolds number shows in table 5.1 and corresponding graph.

Table 5.1 Variation of Nusselt Number with Reynolds Number

Head (cm) h	Reynolds no. Re	Nusselt Number(Nu)				
		smooth	p/e = 6	p/e = 8	p/e = 10	p/e = 12
1	1909.9	8.23	12.76	18.37	28.14	10.38
4	3819.9	11.93	23.08	33.50	49.27	18.85
8	5402.2	15.59	32.84	47.33	68.35	24.90
14	7146.4	20.48	42.06	63.47	90.66	34.53
22	8958.5	25.50	58.04	82.65	112.74	43.30
32	10804.4	31.51	72.75	104.14	137.10	52.27
42	12378.1	39.70	85.07	120.45	157.30	62.92

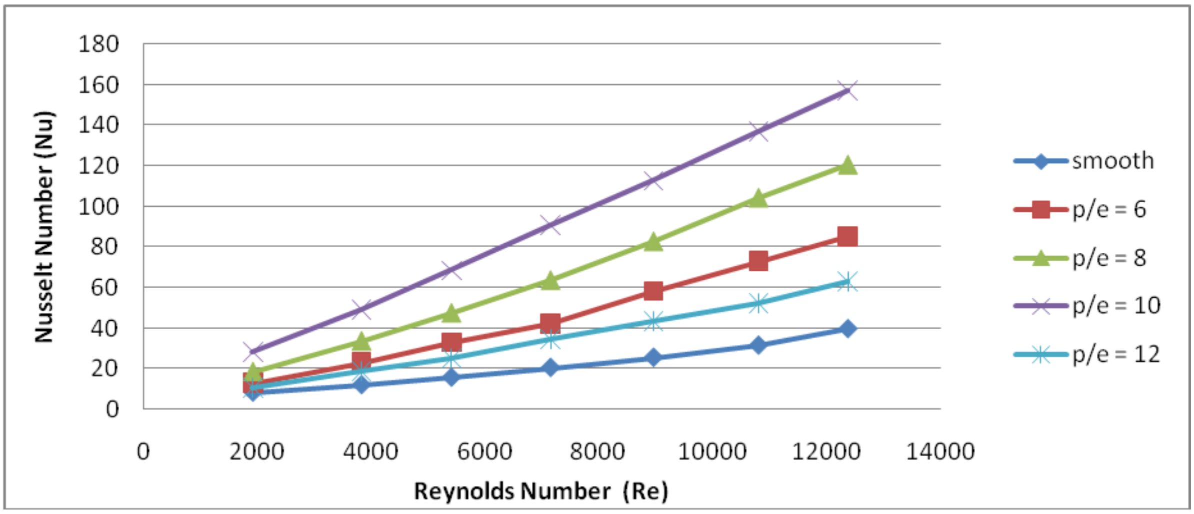

Fig.4:Variation of Nusselt number with the Reynolds number

The Variation of Heat transfer coefficient with Relative Roughness pitchshows in table 5.2 and corresponding graph.

Table 5.2 Variation of Heat transfer coefficient with Relative Roughness pitch

S.No.	Reynolds (Re)	Heat transfer coefficient h(W/m^{2o}K)			
		6	8	10	12
1	1909.9	6.55	9.43	14.44	5.33
2	3819.9	12.87	17.20	24.26	9.67
3	5402.2	18.40	24.29	38.68	12.78
4	7146.4	25.70	32.58	51.67	17.72
5	8958.5	35.95	42.42	65.57	22.23
6	10804.4	42.47	53.46	72.94	26.83
7	12378.1	54.96	61.83	85.87	32.30

