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# THERMO HYDRAULIC PERFORMANCE OF SOLAR AIR DUCT HAVING TRIANGULAR PROTRUSIONS AS ROUGHNESS GEOMETRY

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## 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 triangular protrusions as roughness geometry. An increase in heat transfer and friction loss has been observed in solar 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.

## INTRODUCTION

The role of energy becomes increasingly important to fulfill needs of modern societies and to sustain fast economic and industrial growth worldwide. The rapid depletion of fossil resources has necessitated an urgent search for alternative sources of energy. Solar energy is available freely and an indigenous source of energy provides a clean and pollution free atmosphere. Energy consumption has been increasing at one of the fastest rates in the world due to population growth and economic development. In view of world's depleting fossil fuel reserves and environmental threats, development of renewable energy sources has received an impetus. Renewable energy resources are those energy resources which get renewed by nature gain and again these resources are to be continuously produced in nature .Example would solar energy, wind energy, tidal energy, nuclear fusion, gobar gas, biomass, geothermal energy etc. Renewable energy is derived from natural processes that are replenished constantly. In its various forms, it derives directly from the sun, or from heat generated deep within the earth. Included in the definition is electricity and heat generated from solar, wind, ocean, hydropower, biomass, geothermal resources, and bio fuels and hydrogen derived from renewable resources. Renewable energy resources and significant opportunities for energy efficiency exist over wide geographical areas, in contrast to other energy sources, which are concentrated in a limited number of countries. Rapid deployment of renewable energy and efficiency, and technological diversification of energy sources, would result in significant energy security and economic benefits.

From many alternatives, solar energy stands out as the brightest long range resource for meeting continuously increasing demand for energy. The energy consumption in the world particularly in the industrialized countries has been growing at alarming rate. Fossil fuels which today meet major part of the energy demand are being depleted quickly. World From many alternatives, solar energy stands out as the brightest long range resource for meeting continuously increasing demand for energy. The energy consumption in the world particularly in the industrialized countries has been growing at alarming rate. Fossil fuels which today meet major part of the energy demand are being depleted quickly. World has started running out of oil and it is estimated that 80% of the world's supply will be consumed in our lifetimes. Coal supplies may appear to be large but even this stock may not last longer than a few decades. More over the pollution hazard arising out of fossil fuel burnings become quite significant in recent years. Nuclear power has proposed a number of problems and nuclear fusion is still a speculative technology. Thus we are forced to look for unconventional energy sources such as geothermal ocean tides, wind and sun. It is also hoped that these alternative energy sources will be able to meet considerable part of the energy demand. Various types of unconventional energy sources are such as geothermal ocean tides, wind and sun. All unconventional energy sources have geographical limitations but Solar energy has less geographical limitation as compared to other unconventional energy sources because solar energy is available over the entire globe and only the size of the collector field needs to be increased to provide the same amount of heat or electricity. It is the primary task of the solar energy system designer to determine the amount of quality and timing of the solar energy available at the site selected for installing a solar energy conversion system so among all these solar energy seems to hold out the greatest promise for the mankind. It is inexhaustible, non-polluting and devoid of political control.

Simplest method to utilize solar energy for heating applications is to convert it into thermal energy by using solar collectors. From many alternatives, solar energy stands out as the brightest long range resource for meeting continuously increasing demand of energy. Solar air heaters are considered to be compact and less complicated as compared to solar water heaters. The performance of a solar collector is described by an energy balance that indicates the distribution of incident solar energy into useful energy gain and various heat losses from top, bottom and edges of the collector. Under steady conditions, the useful heat delivered by a solar collector is equal to the energy absorber in the absorber plate minus the heat losses from the surface directly and indirectly to the surroundings. The thermal efficiency of the solar collector is defined as the ratio of the useful heat gain to the incident solar energy. Thermal efficiency of solar air heater depends upon heat transfer coefficient between absorber plate and air flowing through the duct. So efficiency could be increase by enhancing the heat transfer coefficient between absorber plate and flowing air. In the present research work also, methodology of artificial roughness has been used for carrying out experimental investigation on thermal performance of duct used in solar air heater. It has been observed that artificial roughened duct results better thermal performance as compared to smooth duct. In order to include effect of friction penalty due to roughness created on absorber plate, investigation has also been carried out on thermo-hydraulic performance of artificially roughened solar air heaters can be fabricated using cheaper as well as less amount of material as compare to solar water heaters. Thermal efficiency of a solar air heater is generally considered poor because of low heat transfer capability between absorber plate and air flowing in the duct. So efficiency could be increase by enhancing the heat transfer coefficient between absorber plate and flowing air. One of the methods for enhancement of convective heat transfer is by creating turbulence at heat transfer surface with the help of artificial roughness on absorber plate. Artificial roughness breaks the laminar sub-layer on the heat transferring surface due to which thermal resistance between the heat transferring surface and flowing air decreases. However, use of artificial roughness results in higher friction and hence higher pumping power requirements. Therefore, it is desirable that the turbulence should be created in the vicinity of the wall i.e. only in the laminar sub-layer region, which is responsible for thermal resistance. Many investigators have attempted to design a roughness element, which can enhance convective heat transfer coefficient with minimum increase in friction loss. Artificial roughness can be created by different methods such as wire fixation, rib formation by machining processes, using expanded metal mesh ribs and by forming protrusions on the absorber plate. However, generating artificial roughness with a method other than performing protrusions is a very tedious task. Many investigations on forced convective heat transfer in smooth and roughened ducts have been reported in literature. Artificial roughness on the surface of absorber plate can be provided by fixing small diameter wires, ribs formed by machining process, wire mesh or expanded metal mesh and by forming dimple/protrusion shape geometry as has been reported by Bhushan and Singh (2010), Hans et al. (2009) and Varun et al. (2009). Experimental investigations on heat transfer and friction in artificially roughened solar air heater duct have been reported by Gupta et al. (1993), Momin et al. (2002) ,Jaurker et al. (2006), Karwa (2003), Karmare and Tikekar (2007), Prasad and Saini (1998), Saini and Saini (1997). Nusselt number and friction factor correlations have been developed by these investigators by using experimental data. Considerable amount of experimentation has been reported in literature to study effect of different type of roughness geometries on heat transfer and friction in duct of solar air heaters. However, generating artificial roughness with a method other than performing protrusions is a very tedious task. Formation of protrusions on the absorber plate is the latest and simple technique as protrusions are easy to fabricate and do not add extra weight to the absorber plate. It has been revealed from literature that an experimental investigation is required to be carried out on heat transfer and flow characteristics of solar air heater duct having triangular protrusion as roughness geometry. In the present paper, an experimental investigation has been reported for analyzing heat transfer and flow characteristics of duct having protruded absorber plate. In order to predict performance of the system having such type of roughened absorber plate, Nusselt number and friction factor correlations as a function of system and operating parameters have been developed by using experimental data.

