EFFECT OF ELLIPTICAL INCLINED DIMPLES ON NUSSELT NUMBER AND FRICTION FACTOR FOR ARTIFICIALLY ROUGHENED DUCT OF SOLAR AIR HEATERS

Raman Kumar, Harvinder Lal

Address for Correspondence
Department of Mechanical Engineering, Ramgarhia Institute of Engineering and Technology (RIET) Phagwara, Punjab, India

ABSTRACT
An experimental investigation has been carried out for a range of system and operating parameters in order to analyze effect of artificial roughness on heat transfer and friction in solar air heater duct having inclined dimples as roughness geometry. An increase in heat transfer and friction loss has been observed for duct having roughened absorber plate. Experimental data have been used to develop Nusselt number and friction factor correlations as function of system and operating parameters for predicting performance of the system having investigated type of roughness geometry. The aspect ratio of the duct is of order (W/H = 10) and the range of Reynolds number is (8000 – 25000).

KEY WORDS: Solar air heater, artificial roughness, heat transfer coefficient, Nusselt number and friction factor. India

INTRODUCTION
Energy is a basic need for human being; it is a prime agent in the generation and economic development. Energy resources may be classified in two ways conventional and non-conventional energy resources. Solar energy is available abundance on earth in the form of radiation.

Solar energy is used for heating application and converts it into thermal energy. Solar air heater is the cheapest way of converting solar energy into thermal energy. Solar air heaters form the major component of solar energy 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 collection devices. These have found several applications including space heating and crop drying. 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 which increases the absorber plate temperature, leading to higher heat losses to the environment resulting in low thermal performance. Thermal Performance of solar air heaters is comparably poor from solar water heaters. Thermal performance may be increased by increasing convective heat transfer coefficient.

There are two way for increasing heat transfer coefficient either increase the area of absorbing surface by using fins or create the turbulence on the heat transferring surfaces. Turbulence can be increased by artificial roughness. Artificial roughness is basically a passive heat transfer enhancement technique by which thermo hydraulic performance of a solar air heater can be improved. The artificial roughness has been used extensively for the enhancement of forced convective heat transfer, which further requires flow at the heat transferring surface to be turbulent. However, energy for creating such turbulence has to come from the fan or blower and the excessive power is required to flow air through the duct. Therefore, it is desirable that the turbulence be created only in the region very close to the heat transferring surface, so that the power requirement may be lessened. Several investigations have been carried out to study the effect of artificial roughness on heat transfer and friction factor for two opposite roughened surfaces and the correlations were developed by different investigators. The application of artificial roughness in the form of fine wires on the heat transfer surface has been recommended to enhance the heat transfer coefficient by several investigators. When a fluid enters in closed channel at a uniform velocity, the fluid particles in the layer in contact with wall of the channel come to complete rest. This layer also causes the fluid particles in the adjacent layers to slow down gradually as result of friction. To make up this velocity reduction, the velocity of the fluid at the midsection of the rectangular duct has to increase to keep the mass flow rate through the rectangular duct constant. As a result, velocity gradient develops along the channel. Due to roughness element lies under the absorber plate, the flow becomes turbulent because of reattachment point or brakeage of hydrodynamic boundary layer at regular interval. In this phenomenon heat transfer coefficient, friction factor and pumping power of fluid increase through this artificial roughened rectangular duct. The effect of roughness and operating parameters on the heat transfer coefficient has been examined and a comparison of performance of roughened solar air heater with that of conventional solar air heater having smooth duct has been made. The heat transfer coefficient of a solar air heater can be increased by providing artificial roughness on the underside of the absorber plate. An experimental investigation has been carried out in the present work to study the heat transfer by using inclined elliptical dimples on the underside of absorber plate of a solar air heater under simulated conditions. Artificial roughness can be produced by several methods such as by Casting, welding and wire fixation in the form of transverse continuous ribs, transverse broken ribs, inclined and V-shaped or staggered ribs; rib formation by machining process in the form of chamfered ribs, wedge shaped ribs, combination of different integral rib roughness elements and by using expanded metal mesh ribs as has been described by Bhushan and Singh [2]. Gupta [15] et al. reported effect of transverse wire roughness on heat and fluid flow characteristics for solar air heater ducts with an absorber plate having transverse wires fixed on its underside. Muluwork et al. [5] compared thermal performance of staggered discrete V-apex up and down with corresponding transverse staggered discrete ribs. Momin et al. [8] experimentally investigated effect of geometrical parameters of V-shaped ribs on heat transfer and fluid flow.
characteristics in rectangular duct used in solar air heaters. Jaurker et al. [9] experimentally investigated heat transfer and friction characteristics of rib-grooved artificial roughness. Saini and Saini [10] used expanded metal mesh to create artificial roughness on absorber plate and investigated effect of system and operating parameters on heat transfer and friction loss. Karwa et al. [16] performed experimental study to predict the effect of rib head chamfer angle ($\phi$) and duct aspect ratio on heat transfer and friction factor in a rectangular duct roughened with integral chamfered. Bhushan and Singh analyzed the effect of artificial roughness on heat transfer and friction in solar air heater duct having protrusions as roughness geometry. Varun et al. [11] presented a review on artificial roughness investigations reported in literature. Pardeep et al. [12] investigated the effect of streamlined dimples on roughened absorber plate. It has been revealed from various experimental investigations reported in literature that inclined elliptical dimples as artificial roughness is a good technique for enhancing heat transfer coefficient between absorber plate and air flowing in the duct. However, it results in friction loss, due to increase in degree of turbulence in flow regimes, which hamper economical viability of solar air heaters. In order to overcome this problem, an experimental investigation has been carried out in the present work to study the heat transfer and friction loss by using inclined elliptical dimples in the form of protruded dimples on the underside of absorber plate of a solar air heater for a range of system and operating parameters.