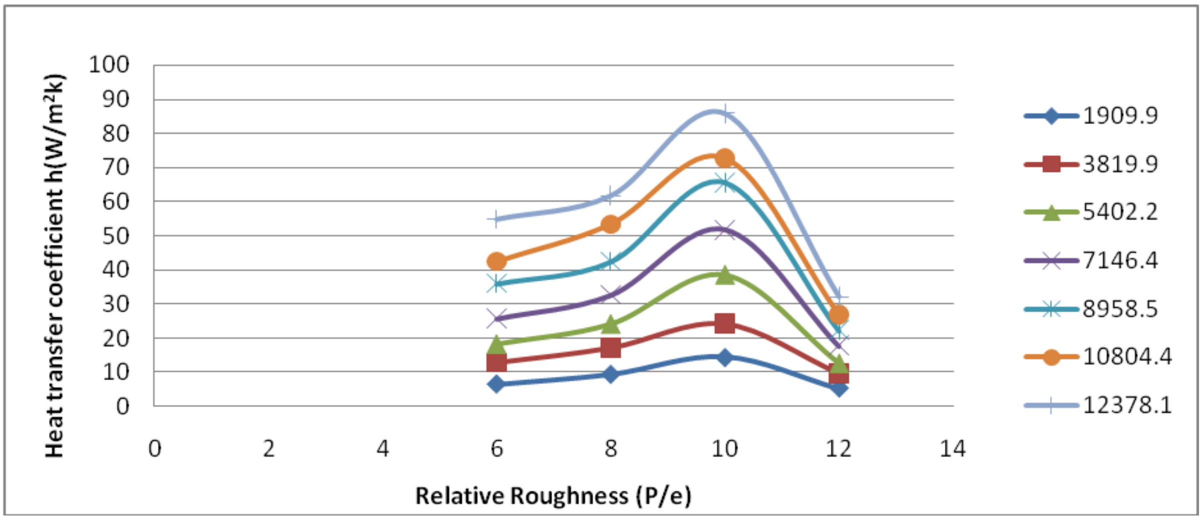

Fig. 5: Variation of Heat transfer coefficient with Relative Roughness pitch

5.3 The Variation of Thermal Efficiency with Reynolds numbershows in table 5.3 and corresponding graph.

Table 5.3 Variation of Thermal Efficiency with Reynolds number

S.No.	Reynolds (Re)	Thermal Efficiency (η%)				
		smooth	p/e = 6	p/e = 8	p/e = 10	p/e = 12
1	1909.9	15.81	26.43	31.24	38.09	20.11
2	3819.9	25.36	44.76	51.81	61.00	34.97
3	5402.2	34.02	56.18	66.80	73.42	42.02
4	7146.4	40.24	64.42	74.12	81.67	50.50
5	8958.5	44.46	73.04	82.19	88.34	56.07
6	10804.4	48.36	77.27	86.20	92.82	60.70
7	12378.1	51.49	79.00	88.88	93.75	64.19

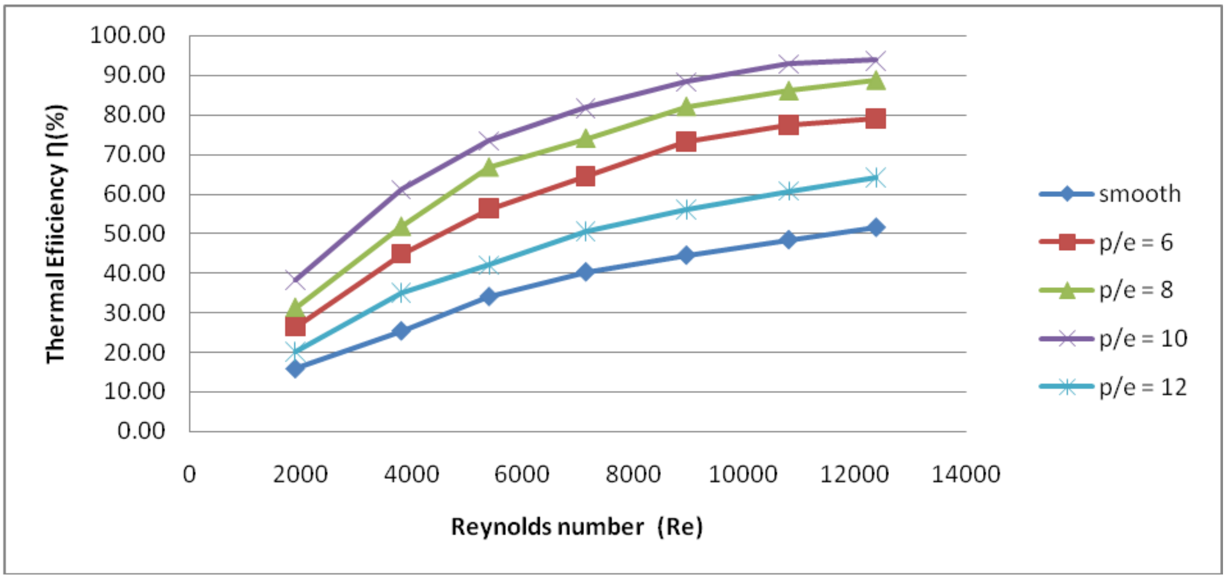

Fig. 6: Variation of Thermal Efficiency with Reynolds number

V. CONCLUSIONS

The following conclusions can be drawn from this work:

[1] In general, Nusselt number increases with an increase of Reynolds number. The values of Nusselt number is substantially higher as compared to those obtained for smooth absorber plates. This is due to distinct change in the fluid flow characteristics as a result of roughness that causes flow separations, reattachments and the generation of secondary flows.

[2] The maximum enhancement of Nusselt number as a result of providing artificial roughness have been found to be 3.96times that of smooth duct for an angle of attack of 60^0. It appears that the flow separation and the secondary flow resulting from the presence of V-shaped ribs and the movement of resulting vortices combine to yield an optimum value of angle of attack.

[3] It found that the Nusselt number increases with an increase of Reynolds number and attains maximum for relative roughness pitch of 10 and the decreases with an increases of relative roughness pitch. The variation of Nusselt number with relative roughness pitch (p/e) is in significant at lower values of Reynolds number, but at higher Reynolds number here is a substantial effect.

[4] The maximum experimental thermal efficiency of two pass solar air heater has been found to be 93% at the relative roughness pitch of 10.

Nomenclatures

A_c	surface area of absorber plate, m^2
C_p	specific heat of air, J/kg K
D, Dh	equivalent or hydraulic diameter of duct, m
e	rib height,
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/m K
L	length of test section of duct or long way length of mesh, m
m	mass flow rate, kg/s
P	pitch, m
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
W	width of duct, m

Dimensionless parameters

e/D_h relative roughness height
e/H rib to channel height rat
Nu Nusselt number
p/e relative roughness pitch
Pr Prandtl number
Re Reynolds number
W/H duct aspect ratio

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

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