## EXPERIMENTAL PROGRAM

### Experimental Apparatus

In order to carry out present experimental work, a test rig available in Solar Energy Lab was used. The rig has been designed and fabricated as per the guidelines proposed in the literature for similar experimental investigations reported by Saini and Saini, (1997) and Saini and Verma, 2008. Schematic diagram and photographic view of experimental set-up is shown in Fig. 1. Aspect ratio of rectangular duct is 10; generally it is kept in the range of 9 to 11 for solar air heaters. Aspect ratio of rectangular duct channel is the ratio of its width (W) and height (H). The length of test section was kept as 1000 mm, whereas length of entry and exit sections were kept as 900 mm and 500 mm respectively which was in accordance with the ASHRAE standard (1977) which recommends a minimum entry and exit length of air duct as $5\sqrt{WH}$ and $2.5\sqrt{WH}$ respectively. Therefore the total length of air duct was 2400 (900+1000+500) mm. Galvanized iron sheet of 20SWG of size 2400 x 330 mm was used as absorber plate and considered as top broad wall of the duct, whereas bottom and side walls of duct were made from 12mm thick plywood. In order to minimize heat loss, outer sides of air duct were well insulated. An electric heater having a size of 1500 x 300 mm was fabricated by combining series and parallel loops of nichrome wire on asbestos sheet and was placed on top of the absorber plate at a gap of 75 mm. Variac was connected to heater in order to control the heat flux. Air was sucked through the duct by means of a centrifugal blower provided at exit side of the pipeline. Control valves were provided at inlet and outlet of centrifugal blower for control of air flow rate through the duct. An orifice plate of 40 mm diameter and made from 3 mm thick brass plate was installed in the pipeline of 80 mm diameter with U-tube manometer, for measuring flow rate of air. Based upon calibration procedure, coefficient of discharge for orifice plate was obtained equal to0.62. Micro-manometer was used to measure the pressure drop across test section of the duct. Calibrated copper-constantan thermocouples were used for measuring plate and air temperatures. Twelve thermocouples were installed in the air duct and twelve thermocouples were fixed on test section of the test thermocouples were used for measuring plate and air temperatures. Twelve thermocouples were installed in the air duct and twelve thermocouples were fixed on test section of the test section of the absorber plate for measuring plate temperature in each set of experimentation. Photographic view of absorber plate roughened by formation of triangular protrusions is shown in Fig. 3.

Arrangement and shape of artificial roughness geometry used in the present experimental investigation is shown in Fig.4.

### Experimental procedure

It was decided to collect heat transfer and friction data in set of experiment under five values of mass flow rate of air in order to cover range of Reynolds number. Continuity of thermocouples was checked properly before starting each set of experiment. In order to collect heat transfer and friction data, electric heater and centrifugal blower were switched ON for starting each set of experiment. Air was sucked through entry section as shown in Fig.1a and desired flow rate of air was set with the help of control valves. In order to collect heat transfer and friction data, electric heater and centrifugal blower were switched on for starting each set of experiment. Air was sucked through entry section as shown in Fig.1a and desired flow rate of air was set with the help of control valves. In order to ensure arrival of quasi-steady state condition, temperature values indicated by all thermocouples were observed at a regular interval of 15 min. On attaining quasi-steady state condition under each value of mass flow rate of air, experimental data were collected from U-tube manometer fluid column ($\Delta h_1$), micro-manometer fluid column ($\Delta h_2$), temperature of absorber plate and air at various locations in test section of the duct and voltage and current supplied to electric heater.