**EXPERIMENTAL SET-UP AND PROCEDURE**

A test rig has been designed and fabricated to generate experimental data which consists of air duct (2400 mm x 840 mm x 70 mm) made up of wooden ply board of 20 mm thickness with length of entry, test and exit sections are 900 mm, 1000 mm and 500 mm respectively. Schematic and photographic views of experimental set-up are shown in Fig. (1) and Fig. (2) Respectively. The air duct is provided with temperature and pressure drop measuring instruments along with mounted electric heater assembly, converging plenum of length 500 mm was provided on exit side of the duct. Electric heater (2400 mm x 840 mm) have been placed on top of the duct being fabricated by combining series and parallel loops of heating wire on mica sheet to get uniform heat flux. Backside of the heater was insulated with glass wool to minimize the heat loss. Variac (0-250V) was provided to control electric supply to the electric heater. Absorber plate (20 SWG GI sheet) of size 2400 mm x 880 mm have been provided with streamlined protruded dimples (artificial roughness element) on air duct side. An orifice-meter with U-tube manometer was used to measure the mass flow rate of the air. Two gate valves one at entry and other at exit of centrifugal blower has been installed for smooth controlling of flow of air through the duct. Copper-constantan thermocouples were used to measure temperature of air and absorber plate at different locations as shown in Figs. (3) and (4). Pressure drop across test section of the duct has been measured with micro-manometer. Photographic view of dimpled plate used in the present experimental investigation is shown in Fig. (5).

\[ \text{Accuracy of experimental data was verified by conducting experiments for a conventional smooth duct. Nusselt number and friction factor values were determined from experimental data and compared with the values obtained from the following Nusselt number and friction factor correlations reported by Momin et al. [8] for rectangular smooth duct. Dittus-Boelter correlation for Nusselt number is} \]

\[ N_u = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \]  \( \text{(1)} \)

\[ \text{Blasius equation for friction factor is} \]

\[ f_s = 0.085 \text{Re}^{-0.25} \]  \( \text{(2)} \)

Fig. (12) and Fig. (13) Shows comparison of Nusselt number and friction factor for smooth absorber plate. A reasonably good data ensures accuracy of the data.
being collected from self designed and fabricated experimental set-up. Error analysis based on the procedure described by Holman [12] has been carried out to find out uncertainties in measured/calculated values of experimental data. Uncertainty in Reynolds number, Nusselt number and friction factor values has been estimated as 2.45%, 2.34%, and 3.04% respectively.

![Fig.3. Different locations of thermocouples used to measure absorber plate temperature.](image1)

![Fig.4: Different locations of thermocouples used to measure air temperature in the duct.](image2)

![Fig.5: Photographic view of inclined dimpled metallic plate](image3)

**Data Reduction**

Following equations were used for calculating pressure drop across test section and orifice plate ($\Delta P_t$ and $\Delta P_o$), mass flow rate of air ($\dot{m}$), velocity of air ($V$), heat transfer rate ($q$), heat transfer coefficient ($h$), Nusselt number ($Nu$) and friction factor ($f$):

$$\Delta P_t = \rho \cdot g \cdot (\Delta h_t)$$

$$\Delta P_o = \rho \cdot g \cdot (\Delta h_o)$$

$$\dot{m} = C_d \cdot A_o \cdot \left( \frac{2 \rho \Delta P_o \cdot \sin \theta}{1 - \beta^4} \right)^{0.5}$$

$$V = \frac{\dot{m}}{\rho \cdot A_o}$$

$$q = \dot{m} \cdot C_p \cdot (T_a - T_i)$$

Also

$$q = h \cdot A_o \cdot (T_{pm} - T_{am})$$

Therefore, from Eq. (7) and (8)

$$h = \frac{q}{A_o \cdot (T_{pm} - T_{am})}$$

Where $T_{pm}$ and $T_{am}$ are mean temperature of absorber plate and air. These were determined from temperature values recorded for absorber plate and air at different locations along test section of the duct. Reynolds number, Nusselt number and friction factor values were calculated by using the following relationships:

