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Performance evaluation of roughened solar air heater having M-shaped as roughness geometry on the absorber plate

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Abstract

As thermal efficiency of conventional solar air heater is low, best way is to enhance its thermal efficiency is make the flow turbulent. This can be achieved by using the artificial roughness on underside of absorber plate. An attempt has been made to enhance its thermal as well as thermohydraulic performance by providing roughness elements. An experimental investigation has been carried out on M-shaped ribs having circular cross section on absorber plate. The duct is having an aspect ratio of (W/H) 11.41, relative roughness height (e/D) 0.033-0.077, relative roughness pitch (P/e) 12.5-75 and angle of attack (α) 30-60°. The range for Reynolds number has been considered to be 3000-22000. The best result of thermal and thermohydraulic performance has been observed at (e/D) 0.077, (P/e) 25 and (α) 60°.

Keywords: Solar air heater; Artificial roughness; Laminar sublayer; Heat transfer; Thermohydraulic performance.

1. Introduction

Energy is the basic requirement for human life and its development. The demand for energy is growing at an alarming rate year after year due to high population growth rate. There are limiting conventional energy resources such as fossil fuels (coal, oil and natural gas). The fossil fuels are rapidly depleting and the fossil fuels era is gradually coming to an end. However, the combustion of fossil fuels has also caused air pollution resulting in global warming and ozone layer depletion. In view of these problems associated with conventional energy resources, focus is now shifting towards the conservation of energy and use renewable energy resources. The best way is to conserve energy by optimization of energy and using highly efficient devices. Also, the use of non-conventional energy resources such as solar energy, wind energy etc. extends the span of conventional energy resource.

Solar energy is cheap and abundantly available in nature. It is also called zero emission energy. It has very wide applications like food cooking, water heating, air heating and power generation. The main disadvantage of it is that it is not be available in night time and cloudy days. Application of solar energy is air heating which is used in drying of agriculture products (vegetables and fruits) and also in heating, ventilation and air-conditioning system in buildings. The device which through air is to be heated is called conventional solar air heater. The component of solar air heater is shown in Figure 1 [1].

The main problem associated with solar air heater is its poor thermal efficiency. The reason of its poor thermal efficiency is very low heat transfer coefficient of air. A laminar sublayer is formed in the vicinity

of absorber plate which causes lower heat transfer rate. The thickness of laminar sublayer is given by expression $\delta = 5\times(\nu/u_t)$ [2].
One way is to reduce the size of laminar sublayer which increases the stream velocity of flow. Because, the fluid will not be held in stagnant condition near the absorber plate. Another way is to provide turbulence mixing in fluid flow. Care has to be taken that turbulence occurs in laminar sublayer and it does not disturb the main fluid flow. Otherwise, it leads to higher pumping penalty which will increase the capacity of blower and consumption of energy. This turbulence in flow is provided under the absorber plate through the artificial roughness. Different geometries have been provided by many investigators on absorber plate to enhance heat transfer rate.

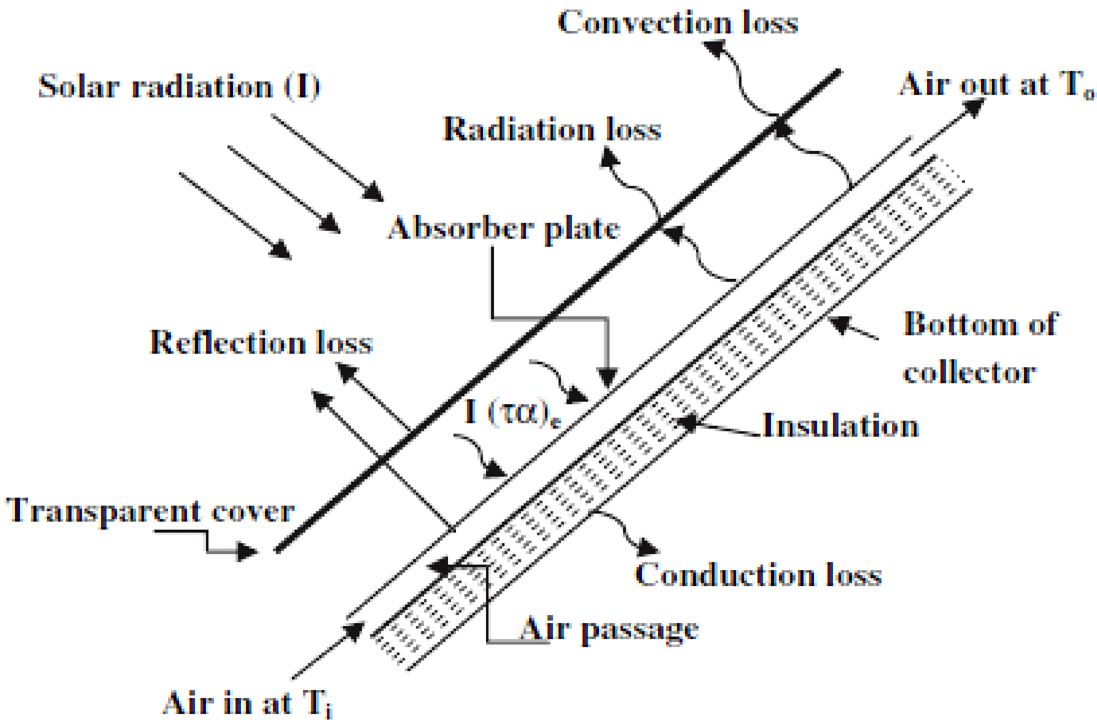

Figure 1. Schematic of conventional solar air heater

1.1 Literature review
Firstly in 1861, Joule made an attempt to enhance heat transfer coefficient in condensing steam. In view of this many investigations on roughness geometries have been made for enhancement of heat transfer in cooling of turbine blades, nuclear reactor and electronics equipments. These roughness geometries have been studied in literature [3-5].
Prasad and Mullick [6] studied the effect of transverse wire ribs on heat transfer, friction factor and plate efficiency of solar air heater. This roughness geometry enhances the plate efficiency from 0.63 to 0.71. Prasad and Saini [7] studied transverse continuous ribs as roughness geometry on the absorber plate; Nusselt number and friction factor value enhance upto 2.38 and 4.25 times respectively. Verma and Prasad [8] performed experimentation in actual climatic conditions and found the optimal value of thermohydraulic performance is about 71% corresponding to the optimal value of relative roughness. The heat transfer enhancement factor was lie in between 1.25-2.8. Karwa [9] used the transverse rectangular cross section ribs as roughness geometry. Stanton number and friction factor enhancement over smooth plate was found 60-90% and 2.68-2.94 times respectively. Sahu and Bhagoria [10] experimentally investigated the effect of 90° broken ribs as roughness geometry and found thermal efficiency of roughened solar air heater in range of 51-83.5% depending on flow parameters and conditions. Gupta et al. [11] carried out an experimentation to see the effect of e/D and α with respect to flow direction and Re on thermohydraulic performance of a roughened solar air heater. It was found out that maximum heat transfer and friction factor was of the order of 1.8 and 2.7 times respectively corresponding to α of 60° and 70°.

