iJRASET
International Journal For Research in Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 7 **Issue:** VIII **Month of publication:** August 2019

DOI: http://doi.org/10.22214/ijraset.2019.8049

www.ijraset.com

Call: 08813907089 | E-mail ID: ijraset@gmail.com

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.177
Volume 7 Issue VIII, Aug 2019- Available at www.ijraset.com

Heat Transfer Enhancement of Two Pass Solar Air Heater using Discrete V-Shape Ribs on Absorber Plate

B.R. Maravi[1], D. S. Rawat[2]
[1]*Research scholar, Department Of Mechanical Engineering, Jabalpur Engineering College Jabalpur.*
[2]*Professor, Department Of Mechanical Engineering, Jabalpur Engineering College Jabalpur.*

***Abstract:** In this paper, heat transfer enhancement in two pass solar air heater with discrete V- shaped rib is investigated. Discrete ribs are attached on absorber plate . The ratio of rib height to hydraulic diameter (e/D) is 0.0287, while the range of rib pitch-to-height ratio is (P/e = 6-12). Experiments are carried out using air as the convective fluid, with the Reynolds number rang (Re = 2000 -14000). Air enters the upper channel of the air heater and subsequently flows 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.*
In addition ,heat transfer increase as P/e increase from 6 to 10 and thereafter decrease in heat transfer coefficient in resulted.
***Keyword:** Enhancement of heat transfer, Two- Pass solar air heater, Thermal efficiency, Mass flow rate ,Reynolds number.*

I. INTRODUCTION

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 of thermal performance. Use of artificial roughness in the form of repeated ribs of various shapes and orientations has been found to be a convenient method to investigate the performance of such systems.

The use of artificial roughness in solar air 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 [3],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 been found to increase by a factor up to 2.7 times of smooth duct. Saini and Saini [1] 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 [5]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^{o}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 [4] 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 artificial roughness. 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 researchers have 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

©IJRASET: All Rights are Reserved

[6]. Han et al. [7], Lau et al. [8] and Taslim et al. [9] 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. 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.

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 discrete V-shaped rib roughened broad wall.

II. EXPERIMENTAL PROGRAM AND PROCEDURE

An experimental set-up has been designed and fabricated to study the effect of discrete V-shaped ribs on heat transfer and fluid flow characteristics of flow in rectangular duct and to develop correlations for Nusselt number for the range of parameters decided 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 five sections, namely, smooth entrance section, smooth first pass glass entrance section, second pass test section, outlet section and transition section.

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 discrete V-shaped rib. 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 AC variac. A schematic diagram of the experimental set-up, cross sectional view of the duct and view of plate with roughness geometry of discrete V-shaped ribs is shown in Figs. 1-4 respectively. The roughness elements used in the roughened plate are G. 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 elements and allowed to dry to ensure that the roughness elements were fixed properly with the surface of the Plate.

The following parameter has been measured.

A. Temperature of air at inlet and outlet of test section of the duct.
B. Temperature of heated plate.
C. 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.

