International Journal of Mechanical and Industrial Engineering

Volume 3 | Issue 1 Article 10

July 2013

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THERMOHYDRAULIC PERFORMANCE OF AN EQUILATERAL TRIANGULAR DUCT WITH ARTIFICIAL ROUGHNESS USED IN SOLAR AIR HEATER

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Abstract—The thermohydraulic performance of artificially roughened equilateral triangular solar air heater duct has been investigated and the comparision of the same has been presented with that of a conventional smooth solar air heater duct. The range of relative roughness height (e/D_h) is from 0.021 to 0.043, value of angle of attack (α) and relative roughness pitch (p/e) has been 30° and 8 respectively. The range of Reynolds number is from 5600 to 28000 and aspect ratio of the duct is 1.15. It has been found that the thermohydraulic performance of artificially roughened triangular solar air heater duct is always more than that of the smooth absorber plate in the range of Reynolds number investigated.

Keywords*-Thermohydraulic performance; Artificial roughness; Solar air heater; Triangular duct.*

I. INTRODUCTION

Energy plays a very crucial role in the life of every one either it is the matter of economic growth or the growth of industrialization [1]. As the population goes on increasing, the demand for energy also goes on increasing day by day. The depletion of the fossil fuels has been started due to the continuous use of fossil fuels for energy generation. On the other hand, energy generation with the help of fossil fuels creates lot of pollution which in turn deteriorates the life cycle of the planet Earth [1]. Development in the field of renewable energy resources work as the catalyst for the energy generation. Use of renewable energy resources for energy generation is free from pollution. Among all the available renewable resources the sun is of dominating nature.

The easiest methodology for making the proper use of solar energy is its conversion to thermal energy using solar collector. These solar collector are the part of solar air heater and solar water heater which are used for heating air and water respectively. Fabrication of solar air heater is quiet easy due to its compactness. Moreover, solar air heater are cheaper as compared to solar water heater due to use of less material. Solar air heater has wide area of application such as curing of industrial products, space heating, crop drying, wood seasoning, etc. Efficiency of solar air heater is low due to the low specific heat of air and low heat transfer rate between absorber plate and flowing air inside duct. So, to make solar air heater efficient it is necessary to enhance the heat transfer rate.

Artificial roughness provides the turbulence to the flow which leads to increase the heat transfer between the air and the heated wall. Roughness is created in such a way that it breaks the laminar sublayer region i.e. near the wall. There are several method to provide artificial roughness on the absorber plate such as casting, forming, machining, blasting, welding ribs and/or fixing thin circular wires, etc [2-4]. The easiest and cheapest way of providing artificial roughness on the underside of the absorber plate is sticking of ribs.

Bharadwaj and Varun [5] experimentally investigated that in an artificially roughened equilateral triangular solar air heater duct maximum heat transfer and friction factor occur at relative roughness height of (e/D_h) of 0.043. Mittal and Varun [6] experimentally observed that maximum thermohydraulic performance in case of discrete inclined ribs and transverse full ribs occur at relative roughness pitch (p/e) of 8. Verma and Prasad [7] carried out an experimental study for the optimization of thermohydraulic performance. They kept the range of Reynolds number from 5000-20000, relative roughness height (e/D_h) 0.01-0.03 and relative roughness pitch (p/e) 10-40. They found that for the given range of parameters optimal thermohydraulic performance comes to be 71%. Gupta et al. [8] experimentally investigated the effect of relative roughness height (e/D_h), angle of attack (α) and Reynolds number on the thermohydraulic performance. They found that with the increase in relative roughness height (e/D_h), the value of Reynolds number decreases for which thermohydraulic performance was maximum. They found that best thermohydraulic performance was reported at relative roughness height (e/D_h) of 0.023 and Reynolds number of 14000. Various investigators like Prasad and Mullick [9], Prasad and Saini [10-11], Sahu and Bhagoria [12], Aharwal et al. [13], Varun et al.[14] utilized different types of rib roughness on the absorber plate of the solar air heater to increase the heat transfer coefficient.

However, any method adopted to increase the heat transfer is always accompanied by an increase in frictional losses in the duct. Therefore, investigation of the thermohydraulic parameter is necessary in order to find the pumping power required to force the air through the duct.