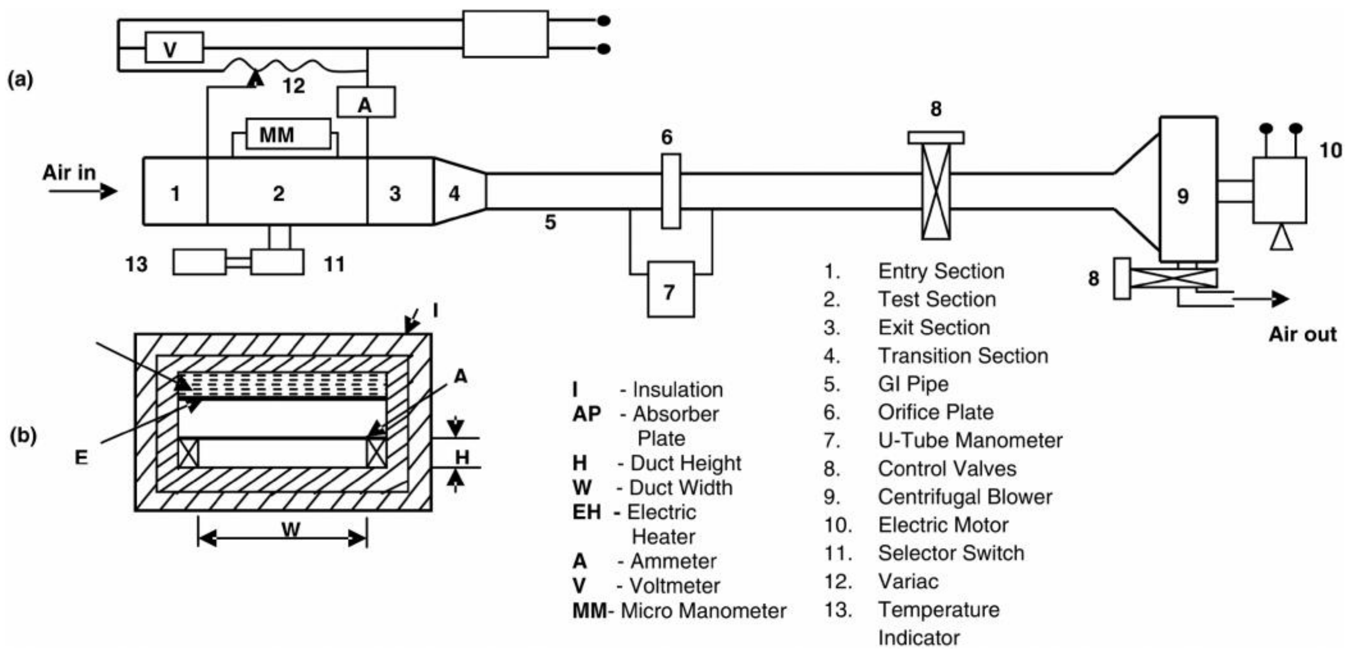


**FIGURE 1 (A).SCHEMATIC OF EXPERIMENTAL SET-UP. B).SECTIONAL VIEW OF DUCT FROM ENTRY SIDE**

**FIGURE 2 PHOTOGRAPHIC VIEW OF EXPERIMENTAL SET-UP**

**FIGURE 3 PHOTOGRAPHIC VIEW OF ABSORBER PLATE**

**TABLE 1 RANGE/VALUE OF SYSTEM AND OPERATING PARAMETERS**

| S.No. | Parameter | Range/Value |
|---|---|---|
| 1. | Reynolds number (Re) | 4000-20000 |
| 2. | Relative long way length (L/e) | 18.75-37.5 |
| 3. | Relative short way length (S/e) | 18.75-37.5 |
| 4. | Relative roughness height (e/D) | 0.03 |
| 5. | Duct aspect ratio | 10 |

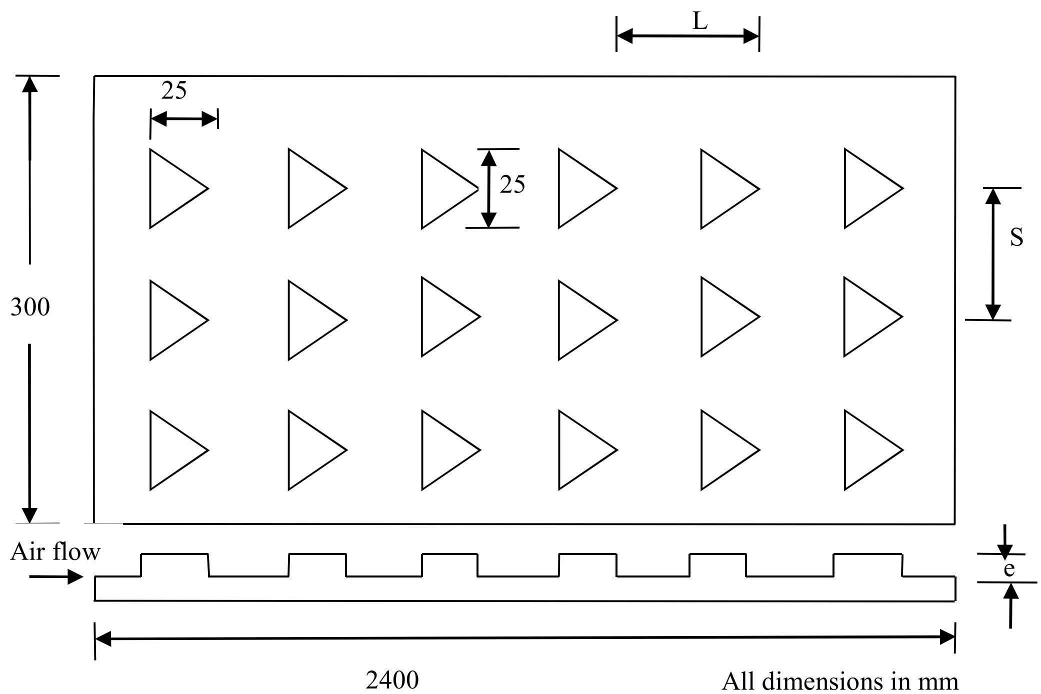


**FIGURE 4 SCHEMATIC OF ROUGHENED ABSORBER PLATE**

## Data Reduction

Numerical values of mass flow rate '$\dot{m}$', useful energy gain 'Qu', heat transfer coefficient 'h', Nusselt number 'Nu', Reynolds number 'Re' and friction factor 'f' were obtained by using experimental observations as described in the following steps.

$$\Delta Po = \rho w g(\Delta h1) \tag{1}$$

$$\dot{m} = CdAt\left[\frac{2\rho \Delta Po \sin\theta}{1-\beta^4}\right]^{0.5} \tag{2}$$

$$V = \frac{\dot{m}}{\rho A} \tag{3}$$

$$\mathrm{Re} = \frac{\rho VD}{\mu} \tag{4}$$

$$Qu = \dot{m}Cp(To - Ti) \tag{5}$$

$$Qu = hAc(Tpm - Tam) \tag{6}$$

From equations (5) and (6)

$$h = \frac{Qu}{Ac(Tpm - Tam)} \tag{7}$$

$T_{pm}$ and $T_{am}$ were determined from temperature values obtained for absorber plate and air at different locations along test section of the duct.

$$Nu = \frac{hD}{k} \tag{8}$$

$$\Delta P = \rho k g(\Delta h2) \tag{9}$$

$$f = \frac{(\Delta P)D}{2\rho L V^2} \tag{10}$$

## Validation of Experimental Data Collected from Experimental Set-Up

A thorough check of experimental set-up was carried out by conducting experimentation on smooth duct. Accuracy of Nusselt number and friction factor data collected for smooth duct was verified by comparing it with the data obtained from following Nusselt number and friction factor correlations reported by Hans et al. (2010) for rectangular smooth duct.