Karwa et al. [12] studied the effect of repeated 60°inclined rectangular cross section on heat transfer and friction factor. The enhancement in Stanton number and friction factor was reported of the order of 22-32% and 1.12-1.16 times over transverse ribs. Aharwal et al. [13] experimentally investigated the effect of inclined split ribs having square cross section on absorber plate. The effect of roughness geometry has been investigated on heat transfer and friction characteristics of rectangular duct. The enhancement in heat transfer was reported to be 1.71-2.59 times while friction factor was found to be 1.48-2.26 times over the smooth duct. Saini and Saini [14] experimentally investigated the effect of wire mesh absorber plate on heat transfer and friction factor of solar air heater. The maximum heat transfer value has been found out to be 4 times the value of heat transfer of smooth duct. Momin et al. [15] carried out an experimental investigation to see the effect of geometrical parameters of V-shaped ribs while Muluwork et al. [16] studied the comparison of thermal performance of staggered discrete V-apex up and down ribs corresponding to transverse staggered discrete ribs. It was found out that that the V-down discrete ribs have higher Nusselt number than V-up and transverse discrete roughened surface.

Varun et al. [17] experimentally investigated the effect of combination of inclined and transverse ribs as roughness geometry on heat transfer and friction factor. It has been reported that the combination of roughness geometry give better performance over the transverse ribs for e/D and P/e as 0.03 and 8 respectively. Varun et al. [18] computed effective efficiency based on experimental values for the range of parameters studied. An attempt has also been carried out to optimize the thermal efficiency for the same system by using Taguchi method under similar condition. Layek et al. [19] studied the effect of repeated transverse compound rib-grooved arrangement on heat transfer and friction factor. It has been found out that enhancement in Nusselt number and friction factor was of the order of 2.4 and 2.6 times respectively over a smooth duct. Jaurker et al. [20] studied the heat transfer and friction characteristics of rib groove roughened rectangular duct. It has been reported that enhancement in heat transfer coefficient and friction factor was obtained 2.7 and 3.6 times respectively. Bhagoria et al. [21] experimentally investigated the effect of wedge shaped transverse integral ribs as roughness geometry on Nusselt number and friction factor. It has been found out that enhancement is of the order 2.4 and 5.3 in Nusselt number and friction factor respectively over smooth duct. Saini and Verma [22] investigated dimpled shape as roughness geometry on absorber plate and found maximum Nusselt number and friction factor corresponding to e/D is 0.0379 and 0.0289 for fixed value of P/e (10). Hans et al. [23] experimentally investigated the effect of geometrical parameters of multiples V-ribs on Nusselt number and friction factor. It has been found out that the enhancement in Nusselt number and friction factor was 6 and 5 times respectively.

Promvonge et al. [24] studied the effect of inline and staggered triangular channel with longitudinal vortex generators on Nusselt number and friction factor. The enhancement factor of Nusselt number and friction factor was order of 2.2-2.5 and 2.2-5.5 respectively. Bhushan and Singh [25] experimentally investigated the effect of geometrical parameters of protruded dimpled shaped roughness geometry on Nusselt number and friction factor. It was reported that enhancement in Nusselt number and friction factor was of the order of 3.8 and 2.2 times respectively over smooth duct. Lanjewar et al. [26] carried out an experimental investigation for different orientation of W ribs roughness on absorber plate of solar air heater. The thermohydraulic performance of W-down and W-up found to be 1.98 and 1.81 times respectively over V shape roughness geometry. Also, it has been seen the effect of different geometrical parameters on heat transfer and friction factor.

2. Material and methods

The experimental set up has been designed and fabricated in order to carry out an experimental study. It may be noted that ASHRAE Standard 93-77 [27] recommends minimum entry and exit length of $5\sqrt{WH}$ and $2.5\sqrt{WH}$ i.e. 236.6 mm and 118.3 mm. The rectangular duct of size 1375 mm × 287 mm × 197 mm has been fabricated from red marandi wood. The schematic diagram of experimental setup is shown in Figure 2. The test section was of 1000 mm long and having cross section of 160 mm (width) × 14 mm (depth). The sectional view of solar air heater duct is shown in Figure 3. An electric heater has been fabricated with the help of nichrome wire of 0.5mm diameter. It has a size of 1000 mm × 160 mm and fabricated by series and parallel loop combination of nichrome wire on 4 mm asbestos sheet. The back side of heater has been insulated with the help of 19 mm thickness plywood panel and glass wool to minimize the heat losses. The actual diagram of fabricated nichrome heater is shown in Figure 4. The absorber plate (Aluminium) having M shape of circular wire cross section as roughness geometry formed on its bottom side.

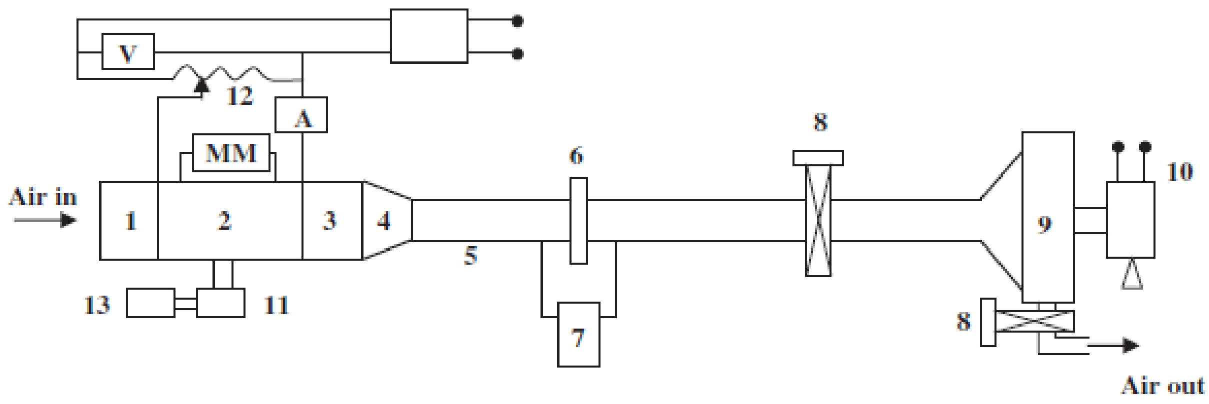

Figure 2. Schematic diagram of experimental setup
1. Entry Section, 2.Test Section, 3. Exit Section, 4. Mixing Plenum, 5. G.I Pipe, 6. Orifice Plate, 7. U-tube Manometer, 8. Control Valves, 9. Centrifugal Blower, 10. Electric Motor, 11. Selector Switch, 12. Voltage Variac, 13.Temperature Indicator, A – Ammeter, V – Voltmeter, MM - Micro-meter