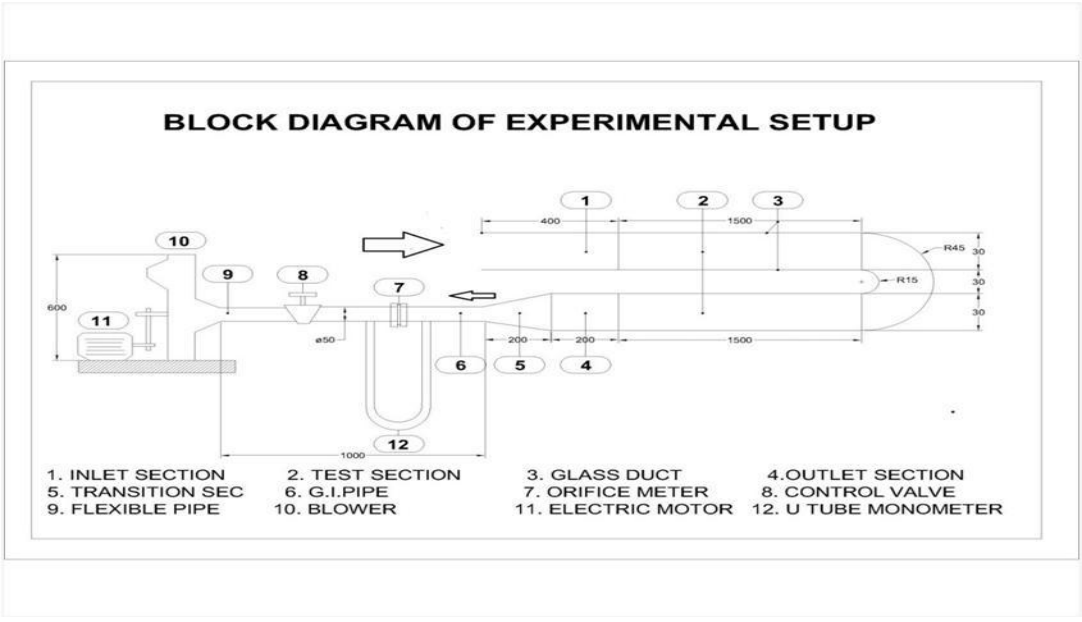


Fig.1- Schematic diagram of the experimental set-up

©IJRASET: All Rights are Reserved

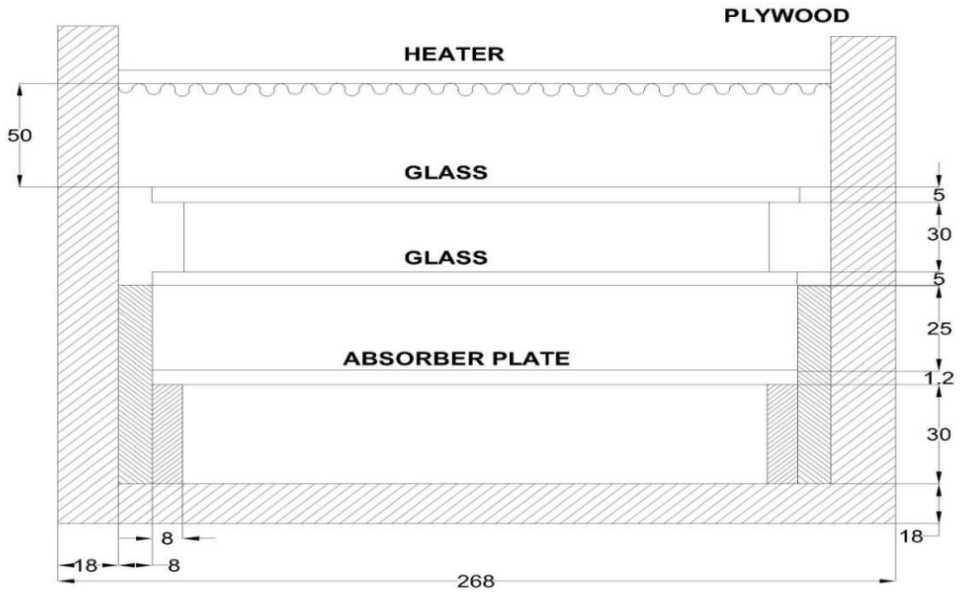


Fig:2- cross sectional view of duct

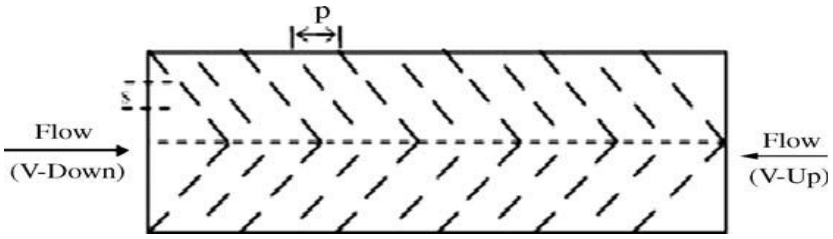


Fig.3- Schematic Diagram of absorber plate with discrete V-shaped ribs

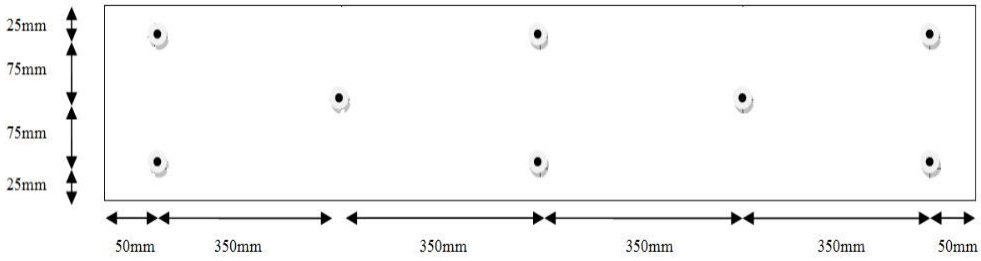


Fig.4- Position of the Thermocouples on the absorber plate

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 transfer coefficient.

1) *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.

©IJRASET: All Rights are Reserved

2) *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 $\Delta P_o = 9.81 \rho_m \Delta h_o \sin\theta$:

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

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

4) *Heat Transfer Coefficient:* The value of heat transfer coefficient between the absorber plate and fluid is given by the equation:

$$m\, C_p [t_o - t_i] = h A_p \left[t_p - t_f\right]$$

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

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

5) *Velocity Measurement:* Velocity measurement by the following equation:

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

Where,

ṁ =mass flow rate, kg/s w w = width of the duct, m h h =height of the duct, m
ρ = density of air, kg/m^3