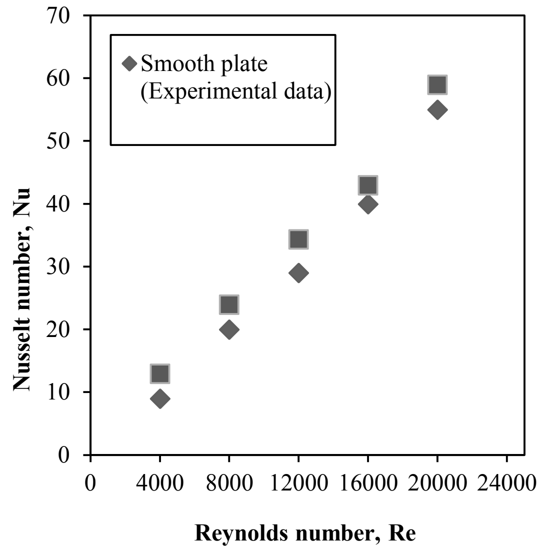


FIGURE 5 COMPARISON OF EXPERIMENTAL AND PREDICTED DATA OF NUSSELT NUMBER FOR SMOOTH PLATE

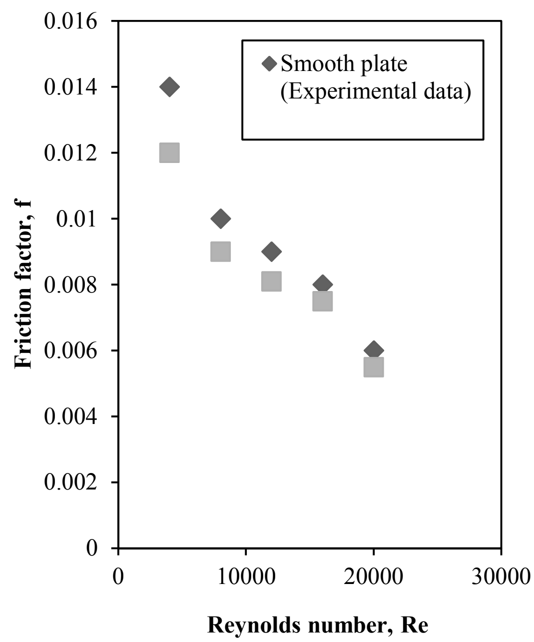


FIGURE 6 COMPARISON OF EXPERIMENTAL AND PREDICTED DATA OF FRICTION FACTOR FOR SMOOTH PLATE

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \text{ (DittusBoelterCorrelation)} \tag{11}$$

$$f = 0.085\,\mathrm{Re}^{-0.25} \text{ (ModifiedBlasiusCorrelation)} \tag{12}$$

Comparison of experimental and predicted data of Nusselt number and friction factor is shown in Figs. 5 and 6 respectively. A reasonably good argument between experimental and predicted data ensures accuracy of the data being collected from the experimental set-up.

## RESULTS AND DISCUSSION

Experimental data of heat transfer and friction in the roughened duct as a function of system and operating parameters have been reported and discussed in the present section. Figs. 7 & 8 represent effect of system and operating parameters on Nusselt number. It has been observed in both the cases, Nusselt number increases monotonously with increase in Reynolds number.

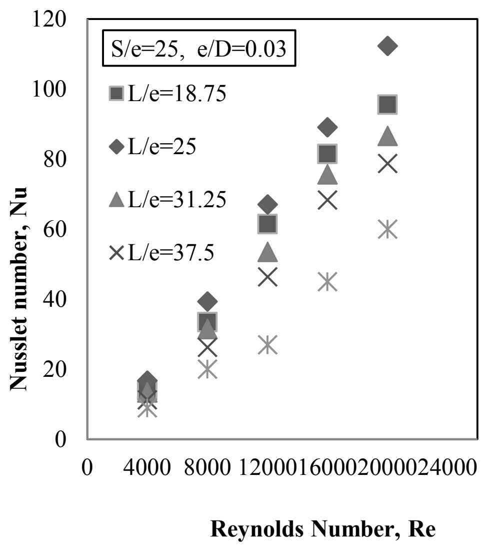


**FIGURE 7 VARIATION OF NUSSELT NUMBER AS A FUNCTION OF REYNOLDS NUMBER FOR RANGE OF RELATIVE LONGWAY LENGTH (L/E)**

It can be observed that at low Reynolds number, Nusselt number for all types of surfaces have same value, which might be due to the fact that in low range of Reynolds number effect of roughness element for breaking of laminar sub-layer of roughness element is insignificant. At higher Reynolds number, this layer is disturbed by the roughened surface; hence boundary layer thickness decreases which results an increases in heat transfer coefficient and consequently results into higher values of Nusselt number as has been reported by Karmare et al. (2007).