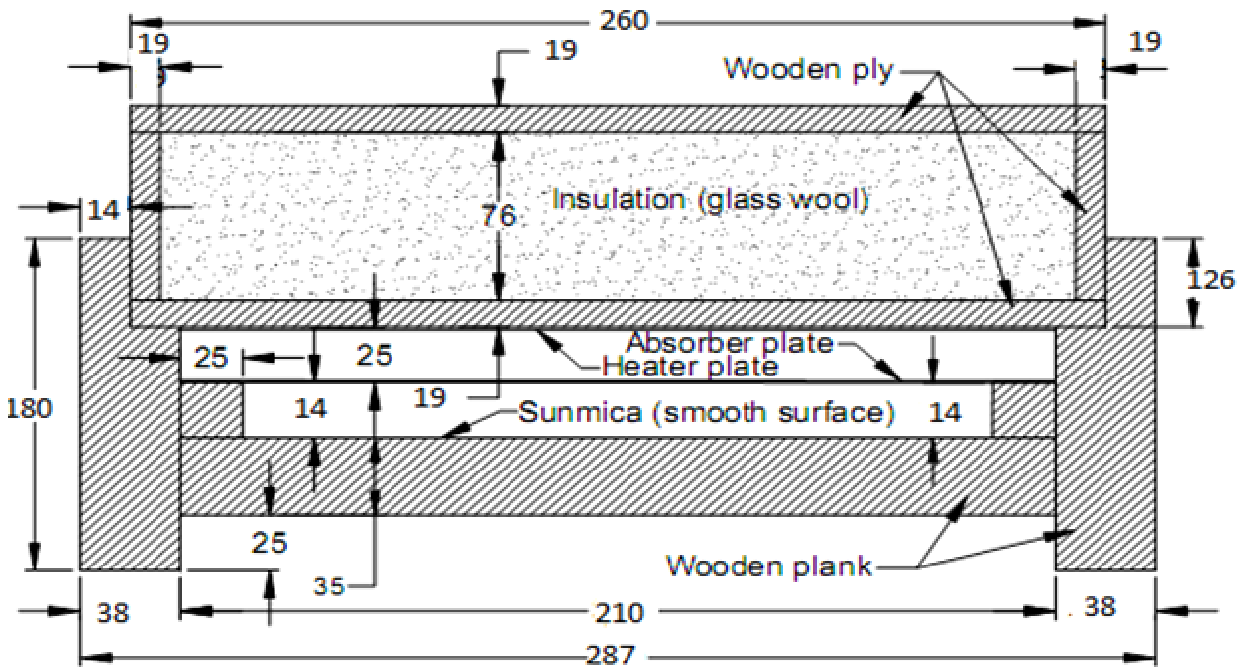

Figure 3. Sectional view of rectangular solar air heater duct

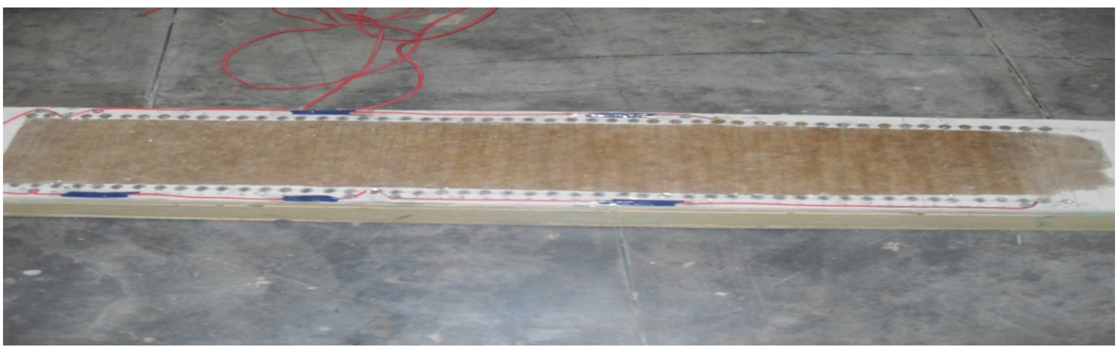

Figure 4. Actual diagram of fabricated Nichrome heater

3. Experiment

3.1 Experimental procedure

All the data measured under steady state condition. Six values of mass flow rates have been measured. For initial set of reading it takes approximately 1.5-2 hours to attain quasi-steady state while in later set of results it would take approximately 1 hour to attain the same.

Following parameters were measured for each set of reading:

1. Absorber plate temperature at 12 locations.
2. Inlet and outlet temperature of air.
3. Ambient temperature.
4. Pressure difference across orifice meter.
5. Pressure drop across the test section of duct.

These were the range of flow and roughness parameters as shown in Table 1.