6) *Reynolds Number:* The value of Reynolds number measurement 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)

7) *Hydraulic Diameter*

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

8) *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

©IJRASET: All Rights are Reserved

9) Thermal efficiency ή

$$\acute{\eta} = \frac{G\ Cp(T_o - T_i)}{I}$$

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

Where

G = mass velocity, kg/s-m^2

A_p = area of the plate , m^2

I = heat flux, W/m^2

10) 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: $N_u = 0.023 Re^{0.8} Pr^{0.4}$ The comparison of experimental and predicted values from equation of Nusselt number has been shown in fig. 3.

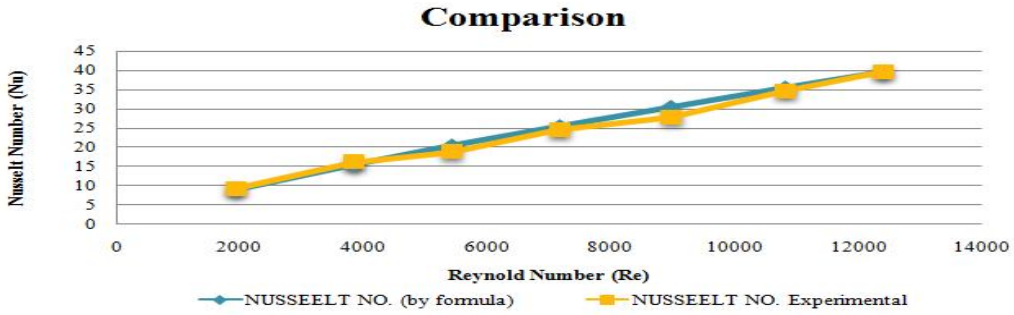


Fig. 5-comparison of expeimental and predicted values of smooth duct

IV. RESULTS AND OBSERVATION

The heat transfer characteristics of rectangular duct roughened with discrete 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. It is observed the effect of Reynolds number on Nusselt number for fixed values of angle relative roughness height (e/D_h) and different values of relative roughness pitch (p/e). Nusselt number of discrete V- shaped roughened duct is higher than that of smooth duct. It is 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.

The variation of Nusselt Number with Reynolds Number show in table 1 and fig. 6 .