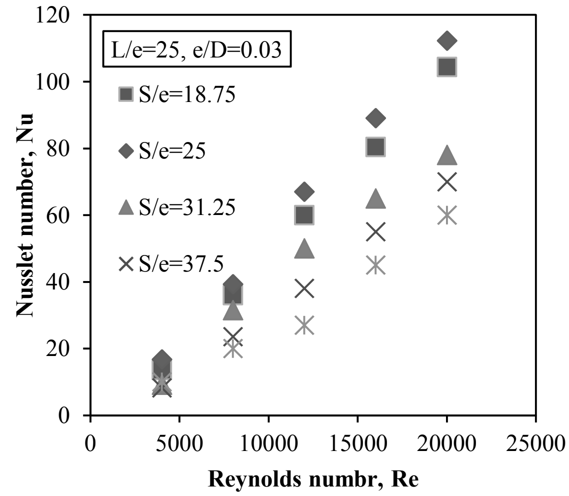


**FIGURE 8 VARIATION OF NUSSELT NUMBER AS A FUNCTION OF REYNOLDS NUMBER FOR RANGE OF RELATIVE SHORTWAY LENGTH (S/E)**

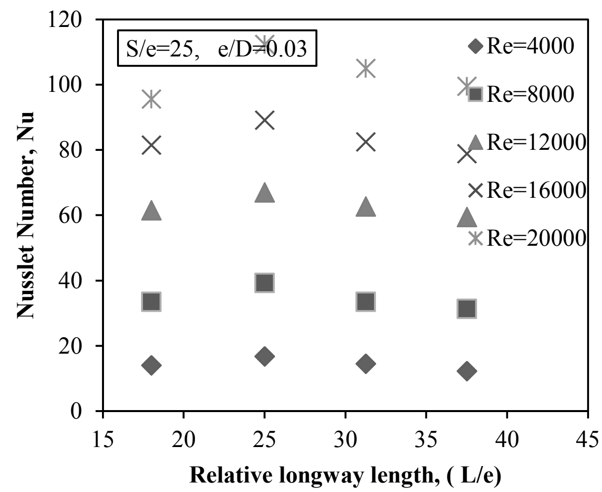


**FIGURE 9 VARIATION OF NUSSELT NUMBER AS A FUNCTION OF RELATIVE LONGWAY LENGTH (L/E) FOR RANGE OF REYNOLDS NUMBER**

Fig. 9. shows the effect of relative longway length (L/e) on Nusselt number at given value of Reynolds number. It can be observed that Nusselt number is maximum corresponding to relative longway length (L/e) of 25 and further it decreases with an increase in relative longway length (L/e). For the values of relative longway length (L/e) below 25, small distance between the protrusions may create hindrance in formation of vortices which results in low values of heat transfer coefficient as compared to that obtained for relative longway length (L/e) of 25, beyond 25,

heat transfer enhancement gets reduced as has been reported by Bhushan and Singh (2011).

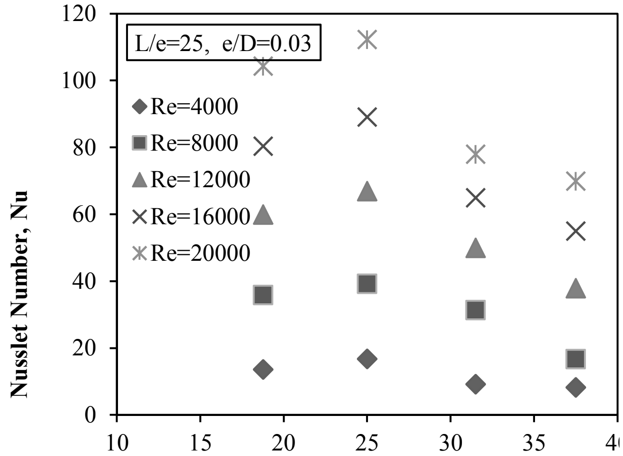


**FIGURE 10 VARIATION OF NUSSELT NUMBER AS A FUNCTION OF RELATIVE SHORTWAY LENGTH (S/E) FOR THE RANGE OF REYNOLDS NUMBER**

Fig. 10. shows the effect of relative shortway length (S/e) on Nusselt number at given value of Reynolds number. It can be observed that Nusselt number is maximum corresponding to relative shortway length (S/e) of 25 and further it decreases with an increase in relative shortway length (S/e). For the values of relative shortway length (S/e) below 25, small distance between the protrusions may create hindrance in formation of vortices which results in low values of heat transfer coefficient as compared to that obtained for relative shortway length (S/e) of 25. Similarly for relative shortway length (S/e) beyond 25, distance between the protrusions may be unable to break the laminar sub-layer and vortices formation may not take place which results in less heat transfer as has been reported by Bhushan and Singh (2011).

Fig. 11 and Fig. 12 show the effect of relative longway length (L/e) and relative shortway length (S/e) on friction factor at given values of Reynolds number. It can be observed that friction factor is minimum corresponding to relative longway length (L/e) of 37.5 and further it increases with decrease in relative longway length (L/e) for different values of Reynolds number. Similarly friction factor is minimum corresponding to relative shortway length (S/e) of 37.5 and it increases with decrease in relative shortway length. With decrease in relative longway length (L/e) and relative shortway length (S/e), effective pitch of protruded geometry decreases which increases the friction factor as has been reported by and Bhushan and Singh (2011).

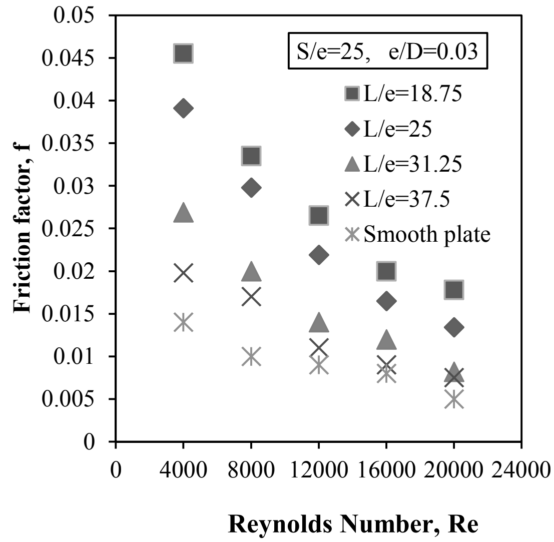


**FIGURE 11 VARIATION OF FRICTION FACTOR AS A FUNCTION OF RELATIVE LOGWAY LENGTH (L/E) FOR THE RANGE OF REYNOLDS NUMBER**