Table 1. Range of flow and roughness parameters

S. No.	Parameters	Range
1	Roughness height, e	1mm, 1.5mm, 2 mm (3values)
2	Rib pitch, P	25mm, 50 mm, 75 mm (3 values)
3	Rib width, w	100 mm
4	Relative roughness height (e/D)	0.03884-0.07768 (3values)
5	Duct Aspect ratio (W/H)	11.43
6	Angle between geometry (α)	30-60° (3 values) (Figures 5, 6, and 7)
7	Reynolds Number (Re)	3000-22000
8	Solar Insolation (I)	1000 W/m^2

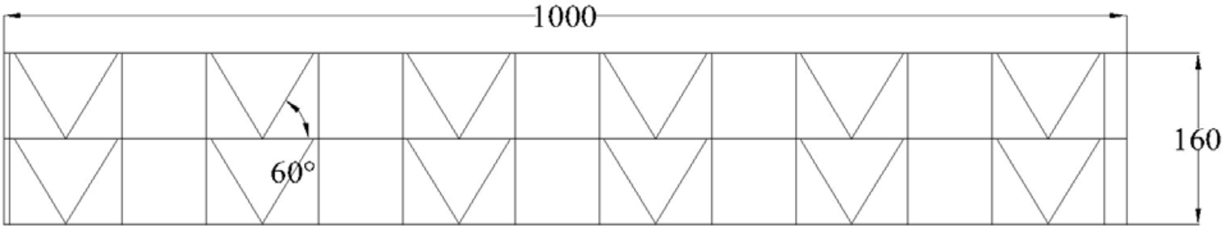

Figure 5. Roughness geometry P = 75mm, angle = 60°

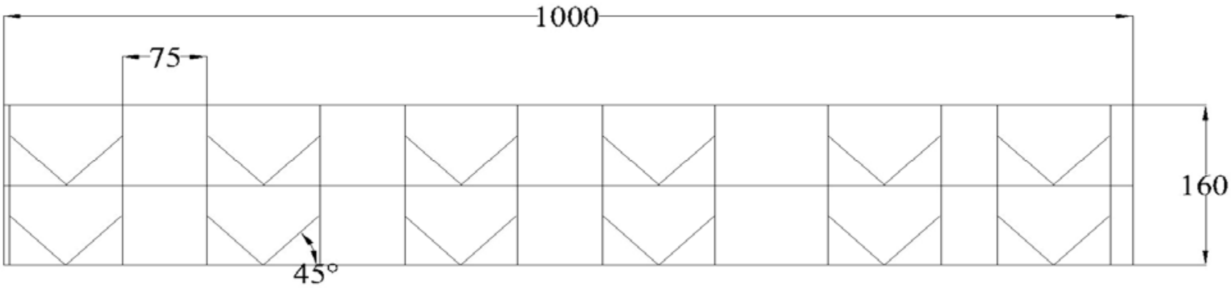

Figure 6. Roughness geometry P = 75mm, angle = 45°

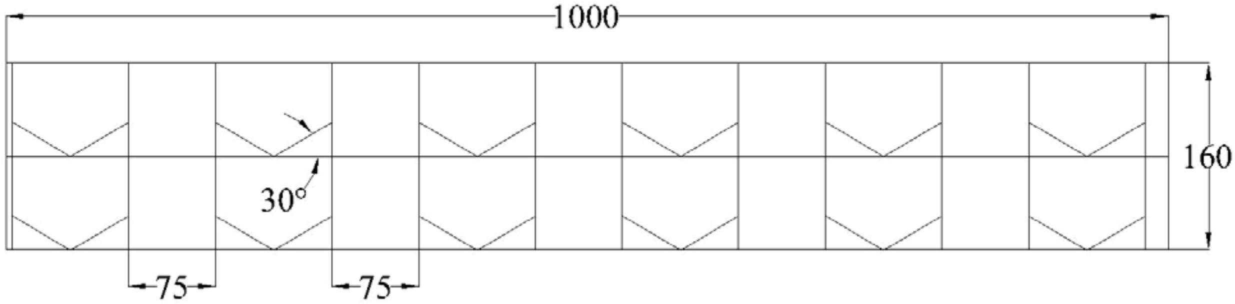

Figure 7. Roughness geometry P = 75mm, angle = 30°

3.2 Data reduction

The mass flow rate (m) of air has been calculated with the help of measurement of area of an orifice meter and pressure drop across the orifice meter. Following equation has been used to calculate mass flow rate.

$$m = C_d A_o \left[\frac{2\rho \Delta P_o}{1-\beta^4} \right]^{0.5} \tag{1}$$

The heat transfer rate to air (Q_u) has been calculated from equation (2) as mention below:

$$Q_u = mC_p(T_o - T_i) \tag{2}$$

The heat transfer co-efficient has been calculated from equation (3) as given below:

$$h = \frac{Q_u}{A_p(T_{pm} - T_{fm})} \tag{3}$$

Nusselt number has been calculated from equation (4) as given below:

$$Nu = \frac{hD}{k} \tag{4}$$

Thermal efficiency is calculated from equation (5) as given below:

$$\eta_{th} = mC_p(T_o - T_i)/(IA_p) \tag{5}$$

The friction factor has been calculated from equation (6) as given below:

$$f = \frac{2\Delta P_d D}{4\rho L V^2} \tag{6}$$

In order to investigate the effect of roughness parameters on thermohydraulic efficiency, effective efficiency is calculated from equation (7).

$$\eta_e = (Q_u - P_m / C)/ IA_c \tag{7}$$

where C (= $\eta_F \times \eta_M \times \eta_{Tr} \times \eta_{Th}$) is the conversion factor accounting for net conversion efficiency from thermal energy of the resource to mechanical energy.

3.3 Validity test

Figures 8 and 9 have shown the comparison between predicted values calculated from correlations and the experimental values. The Nusselt number and friction factor correlations have been given by Dittus – Boelter and Modified Blasius for smooth duct.

Correlation for Nusselt number of smooth duct [28]

$$Nu_s=0.023Re^{0.8}Pr^{0.4}(10^4\leq Re\leq 1.24\times 10^5) \tag{8}$$

Correlation for Friction factor of smooth duct [28]

$$f_s=0.085Re^{-0.25}(4\times 10^3\leq Re\leq 10^5) \tag{9}$$

The absolute deviation between predicted and experimental values of Nusselt number and friction factor were ± 6.7% and ± 3.3 % respectively. Hence experimental values are in good agreement with predicated values.