Table 1 - Variation of Nusselt Number with Reynolds Number

Head (cm) h	Reynold number (Re)	Nusselt Number (Nu)				
		smooth	p/e = 6	p/e = 8	p/e = 10	p/e = 12
1	2112.18	4.33	7.25	10.32	16.09	12.63
4	4224.36	10.63	13.46	17.11	21.68	19.16
8	5974.15	18.97	22.59	26.06	31.88	28.78
14	7903.06	28.23	32.52	32.68	41.53	38.16
22	9907.01	36.38	39.35	43.55	52.07	47.54
29	11374.45	41.27	45.35	49.06	59.55	55.43
32	11948.31	43.36	47.83	52.13	63.03	59.15

©IJRASET: All Rights are Reserved

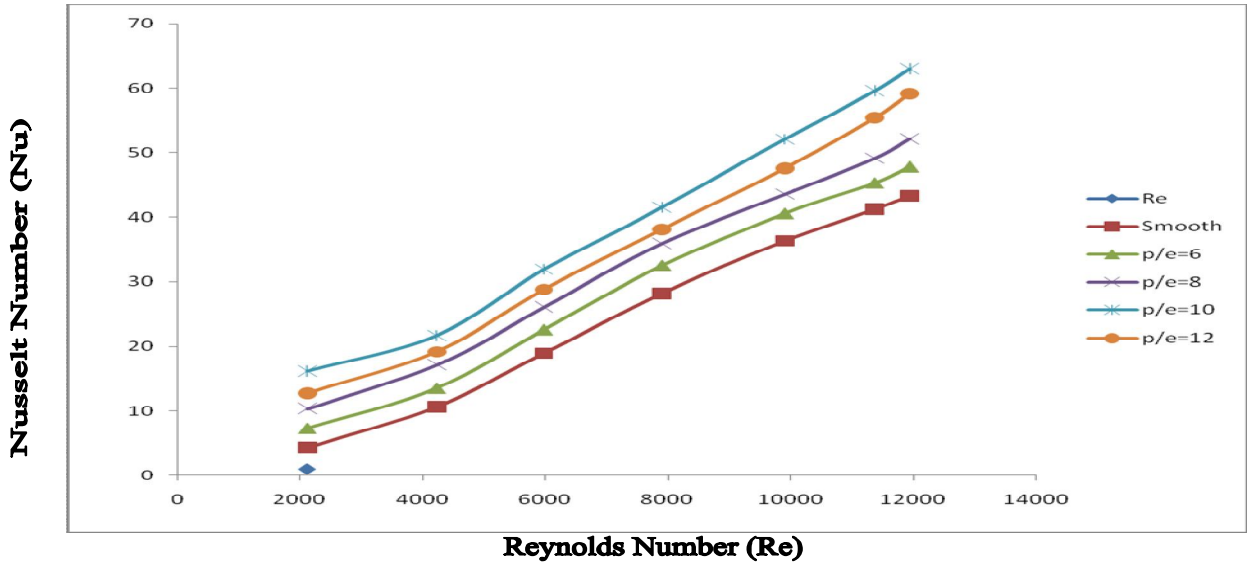


Fig. 6 - Variation Of Nusselt Number With Reynolds Number

The Variation of Heat transfer coefficient with Relative Roughness pitch shows in table 2 and fig. 7 .

Table 3 -Variation of Thermal Efiiciency with Reynolds number

S.No.	Reynolds number (Re)	Thermal Efficiency η (%)				
		smooth	p/e = 6	p/e = 8	p/e = 10	p/e = 12
1	2112.18	5.79	10.14	11.59	15.94	13.04
2	4224.36	14.49	20.28	26.08	31.88	28.98
3	5974.15	28.69	36.88	40.98	49.18	45.08
4	7903.06	37.95	48.8	54.22	65.06	59.64
5	9907.01	54.37	61.17	67.97	81.56	74.76
6	11374.45	62.43	70.23	78.03	93.64	85.84
7	11948.31	65.58	73.77	81.97	98.3	90.17