In this investigation, circular ribs of relative roughness height (e/D_h) 0.021-0.043 are attached to one side of the absorber at inclination of 30° and relative roughness pitch (p/e) of 8.

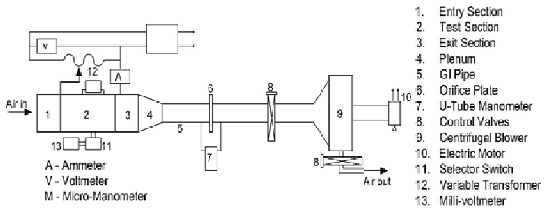
II. CONCEPT OF ARTIFICIAL ROUGHNESS

Surface roughness is one of the first active techniques to be considered for the augmentation of forced convection heat transfer. It is necessary that the flow near the heat transfer surface should be turbulent so as to attain higher coefficient of heat transfer. However, energy for creating such turbulence has to come from the fan or blower and the excessive turbulence leads to excessive power requirement to make the air flow through the duct.

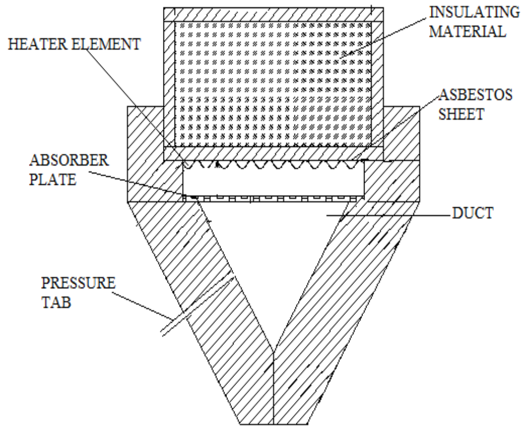
Hence, it is necessary that the turbulence must be created in the vicinity of heat transfer surface i.e. laminar sublayer only where the heat exchange takes place and the flow should not be unduly disturbed so as to avoid excessive friction losses. This can be done by keeping the height of the roughness element to be small in comparison with the duct dimensions.

III. EXPERIMENTAL SET-UP

The schematic and sectional view of the equilateral triangular duct is shown in Fig. 1 (a and b). The duct is fabricated from wooden planks of different cross-sections. The inner dimension of the duct is 2300 mm × 160 mm × 138 mm and the aspect ratio (W/H) of the duct is 1.15. The length of the test-section is 1000 mm. The entry and exit length are kept equal. Consequently, the flow can be assumed to be fully developed turbulent flow in the entire length of the test section. Hence, the entry and exit lengths are 650 mm and 650 mm respectively. Entry and exit length are required in order to minimize the end effects in the test section. An electric heater of size 1000 mm × 160 mm was fabricated using series and parallel loops of nichrome wires. Heater was designed in such a way that it can provide maximum heat flux upto 800 W/m^2. Heat flux may be varied using a variac which is connected across an electric heater. 60 mm thick glasswool layer and 12 mm thick wooden planks are provided on the back side of the heaters to minimize the heat losses from the top side of the heater.



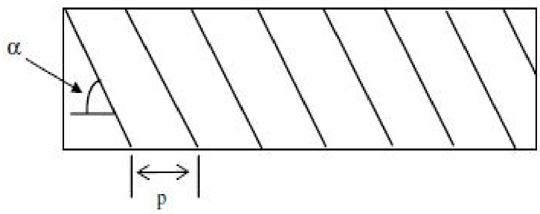
(a)



(b)

Figure 1. (a) Schematic diagram of the set-up (b) Sectional view of the duct

The roughness was produced by fixing the ribs of different diameter at 30° inclination and relative roughness pitch (p/e) of 8 on the underside of the absorber plate. The schematic and pictorial view of the absorber plate is shown in Fig. 2 (a and b). At the end of the duct, a plenum was provided to connect the triangular duct to a circular pipe.



(a)

(b)

Figure 2. (a) Schematic diagram of the absorber plate (b) Pictorial view of the absorber plate

Ambient air is sucked through the duct by means of a centrifugal blower driven by 1.5 kW, three phase, 230 V and 2820 rpm motor. For precise control of the air flow rate through the system, two gate valves are provided, one on the inlet side and other on the outlet side of the blower.