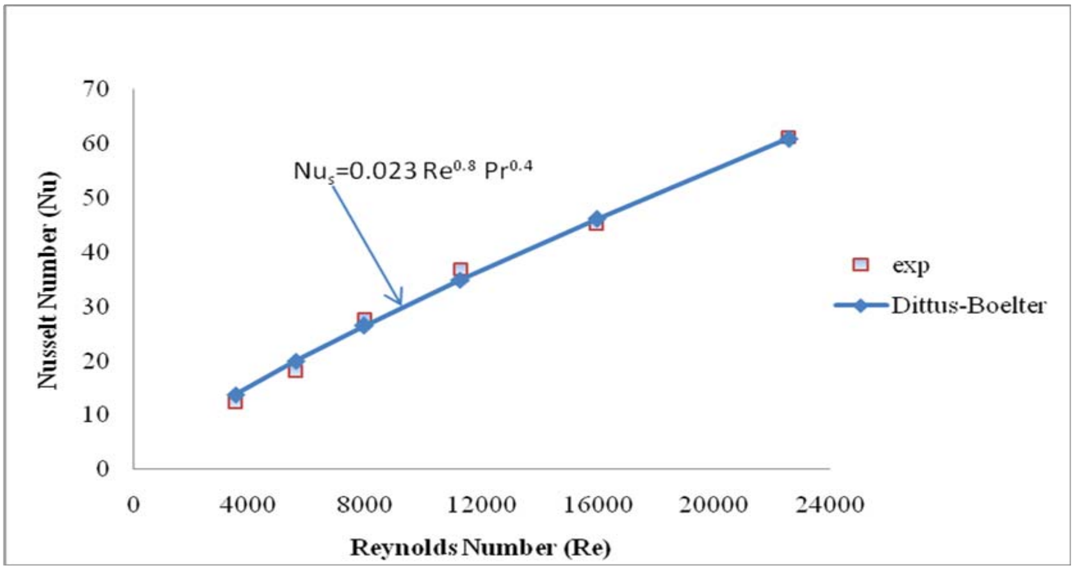

Figure 8. Comparison between predicted and experimental values of Nusselt number

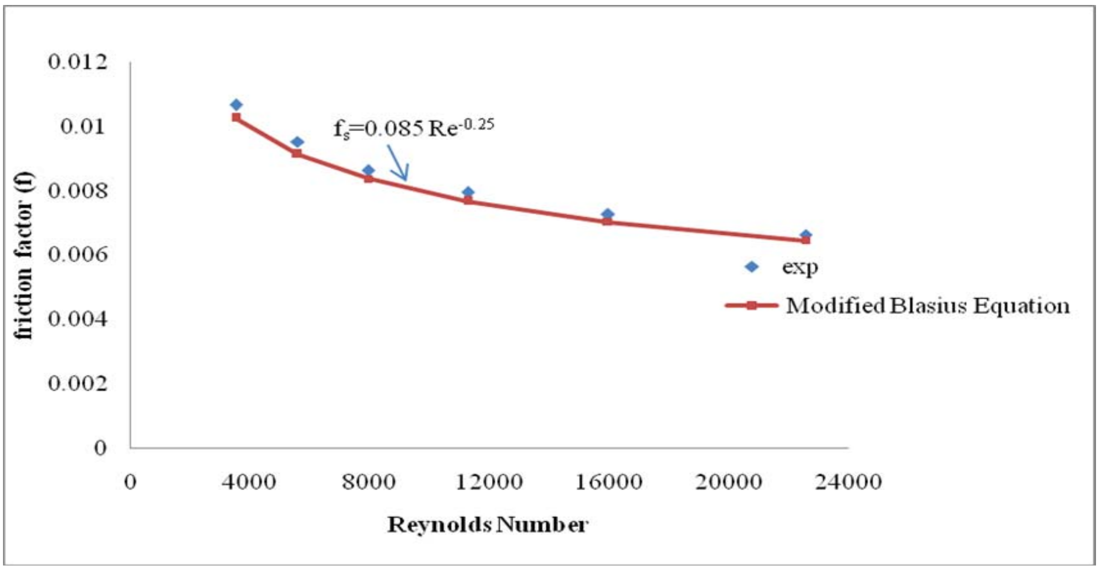

Figure 9. Comparison between predicted and experimental values of friction factor

4. Results and discussion

4.1 Effect of Reynolds number on Nusselt number and friction factor

It has been seen that Nusselt number increase with the Reynolds number while friction factor is decreasing with Reynolds number [29].

4.2 Thermal efficiency

The thermal efficiency of roughened as well as smooth absorber plate solar air heaters has been computed on the basis of first law of thermodynamics. The relative roughness height (e/D) and relative roughness pitch (P/e) are considered as strong parameter of roughness element. The performance of solar air heaters with different roughness and operating parameters have been evaluated and presented in Figure 10. The performance lines have been drawn for solar air heater having M shape continuous roughness geometry. Results show that for given values of roughness parameters, similar trend in variation of thermal efficiency is obtained with the Reynolds number. The thermal efficiency increases with the increase of Reynolds number up to certain extent but after that it starts to stabilize.

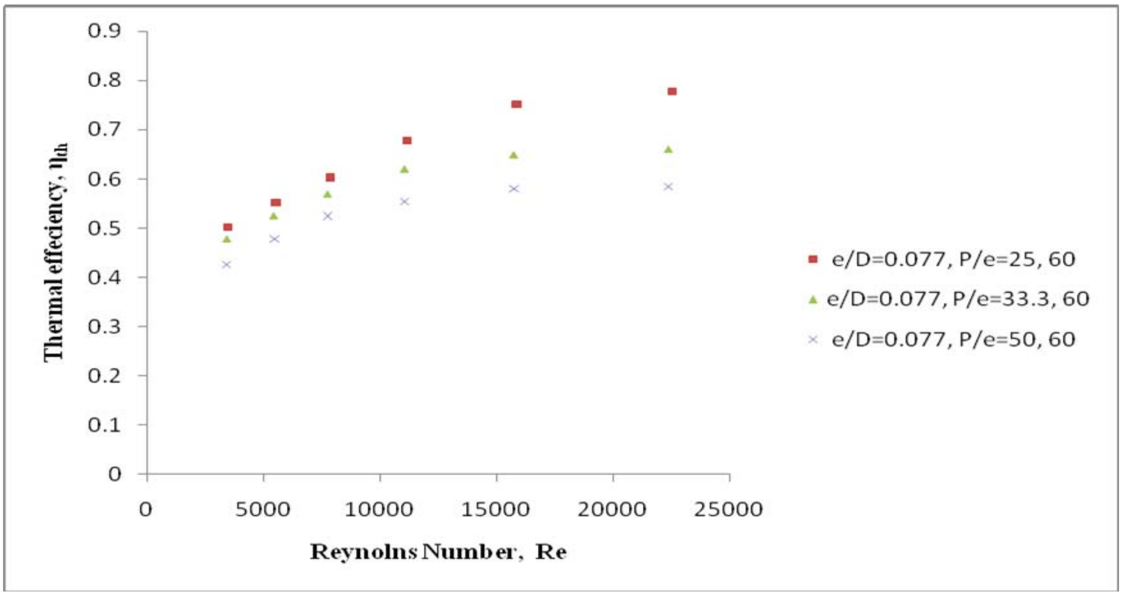

Figure 10. Variation of thermal efficiency with Reynolds number (Re)

4.2.1 Effect of relative roughness height (e/D)

Figure 11 shows the behavior of three different relative roughness heights (e/D) for same value relative roughness pitch (P/e) 50 and angle of attack (α) 60°. It is seen that as the relative roughness height increases, Thermal efficiency increases. But if it exceed beyond the height of boundary layer then it is not helpful in making the reattachment point. So, after this height thermal efficiency has no affect and ribs will act as fins.