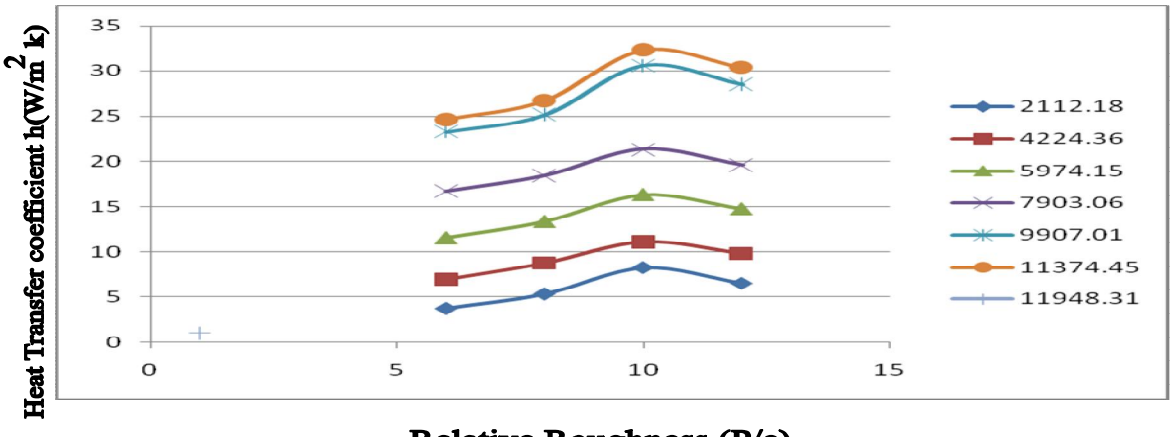


Fig. 7- variation of heat transfer coefficient with relative roughness pitch

©IJRASET: All Rights are Reserved

The Variation of Thermal Efficiency with Reynolds number shows in table 3 and fig. 8 .

Table 2 - Variation of Heat Transfer coefficient with Relative Roughness pitch

S.No	Reynolds number (Re)	Heat Transfer coefficient h ($W/m^{2o}K$)			
		6	8	10	12
1	2112.18	3.72	5.3	8.26	6.48
2	4224.36	6.91	8.78	11.12	9.83
3	5974.15	11.59	13.37	16.36	14.77
4	7903.06	16.69	16.77	21.32	19.58
5	9907.01	20.19	22.35	26.73	24.4
6	11374.45	23.28	25.18	30.56	28.45
7	11948.31	24.55	26.75	32.35	30.36