IV. INSTRUMENTATION

A. Temperature Measurement

Calibrated copper-constant (T type), thermocouples were used for the temperature measurement of air and the absorber plate. 12 Thermocouples were mounted on the absorber plate to measure its mean temperature. The location of thermocouples on the absorber plate is shown in Fig 3. 2 thermocoples were inserted at inlet and outlet section of the duct in order to measure the temperature of air.

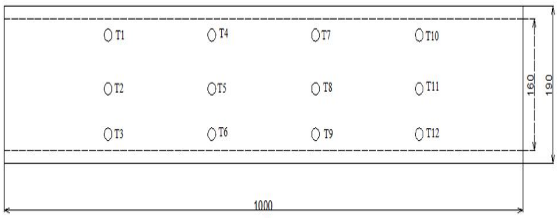


Figure 3. Location of thermocouples on the absorber plate

B. Measurement of Air flow

The air flow rate through the duct was measured by using concentric orifice plate with 45° beveled edges which was designed, fabricated and fitted in the 80 mm pipe carrying the air from plenum to the blower. The orifice plate was calibrated against pitot tube and the value of coefficient of discharge (C_d) was determined as 0.612 and it is used to measure mass flow rate of the air. Pressure drop across the orifice meter was measured by means of a U-tube manometer.

C. Pressure Drop Measurement

The pressure drop across the test section of the duct was measured with the help of a micro-manometer having a least count of 0.01 mm. The micro-manometer consists of a movable reservoir, a fixed reservoir and a transparent tube connected to these reservoirs through flexible tubing. The movable reservoir is mounted using a lead screw having a pitch of 1.0 mm and a graduated dial having a 100 division; each division showing a movement of 0.01 mm of the reservoir. The two reservoirs were connected with the air traps of the duct through flexible tubes. The meniscus is maintained at fixed prescribed mark by moving the reservoir up and down and the movement is noted, which yields the pressure difference across the two pressure tapping.

V. EXPERIMENTAL PROCEDURE

The measuring equipments i.e. orifice meter, milli-voltmeter and U-tube manometers were properly checked. The micro-manometer and the U-tube manometer were properly leveled by means of spirit level. Thermocouples were tested to insure that all thermocouples yield the same output corresponds to ambient temperature before heater has been switched on. Pressure tapings and the tubes were cleaned and checked for leakage and blockage before each test run. After proper checking of instruments the set up was used for conducting experiment. The power supply to the centrifugal blower and the electric heater was switched on and the desired flow rate was set with the help of control valves.

The experimental runs were conducted under quasi-steady state to collect the relevant data for heat transfer and friction factor. The setup was allowed to attain quasi-steady state before the data was recorded at different mass flow rates. The following data were recorded:

1. Air temperature at different points on the duct and the temperature of absorber plate at 12 different locations on the absorber plate.
2. Pressure drop across the test section.
3. Pressure measurement across the orifice meter.

Before starting the experimentation for the roughened duct , the set-up was checked by conducting the experiment for a smooth duct. The Nusselt number and friction factor calculated from the experimental data for smooth duct have been compared with the predicted value of the Nusselt number and friction factor respectively obtainted from Dittus-Boelter equation [14] and Modified Blasius equation [15]

Dittus-Boelter equation

$$Nu_s = 0.024\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \qquad (1)$$

Modified Blasius equation :

$$f_s = 0.085\,\mathrm{Re}^{-0.25} \qquad (2)$$

VI. DATA REDUCTION

The raw data have been reduced to obtain mean plate temperature, mean air temperature, mass flow rate and Reynolds number. These data were than used to determine heat transfer coefficient, Nusselt number and friction factor. Relevant expressions for the computation of above mentioned parameters and some intermediate parameters have been given below;

Mass flow rate of air has been determined from the pressure drop measurement across the calibrated orifice meter using the following expression.

$$m = C_d A_o \sqrt{\frac{2\rho(\Delta P_o)}{1-\beta^4}} \qquad (3)$$

The heat transfer coefficient is calculated from the relationship given below;

$$h = \frac{Q_u}{A_p\left(T_{pm} - T_{fm}\right)} \qquad (4)$$

Where heat transfer rate (Q_u) to the air is given by

$$Q_u = mC_p(T_o - T_i) \qquad (5)$$

The heat transfer coefficient calculated using equation (4) is then used to determine the Nusselt number as given below;

$$Nu = \frac{hD_h}{k} \qquad (6)$$

Where D_h is the hydraulic diameter of the duct.