4.2.2 Effect of Relative Roughness Pitch (P/e)

Figure 12 shows the variation of Thermal efficiency with relative roughness pitch (P/e) parameters for considered geometry. It has been observed that with increase in the value of P/e, Thermal efficiency of considered geometries decreases. This is explained on the basis of flow separation. For higher P/e the flow which separates after each rib does not reattach before it reaches the succeeding rib. The maximum thermal efficiency occurred at (P/e) 25 at higher Reynolds number.

4.2.3 Effect of angle of attack (α)

The variation of Thermal efficiency with the angle of attack for different Reynolds number is shown in Figure 13. It has been observed that the initially Thermal efficiency increase and then decreases with angle of attack and maximum value of it lies at 60°. The reason for this phenomenon may be separation of flow resulting from the presence of M shape continuous ribs.

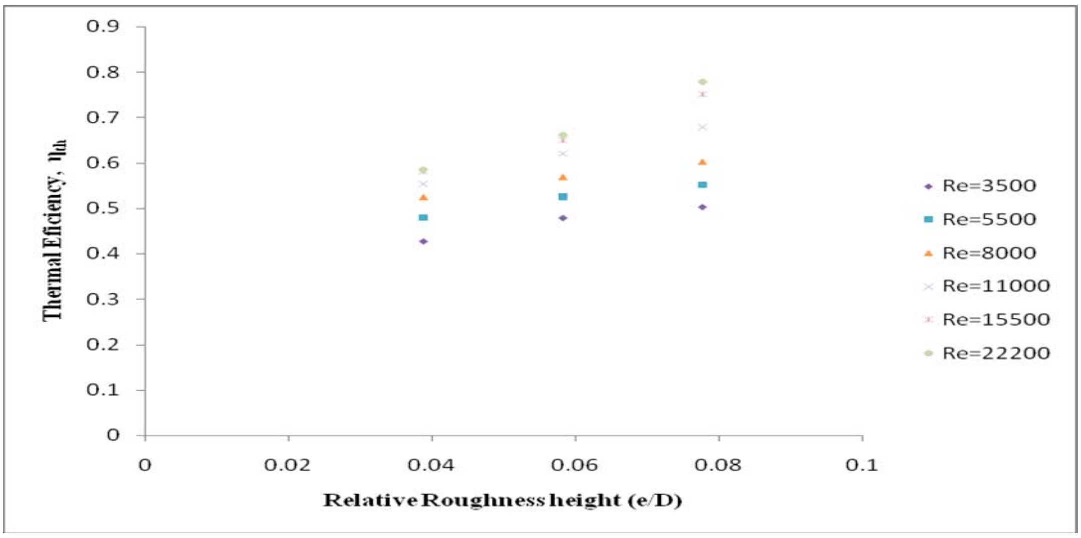

Figure 11. Effect of relative roughness height (e/D) on thermal efficiency

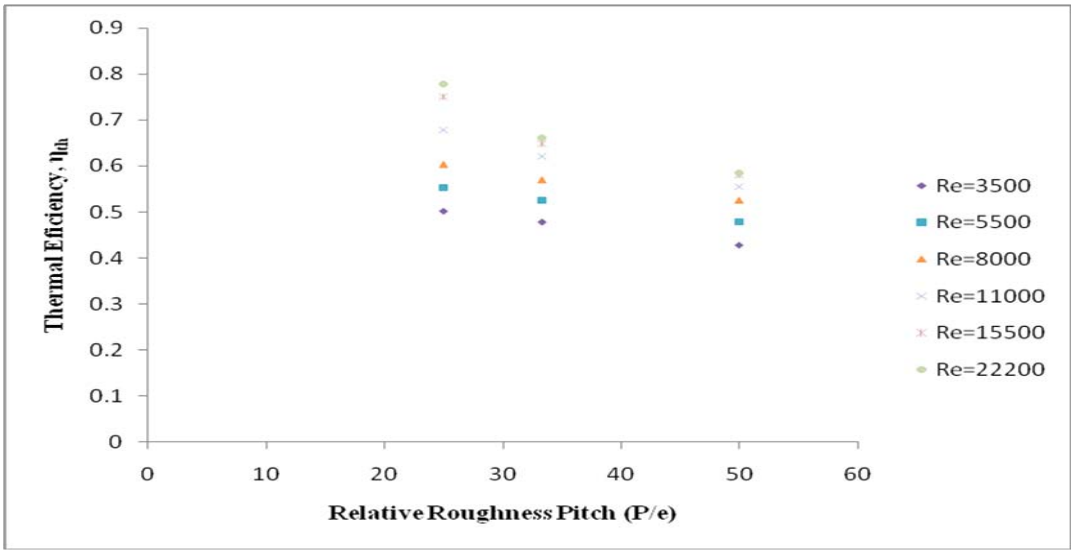

Figure 12. Effect of relative roughness pitch (P/e) on thermal efficiency

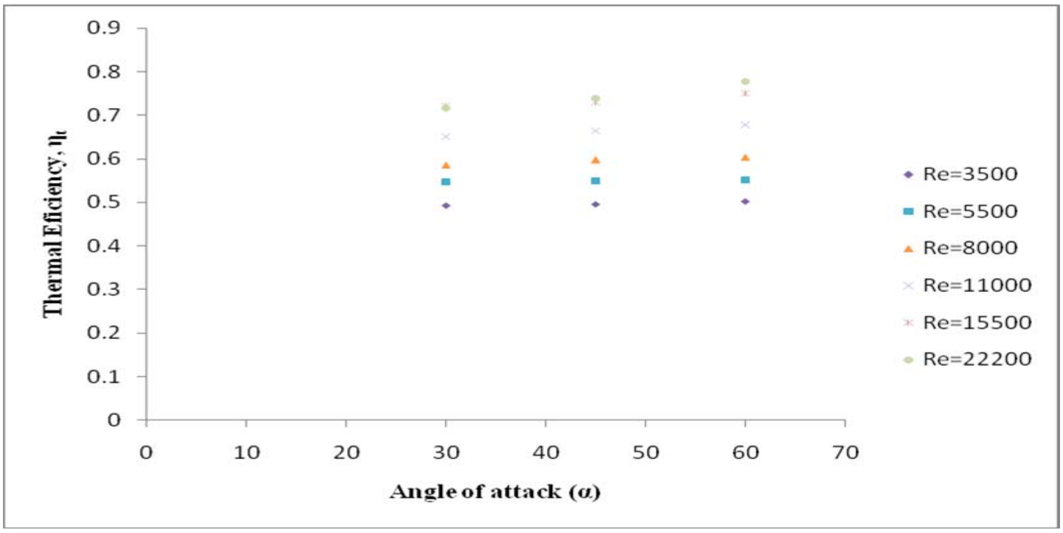

Figure 13. Effect of angle of attack (α) on thermal efficiency

4.3 Thermohydraulic performance

Performance of the system based on the consideration of thermal as well as hydraulic characteristics is termed as thermohydraulic performance of solar air heater. It is necessary to take into account the electrical energy required for pumping, while calculating the performance of solar air heater. Cortes and Piacentini [30] evaluated the thermohydraulic performance of unglazed solar air heaters having artificial roughness on the absorber plates. It is also named as effective efficiency.

The effective efficiency (η_e) of roughened as well as smooth absorber plate solar air heaters has been computed on the basis of method proposed by Cortes and Piacentini [30]. The relative roughness height (e/D) is considered as strong parameter of roughness element for performance evaluation. Figure 14 has been prepared to show the effect of relative roughness height on effective efficiency for the optimal values of other parameters. It is evident from these figures that solar air heaters provided with rib roughness results in better effective efficiency than the smooth duct. It can be observed from this figure that for given values of roughness parameters, similar trend in variation of effective efficiency is obtained with Reynolds number. Initially, effective efficiency increases with the increase of Reynolds number, attains maximum and thereafter it starts decreasing. This is due to the fact that the quality of collected heat decreases and pump work increases. It is also observed that effective efficiency corresponding to higher values of roughness height is better in lower range of Reynolds number; however value of effective efficiency is reversed in higher range of Reynolds number. The reason for the fact that the Reynolds number corresponding to maximum effective efficiency shifts to a lower value as e/D increases, may be attributed to the decrease of the rate of useful energy collected as the relative roughness height increases. The rate of useful energy collected decreases, whereas the friction losses rise with increasing e/D causing increased energy consumption.

Figures 15-17 illustrates that behavior of thermohydraulic performance of roughened absorber plate for different operating parameters. With the help of roughened absorber effective efficiency can be achieved 60-65% and thermal efficiency up to 80-85%.