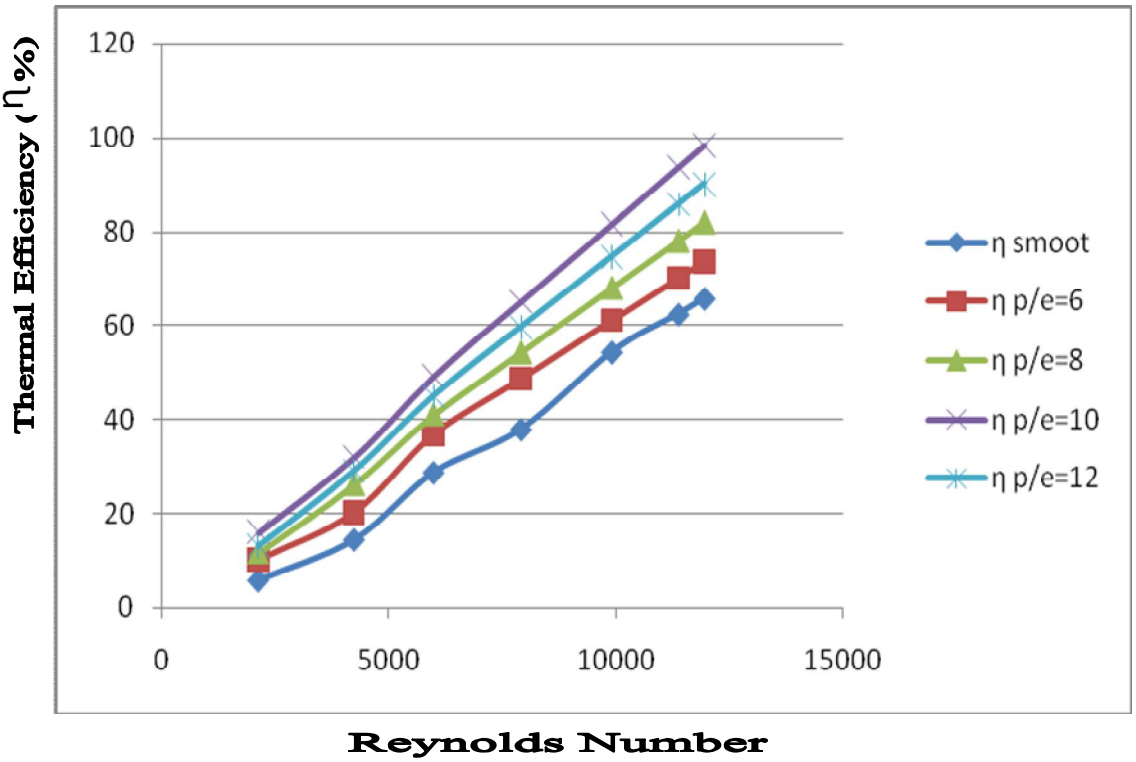


Fig. 8 - variation of thermal efficiency with reynolds number

V. CONCLUSION

The following conclusion can be made from this work:

A. 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 flow.

B. It found that the Nusselt number increases with an increase of Reynolds number and attains maximum for relative roughness pitch of 10 and then , 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 , there is a substantial effect.

C. The maximum experimental thermal efficiency of two pass solar air heater has been found to be 98% at the relative roughness pitch of 10.

©IJRASET: All Rights are Reserved

1) 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
Tpm	mean plate temperature, K
W	width of duct, m