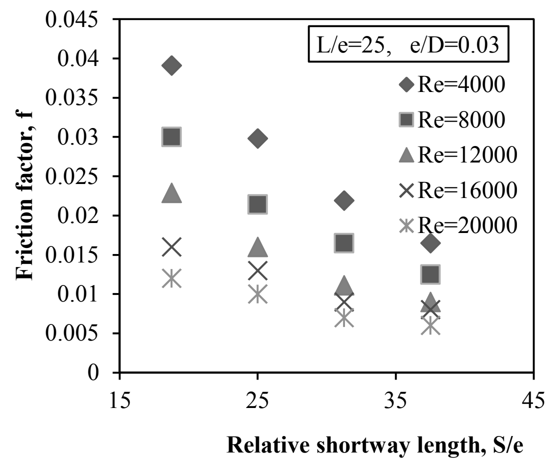


**FIGURE 12 VARIATION OF FRICTION FACTOR AS A FUNCTION OF RELATIVE SHORTWAY LENGTH (S/E) FOR THE RANGE OF REYNOLDS NUMBER**

## Development of Nusselt Number and Friction Factor Correlations

It is revealed from experimental data that Nusselt number and friction factor are strong function of system and operating parameter. Functional relationship for Nusselt number and friction factor can be written as:

$$Nu=f_1(Re,L/e,S/e) \tag{13}$$

$$f=f_2(Re,L/e,S/e) \tag{14}$$

In order to predict performance of an artificially roughened duct of a solar air heater having protrusions as roughness geometry, Nusselt number and friction factor correlations w Sigma plot software has been utilized in order to carry out regression analysis of experimental data.ere developed for range of system and operating parameters.

### Nusselt Number Correlation

Experimental data of Nusselt number were plotted on log-log scale as shown in Fig. 13. It shows that Nusselt number and Reynolds number have almost a linear relationship. From first order regression of the data, it has been found that average slope of all lines is 1.25. Variation of function $\frac{Nu}{\mathrm{Re}^{1.25}}$ with relative longway length (L/e) was plotted on log-log scale as shown in Fig. 14 and from the second order regression, following correlation was obtained.

$$\frac{Nu}{\mathrm{Re}^{1.25}} = B_o (L/e)^{10.08} \exp[-8.44\{log(L/e)\}^2] \quad (15)$$

Similarly variation of function $\frac{Nu}{5.88X10^{-11}\,\mathrm{Re}^{1.25}\,L/e^{10.08}\exp[-8.44\{\log(L/e)\}^2]}$ with relative shortway length (S/e) was plotted on log-log scale as shown in Fig. 16. and following correlation was obtained.

$$\frac{Nu}{5.88X10^{-11}(\mathrm{Re})^{1.25}(L/e)^{10.08}\exp[-8.44\{\log(L/e)\}^2]} = C_o(\mathrm{Re})^{1.25}(L/e)^{10.08}(S/e)^{12.29}\exp[-(10.49)\{\log(S/e)\}^2] \quad (16)$$