$$Re = \frac{\rho \cdot V \cdot D}{\mu}$$

$$Nu = \frac{h \cdot D}{k}$$

$$f = \frac{2 \cdot \Delta P \cdot D}{\rho \cdot V^2 \cdot L}$$

**RESULTS AND DISCUSSION**

Variation of heat transfer coefficient as a function of mass flow rate of air for smooth and roughened absorber plate is shown in Fig.10. It is noted that heat transfer coefficient is very low for lower values of mass flow rate of air, but increases as mass flow rate of air increases. It may have happened due to turbulence caused by roughness elements in the laminar sub-layer only. Variations of Nusselt number as a function of Reynolds number for smooth and roughened absorber plate is shown in Fig.9. It has been observed that Nusselt number increases consistently with positive slope for entire range of Reynolds number. Also, Roughened plate has shown higher Nusselt number values than that of smooth plate, due to mixing of flow and generation of secondary flows in flow regimes similar to as reported by Saini and Saini [10] for expended metal mesh. Pressure drop increases with rise of mass flow rate of air for roughened and smooth plate as shown in Fig. 12. It is also observed that pressure drop is higher in case of plate roughened with $l/e = 20$ for entire range of mass flow rate of air, due to increase in degree of turbulence in flow regimes, because of presence of roughness element in the duct. Variation of friction factor as a function of Reynolds number for smooth and roughened plate is shown in Fig.11. Mixing of flow may be responsible for higher friction factor for roughened plate as compared to the smooth plate.

![Fig.9: Effect of Reynolds number on Nusselt number.](image4)

![Fig.10: Effect of mass flow rate on heat transfer coefficient](image5)

![Fig.11: Effect of Reynolds number on friction factor.](image6)
CONCLUSION

An experimental investigation on performance of artificially roughened duct used in solar air heaters has been reported in the present paper. Effect of artificial roughness (created by inclined elliptical dimples on absorber plate) on heat transfer and friction has been investigated for Reynolds numbers range of 8000–25000. It has been observed that roughened absorber plate results into higher heat transfer coefficient at the cost of frictional penalty. In order to predict performance of the system thermal efficiency and effective efficiency for roughened solar air heater has been found of the order of 2.3 and 1.9 times respectively as compared to solar air heater having smooth absorber plate.

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>cross-sectional area of duct, m²</td>
</tr>
<tr>
<td>Aₐ</td>
<td>area of orifice plate at the throat, m²</td>
</tr>
<tr>
<td>C₁</td>
<td>coefficient of discharge (dimensionless)</td>
</tr>
<tr>
<td>C₂</td>
<td>specific heat of air, J kg⁻¹ K⁻¹</td>
</tr>
<tr>
<td>D</td>
<td>hydraulic diameter of duct, m</td>
</tr>
<tr>
<td>e</td>
<td>height of roughness element, m</td>
</tr>
<tr>
<td>Dₚₐ</td>
<td>pressure drop across orifice plate, N m⁻²</td>
</tr>
<tr>
<td>f</td>
<td>friction factor (dimensionless)</td>
</tr>
<tr>
<td>g</td>
<td>acceleration due to gravity, m s⁻²</td>
</tr>
<tr>
<td>H</td>
<td>height of the duct, m</td>
</tr>
<tr>
<td>lₑ</td>
<td>relative short-way length</td>
</tr>
<tr>
<td>lₑ₀</td>
<td>relative long-way length</td>
</tr>
<tr>
<td>∆hₚ</td>
<td>difference of manometric fluid levels in micro-manometer, m</td>
</tr>
<tr>
<td>Tₑ</td>
<td>inlet air temperature of the test section, K</td>
</tr>
<tr>
<td>Tₑ₀</td>
<td>mean temperature of air, K</td>
</tr>
<tr>
<td>Tₑ₀₀</td>
<td>mean temperature of absorber plate, K</td>
</tr>
<tr>
<td>V</td>
<td>velocity of air, m s⁻¹</td>
</tr>
<tr>
<td>W₀ₚ</td>
<td>width of duct, m</td>
</tr>
<tr>
<td>rₑ</td>
<td>nose radius ofroughness element</td>
</tr>
<tr>
<td>K</td>
<td>length of duct, m</td>
</tr>
<tr>
<td>L₀</td>
<td>length of roughness element, mm</td>
</tr>
<tr>
<td>ρ₀</td>
<td>density of air, kg m⁻³</td>
</tr>
<tr>
<td>ρₚ₀</td>
<td>density of fluid used in micro-manometer, kg m⁻³</td>
</tr>
<tr>
<td>θ₀</td>
<td>inclination of U-manometer, degree</td>
</tr>
<tr>
<td>μ</td>
<td>dynamic viscosity of air, kg s⁻¹ m⁻¹</td>
</tr>
</tbody>
</table>

REFERENCES

2. T. Mahajan, R. Singh, B. Bhushan, “Performance Investigation Of Artificially Roughened Duct Used In Solar Air Heaters”