The friction factor is determined from the measured values of pressure drop $(\Delta P)_d$ across the test section length using Darcy Wiesbach equation as below,

$$f = \frac{2(\Delta P)_d D_h}{4\rho L V^2} \qquad (7)$$

VII. THERMOHYDRAULIC PERFORMANCE

Thermohydraulic performance is the performance of the system that includes the consideration of thermal as well as hydraulic characteristics. It is necessary to take electrical energy required for pumping into account, while evaluating the performance of solar air heater. From second law of thermodynamics considerations, the power from the thermal output of the collector, loosing always considerable part of the energy in conversion and transmission. Therefore, the pumping power required is converted to equipment thermal energy to obtain evaluated the real performance of the collector in terms of the effective efficiency that taken into account the useful thermal gain and equivalent thermal energy that will be required to provide corresponding mechanical energy for overcoming friction power losses, and is given by:

$$\eta_{eft} = \frac{Q_u - \frac{P_m}{C}}{IA_p} \qquad (8)$$

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

η_F = Efficiency of fan or blower.

η_M = Efficiency of the electrical motor used for driving fan.

η_{Tr} = Efficiency of electrical transmission.

η_{Th} = Thermal conversion efficiency of power plant.

The rate of useful thermal energy gain for roughened solar air heater can be calculated by using the following expression:

$$Q_u = mC_p(T_o - T_i) \qquad (9)$$

The mechanical power required to propel the air through the solar air heater duct is the product of volume flow rate and the pressure drop ΔP_d across the duct and is given by:

$$P_m = VA_c(\Delta P_d) \qquad (10)$$

The effective efficiency has been evaluated by using equation (8) for three set of roughness plates. The thermohydraulic performance has been obtained by varying the Range of Reynolds number along with the values of geometrical and operating parameters as given in table I.

TABLE I. Values of geometrical and operating parameters

Parameters	Values
Length	1 metre
Width	160 mm
Reynolds number	5600-28000
Relative roughness height (e/D_h)	0.021-0.043
Relative roughness pitch (p/e)	8
Angle of attack (α)	30°
Intensity	800 W/m^2

VIII. RESULT AND DISCUSSION

In this section of the paper variation of the Nusselt number and thermohydraulic performance have been presented and discussed.

A. *Effect of Reynolds number on Nusselt number*

Fig. 4 shows the variation of Nusselt number as a function of relative roughness height (e/D_h) for various values of the Reynolds number and for fixed value of relative roughness pitch (p/e) of 8 and angle of attack (α) of 30°. It can be clearly seen from the fig. 4 that the Nusselt number increases with the increase in Reynolds number.

Replotting fig. 4 we get fig. 5. Fig. 5 shows the variation of Nusselt number as a function of Reynolds number for different value of relative roughness height (e/D_h) and for fixed value of relative roughness pitch (p/e) of 8 and angle of attack (α) of 30°. The value of the Nusselt number increase with the value of relative roughness height (e/D_h). The maximum Nusselt number occur at 0.043 and minimum at 0.021 as reported by Bharadwaj and Varun [2].This is due to the reason that relative roughness height should be sufficient in order to break the laminar sublayer so that maximum heat transfer takes place. It can also be seen that roughened plate gives the better enhancement in the heat transfer than that of the smooth plate.

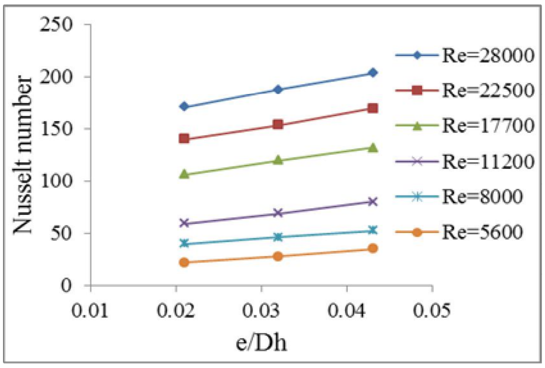


Figure 1. Variation of the Nusselt number as a function of Reynolds number for different values of relative roughness height and for fixed relative roughness pitch and angle of attack.