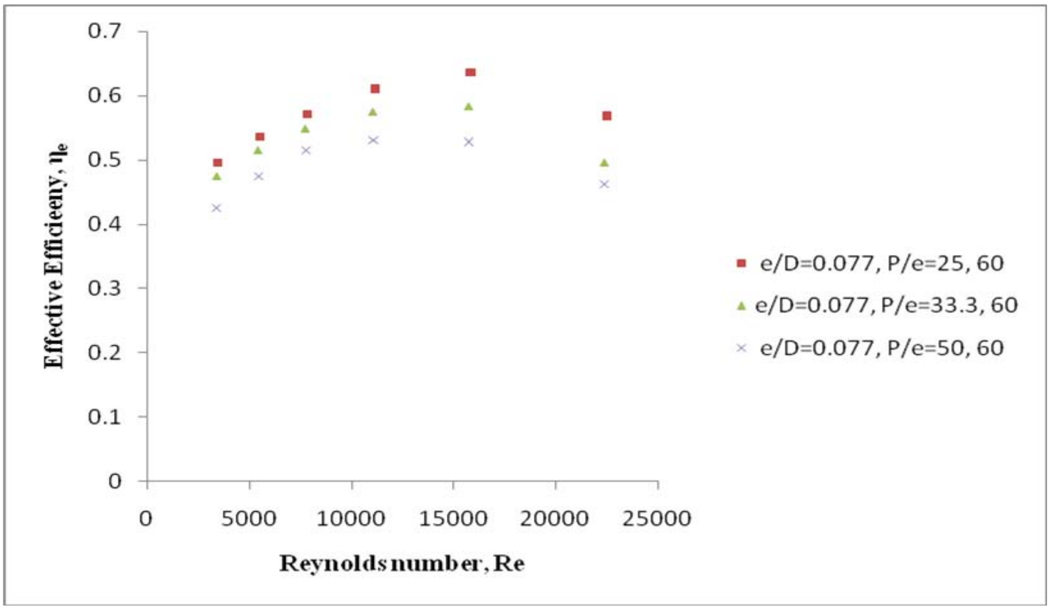

Figure 14. Variation of effective efficiency with Reynolds number

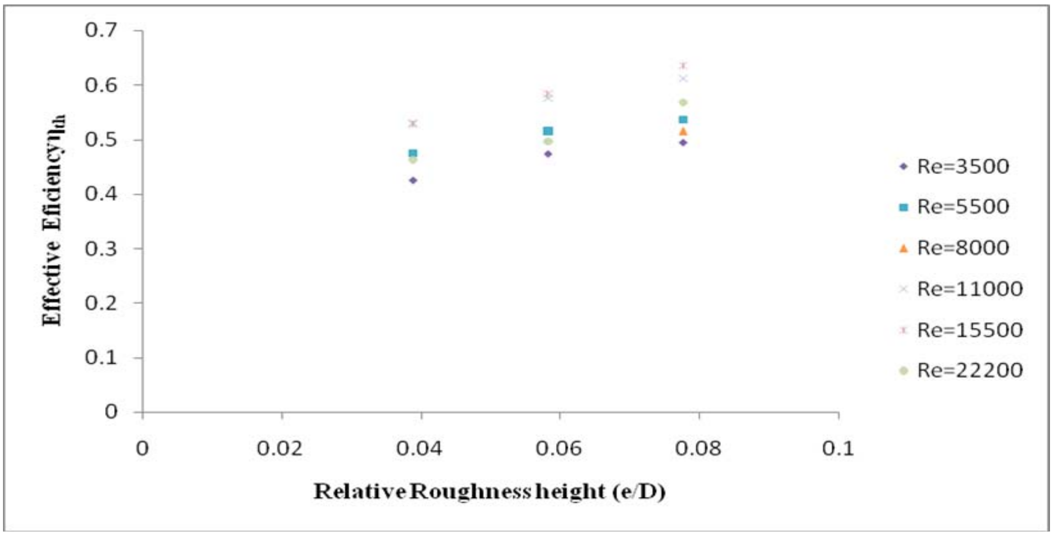

Figure 15. Effect of relative roughness height (e/D) on effective efficiency

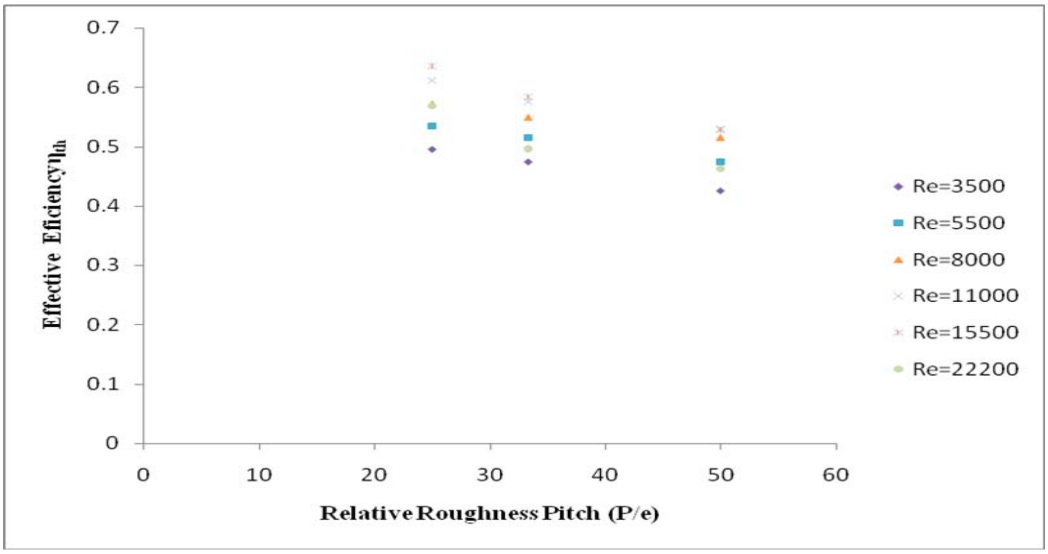

Figure 16. Effect of relative roughness pitch (P/e) on effective efficiency

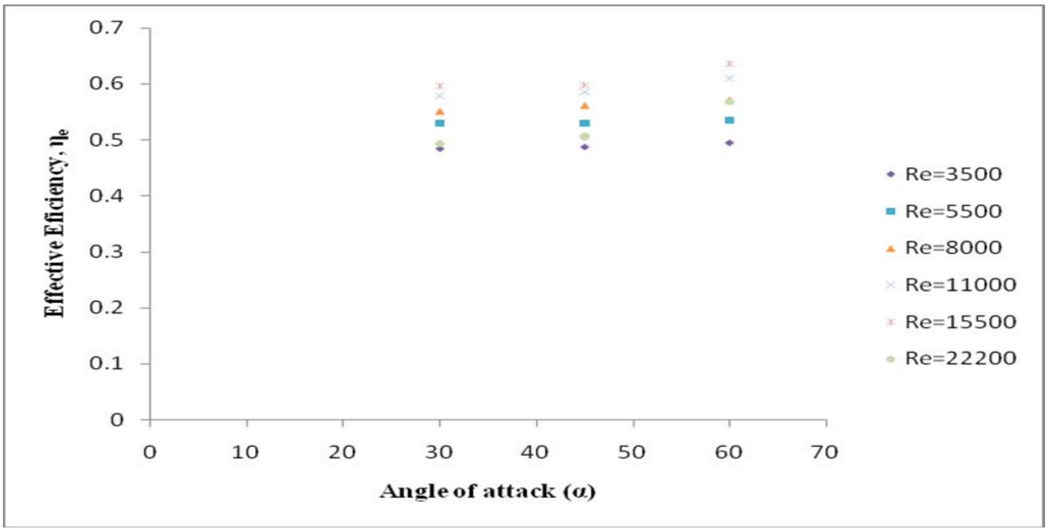

Figure 17. Effect of angle of attack (α) on effective efficiency

5. Conclusion

The major findings of thermal and thermohydraulic performance of study of artificial roughened solar air heater having M shape continuous ribs as roughness geometry on absorber plate are:

1. There is considerable enhancement in thermal efficiency from 60% to 85% of artificial roughened solar air heater due to enhancement in Nusselt number over smooth duct.
2. The artificial roughened solar air heater also causes 6-7 times enhancement in friction factor.
3. The enhancements in the Nusselt number, friction factor and thermal efficiency have been observed strong function of the relative roughness height (e/D). The greatest enhancement found for solar air heater having high relative roughness value.
4. Initially, the effective efficiency increase with the increase in the Reynolds number, attains the maximum value depending on roughness parameters and thereafter it start decreasing because the pumping power start increasing with cube of the flow velocity.