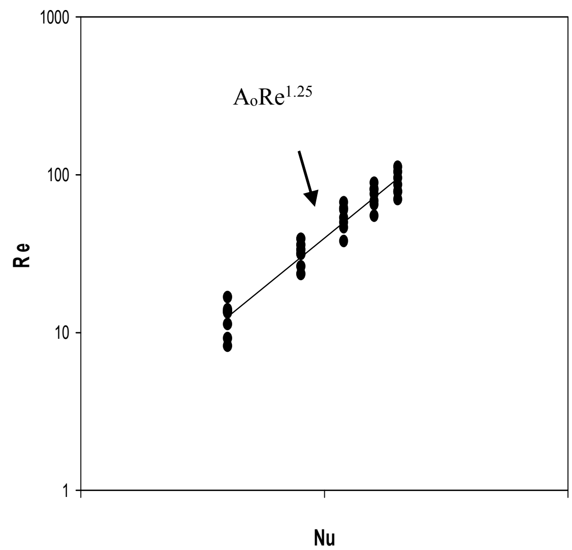


**FIGURE 13 PLOT OF NUSSELT NUMBER (NU) VS REYNOLDS NUMBER (RE)**

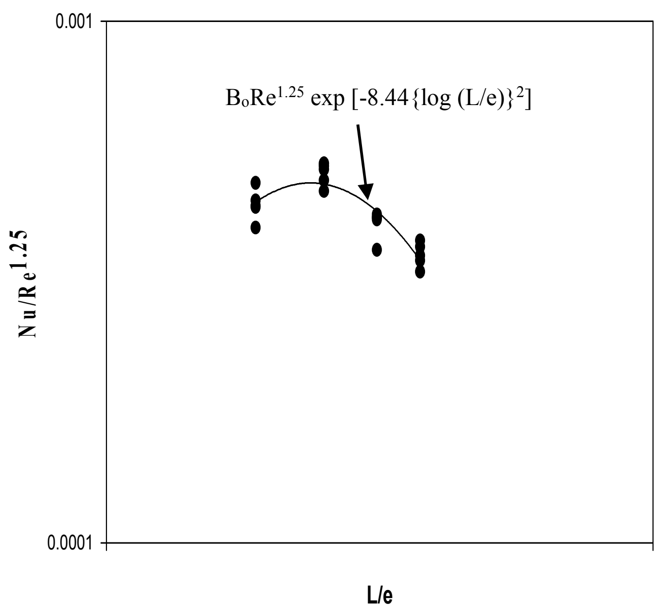


**FIGURE 14 PLOT OF NU/RE 1.25 VS RELATIVE LONGWAY LENGTH (L/E)**

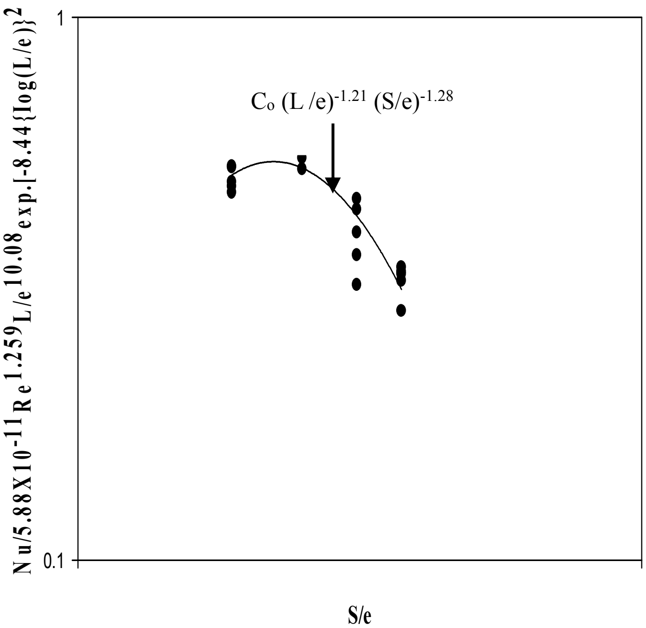


**FIGURE 15 PLOT OF F/6.76(RE)$^{-0.644}$(L/E)$^{-1.21}$ VS RELATIVE SHORTWAY**

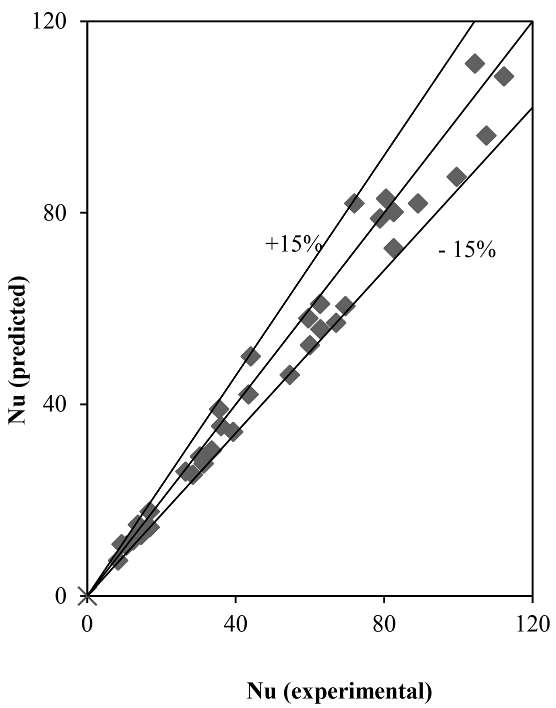


**FIGURE 16 COMPARISON OF EXPERIMENTAL AND PREDICTED DATA OF NUSSELT NUMBER**

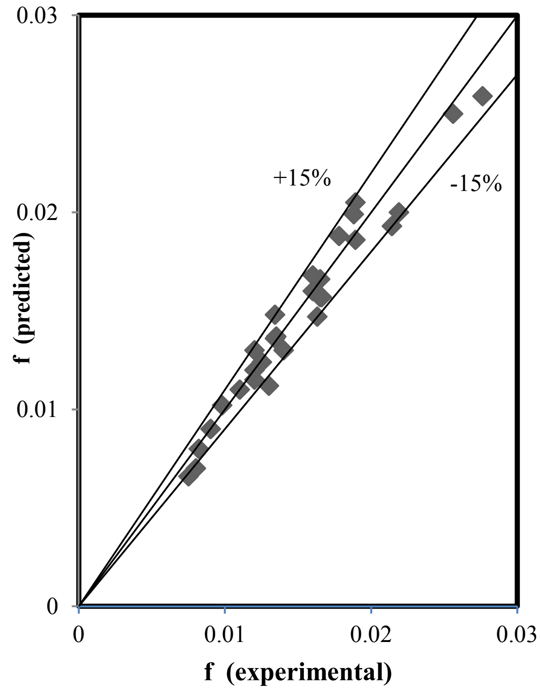


**FIGURE 17 COMPARISON OF EXPERIMENTAL AND PREDICTED DATA OF FRICTION FACTOR**

## Thermo-Hydraulic Performance of Solar Air Heater Duct

It has been observed in the present experimental investigation that artificial roughness is very useful methodology in order to enhance thermal performance of a solar air heater. However, roughened absorber plate results higher pressure drop i.e. more power consumption for making flow of air through the duct. Therefore, for such type of a system, one should evaluate thermo-hydraulic performance instead of thermal performance only in order to take into account friction penalty also.

Performance of the system based upon thermal as well as hydraulic characteristics is termed as thermo-hydraulic performance. In order to evaluate thermo-hydraulic performance of solar air heater duct, following relationship reported by Webb (1972) has been used.

$$\text{Performance factor} = \frac{(Nu / Nu_s)}{(f / f_s)^{1/3}} \tag{20}$$

From Fig.18. It can be observed that performance factor increases, attains maxima and then decreases with increase in Reynolds number. It could have happened due to the fact that up to maxima, effect of increase in heat transfer coefficient was more dominating as compared to friction factor. Reverse has happened after the point of maxima. It can be observed that solar air heater duct with roughened absorber plate having relative longway length (L/e) of 25 results higher value of performance factor for the range of Reynolds number.

However, for each absorber plate maximum value of performance factor has been obtained at Reynolds number of 12000. Based upon thermo-hydraulic performance analysis of solar air heater duct, it has been found that optimum values of system and operating parameters i.e. longway length (L/e), shortway length.