2) Dimensionless Parameters

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

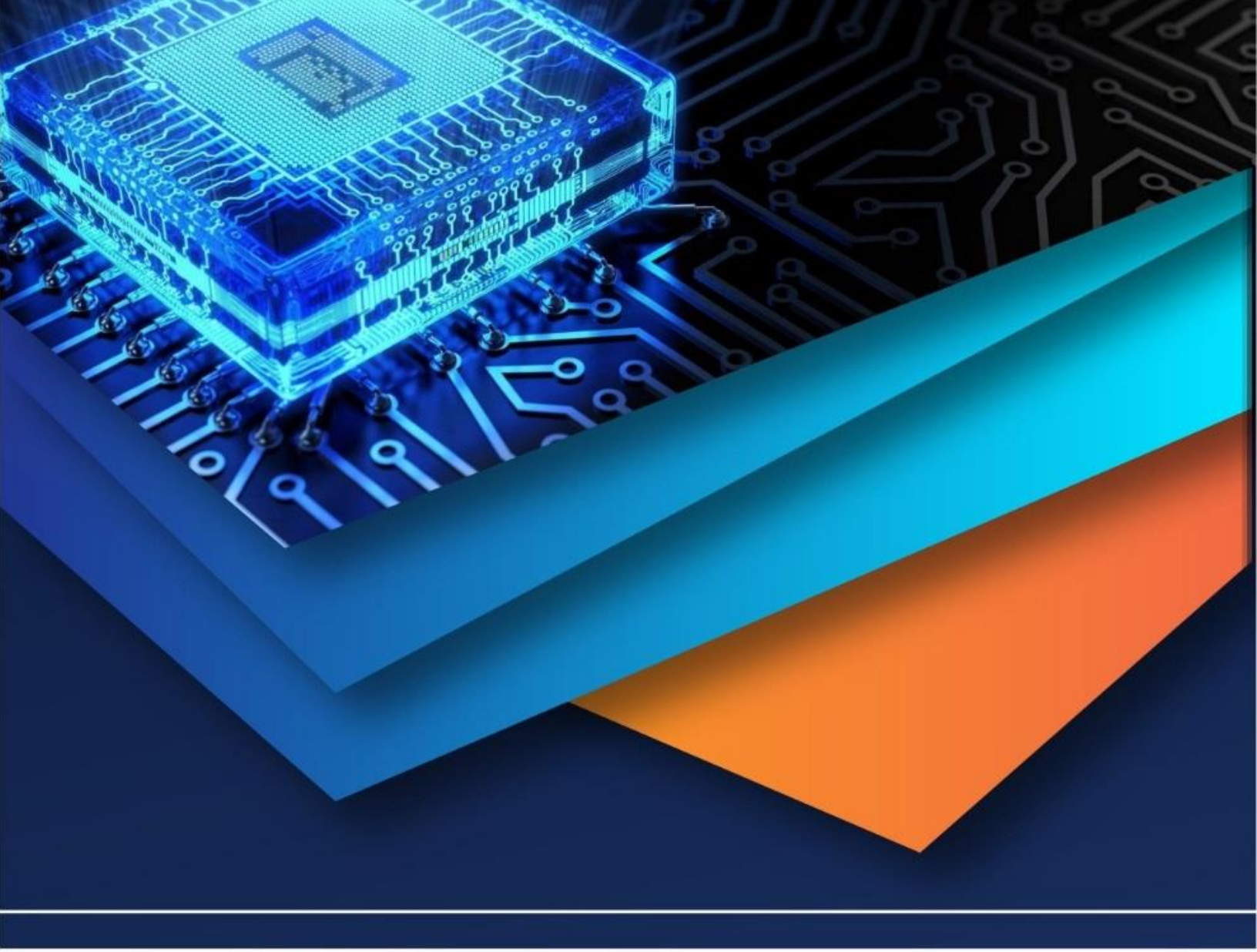
Greek

ηth	thermal efficiency
μ	dynamic viscosity, Ns/m^2
ρ	density of air, kg/m^3

REFERENCES

[1] R.P. Saini, J.S. Saini,"Heat transfer and friction factor correlations for artificially roughened ducts with expanded metal mesh as roughness element", Int. J. Heat Mass Transfer 40 (1997) 973–986.
[2] D. Gupta, "Investigations on fluid flow and heat transfer in solar air heaters with roughened absorbers", Ph.D. Thesis. University of Roorkee, India, June 1993.
[3] Prasad, S.C. Mullick," Heat transfer characteristics of a solar air heater used for drying purposes", Appl. Energy 13 (1983) 83– 93
[4] B.N. Prasad, J.S. Saini," Effect of artificial roughness on heat transfer and friction factor in a solar air heater", Solar Energy 41 (1988) 555–560.
[5] Karwa R. "Experimental studies of augmented heat transfer and friction in asymmetrically heated rectangular ducts with ribs on the heated wall in transverse, inclined, v-continuous and v-discrete pattern". Int Comm Heat Mass Transf 2003; 30(2):241–50.
[6] Karwa R. "Experimental studies of augmented heat transfer and friction in asymmetrically heated rectangular ducts with ribs on the heated wall in transverse, inclined, v-continuous and v-discrete pattern". Int Comm Heat Mass Transf 2003; 30(2):241–50.
[7] S.C. Lau, R.T. Kukreja, R.D. McMillin," Effects of V-shaped rib arrays on turbulent heat transfer and friction of fully developed flow in a square channel", Int. J. Heat Mass Transfer 34 (1991) 1605–1616.
[8] Thombre, S.B., Sukhatme,S.P., 1995. "Turbulent flow heat transfer and friction factor characteristics of Shrouded fin arrays with uninterrupted fins". Experimental Thermal and Fluid Science, 10,388.
[9] M.E. Taslim, T. Li, D.M. Kercher, "Experimental heat transfer and friction in channels roughened with angled, V-shaped, and discrete ribs on two opposite walls", ASME J. Turbomachinery 118 (1996) 20–28.
[10] Singh, S.N., 2006."Performance studies on continuous longitudinal fins solar air heater". Proceedings of the 1st National Conference on Advances in Energy Research, IIT Bombay, 205-210.
[11] Satcunanathan, S., Deonarine,S., 1973." A two pass solar air heater." Solar Energy, 15,41.
[12] Wijeysundera, N. E., Ah, L.L and Tjoie, L.E., 1982. "Thermal performance study of two-pass solar air heater", Solar Energy, 28,363.

©IJRASET: All Rights are Reserved



10.22214/IJRASET

45.98

IMPACT FACTOR: 7.129

IMPACT FACTOR: 7.429

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Call : 08813907089 (24*7 Support on Whatsapp)