Nomenclature

A_p	surface area of Absorber plate (m^2)
A_C	cross sectional area of rectangular duct (m^2)
A_o	area of orifice meter (m^2)
C_p	specific heat of air (J/kg K)
C_d	coefficient of discharge
D	hydraulic diameter (m)
D_o	diameter of pipe (m)
W	width of duct (m)
H	height of duct (m)
W/H	aspect ratio of duct
L	length of test section in duct (m)
e	rib height (m)
e/D	relative roughness height
P	pitch (m)
P/e	relative roughness pitch
f	fanning friction factor
k	thermal conductivity (W/m K)
V	mean flow velocity in duct (m/s)
u_t	friction velocity (m/s)
h	heat transfer coefficient (W/m^2K)
Nu	Nusselt number
Re	Reynolds number
ΔP_o	pressure drop across orifice plate (Pa)
m	mass flow rate of air (kg/s)
Q_u	useful heat gain (W)
T_o	air outlet temperature (K)
T_i	air inlet temperature (K)
T_{pm}	mean temperature of absorber plate (K)
T_{fm}	mean temperature of air (K)
Greek Symbols	
δ	thickness of laminar sublayer (m)
ν	kinematic viscosity of fluid (m^2/s)
α	angle of attack (°)
μ	dynamic viscosity (Ns/m^2)
ρ	density of air (kg/m^3)
ρ_o	density of fluid in U-tube manometer (kg/m^3)
β	ratio of orifice diameter to pipe diameter

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