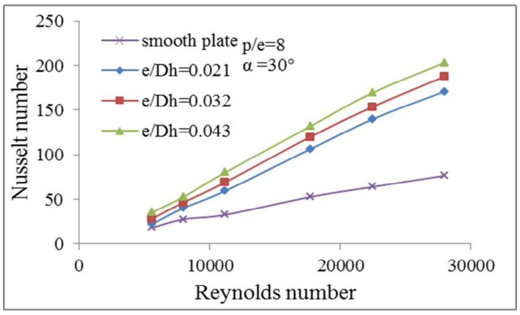


Figure 5. Variation of the Nusselt number with the Reynolds number for different values of relative roughness height and for fixed value of angle of attack and relative roughness pitch.

B. *Effect of Reynolds number on Effective Efficiency*

It has been found that there is a considerable enhancement in the heat transfer by providing the inclined ribs on the absorber plate of the equilateral triangular solar air heater duct. This enhancement in the heat transfer is accompanied by an ample increase in friction factor. It is evident from the fig. 6 that roughened solar air heater duct results in better effective efficiency than the smooth duct. It can also be seen from the figure that effective efficiency increases with the increase in Reynolds number and after a certain value it starts decreasing. This is due to the reason that the quality of collected heat decreases and pump work increases.

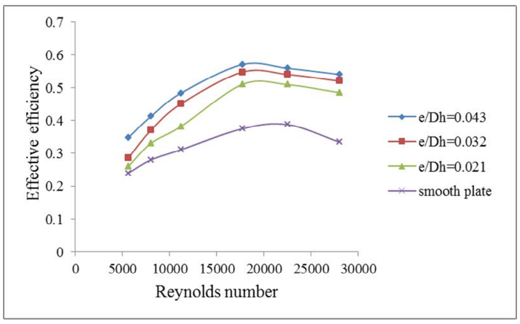


Figure 6. Effect of Reynolds number on Effective Efficiency

IX. CONCLUSION

It is concluded that the presence of the inclined ribs on the absorber plate of a equilateral triangular solar air heater duct enhance the heat transfer rate. The maximum heat transfer and thermohydraulic performance of the roughened solar air heater duct occur at the relative roughness height (e/D_h) of 0.043.

NOMENCLATURE

A_c	Area of the flow (m^2)
A_o	Throat area of the orifice (m^2)
A_p	Area of the absorber plate (m^2)
C_d	Coefficient of discharge for the orifice meter
C_p	Specific heat of air (kJ/kg/K)
D_h	Hydraulic diameter of the duct (m) (= 4×0.5WH/(3W))
e	Height of the roughness element (m)
f_r	Friction factor for roughened absorber plates
f_s	Friction factor for the smooth absorber plate
H	Height of the duct (m)
h	Average heat transfer coefficient ($W/m^2/K$)
k	Thermal conductivity (W/m/K)
L	Length of the absorber plate (m)
m	Mass flow rate (kg/s)
Nu	Nusselt number for the roughened plates
Nu_s	Nusselt number for the smooth plates
P	Roughness pitch (m)
e/D	Relative roughness height
P/e	Relative roughness pitch
Pr	Prandtl number
ΔP_o	Pressure drop across the orifice meter (N/m^2)
ΔP_d	Pressure drop across the test section (N/m^2)
Re	Reynolds number
W	Width of the duct (m)
D_1	Diameter of orifice (m)
D_2	Diameter of pipe (m)
t_i	Inlet temperature of air (°C)
t_o	Outlet temperature of air (°C)
T_{fm}	Average temperature of air (°C)
T_{pm}	Average temperature of the absorbing plate (°C)
ρ	Density of fluid (kg/m^3)
β	Diameter ratio, D_2/D_1
α	Angle of attack
η_{eft}	Effective efficiency

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