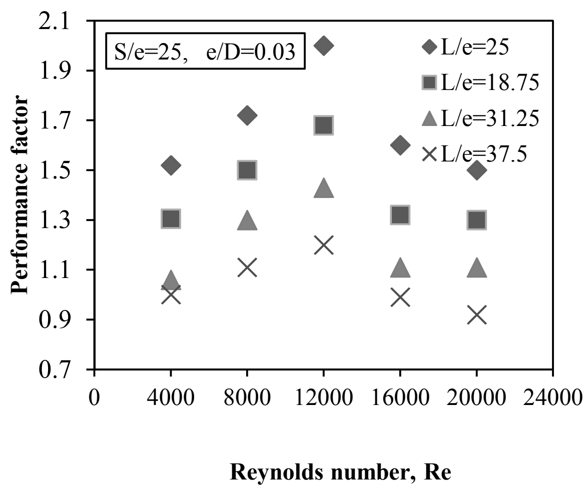


**FIGURE 18 EFFECT OF RELATIVE LONGWAY LENGTH (L/E) ON PERFORMANCE FACTOR**

Fig.19 shows plot between performance factor and Reynolds number for the range of relative longway length (L/e) at fixed value of relative shortway length (S/e) and relative roughness height (e/D). It can be observed that performance factor increases, attains maxima and then decreases with increase in Reynolds number. It could have happened due to the fact that up to maxima, effect of increase in heat transfer coefficient was more dominating as compared to friction factor. Reverse has happened after the point of maxima. It can be observed that solar air heater duct with roughened absorber plate having relative shortway length (S/e) of 25 results higher value of performance factor for the range of Reynolds number. However, for each absorber plate maximum value of performance factor has been obtained at Reynolds number of 1200.(S/e) and Reynolds number (Re) are 25, 25 and 1200 respectively as shown in Table 7.1. as compared to friction factor. Reverse has happened after the point of maxima. It can be observed that solar air heater duct with roughened absorber plate having relative shortway length (S/e) of 25 results higher value of performance factor for the range of Reynolds number. However, for each absorber plate maximum value of performance factor has been obtained at Reynolds number of 1200. (S/e) and Reynolds number (Re) are 25, 25 and 1200 respectively as shown in Table 2.

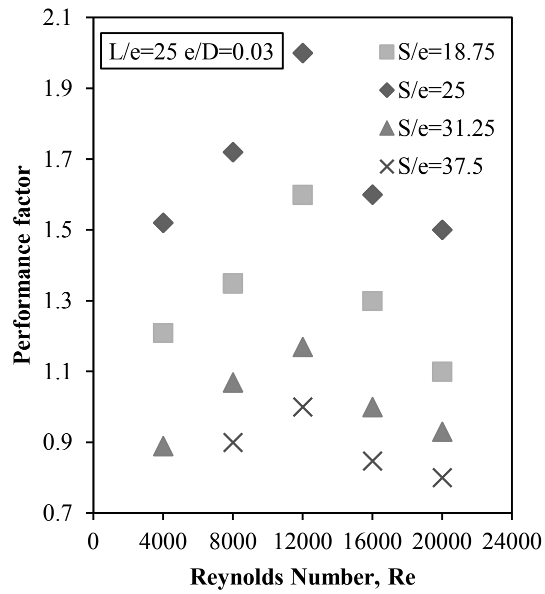


**FIGURE 19 EFFECT OF RELATIVE SHORTWAY LENGTH (S/E) ON PERFORMANCE FACTOR**

**TABLE 2 OPTIMUM VALUES OF SYSTEM AND OPERATING PARAMETERS**

| S.No. | Parameter | Optimum value |
|---|---|---|
| 1. | Relative longway length (L/e) | 25 |
| 2. | Relative shortway length (S/e) | 25 |
| 3. | Reynolds number (Re) | 25 |

## CONCLUSIONS

In the present experimental work, investigation was carried out on heat transfer and flow characteristics of artificially roughened duct used in solar air heaters. Roughness has been created by forming triangular type protrusions on the absorber plate. On the basis of present investigation, following conclusions were drawn:

1. Heat transfer coefficient/Nusselt number increases monotonously with an increase in mass flow rate of air/Reynold number for smooth as well as roughened duct. Heat transfer coefficient/Nusselt number for roughened duct has been found higher than the smooth duct for range of system and operating parameters.

2. Nusselt number maximum at L/e of 25, decreases with increase in relative longway length (L/e); for range of Reynolds numbers.

3. Nusselt number maximum at S/e of 25, decreases with increase in relative shortway length (S/e); for range of Reynolds numbers

4. Pressure drop increases monotonously with an increase in mass flow rate of air for smooth as well as roughened duct. Friction factor decreases with an increase in Reynolds number for smooth as well as roughened duct. Pressure drop/friction factor for roughened duct was found higher than the smooth duct for range of system and operating parameters.

5. Friction factor decreases with increase in relative longway length (L/e) for range of Reynolds number at fixed relative roughness height (e/D).

6. Friction factor decreases with increase in relative shortway length (S/e) for range of Reynolds number at fixed relative shortway length.

7. Nusselt number and friction factor correlations were developed by using the experimental data. Comparison of experimental data and that predicted from the Nusselt number and friction factor correlations shows good agreement.

8. Thermo-hydraulic performance for all types of roughened absorber plates have been carried out to find out system and operating parameter for having best thermo-hydraulic performance.

9. Absorber plate with relative longway length (L/e) of 25 and relative shortway length (S/e) of 25 has shown optimum thermo-hydraulic performance at Reynolds number of 12000.

## FUTURE SCOPE OF WORK

1. A new design of solar air heater may be proposed and tested under actual climatic conditions by using the investigated type of roughness geometry.

2. Experimental investigation can be carried out using similar type of roughness geometry in reverse direction.

3. Experimental investigation can be carried out on new types of roughness geometries under all categories of artificial roughness geometries reported in literature.

## NOMENCLATURE

| | |
|---|---|
| A | cross-sectional area of duct, $m^2$ |
| $A_c$ | collector area, $m^2$ |
| $A_t$ | area of orifice plate at the throat, $m^2$ |
| $C_d$ | coefficient of discharge (dimensionless) |
| $C_p$ | specific heat of air, J/kgK |
| D | hydraulic diameter of duct, m |
| E | height of roughness element, m |
| e/D | relative roughness height (dimensionless) |
| f | friction factor (dimensionless) |
| $f_s$ | friction factor for smooth plate (dimensionless) |
| g | acceleration due to gravity, $m^2/s$ |
| H | height of duct, m |
| I | insolation, $W/m^2$ |
| $\Delta h_1$ | difference of manometric fluid in U-tube manometer, m |
| $\Delta h_2$ | difference of manometric fluid levels in micro- manometer, m |
| h | heat transfer coefficient, $W/m^2$ K |
| k | thermal conductivity of air, W/m K |
| L | length of test section, m |

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