A Study on the Performance of Solar Air Heater Using Arc Shaped Turbulators on the Absorber Plate

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Abstract

Solar energy is most important renewable energy resource due to its quantitative abundance. The simplest and most efficient way to utilize solar energy is to convert it into thermal energy for heating applications. A solar collector is a device designed to absorb the incoming solar radiant energy converting into thermal energy at the absorbing surface and transforming this energy to a fluid flowing through the collector. Solar air heaters, because of their inherent simplicity, are cheap and most widely used collection devices. The primary disadvantage of solar air heaters is the need for handling relatively large volumes of air because of its low thermal capacity as working fluid. The thermal performance of a conventional solar air heater is poor because of relatively low heat transfer coefficient between absorber plate and the carrier fluid (air). Use of artificial roughness on the absorber plate has been found to be an effective method of enhancing heat transfer coefficient. In this present work an attempt is made to study the performance of solar air heater using wire geometry of arc shaped thin circular wires as a turbulators on the absorber plate. The investigation has covered a Reynolds number (Re) range of 3000-8000, relative roughness height (e/D_h) of 0.034, duct aspect ratio (W/B) is kept 5, the relative roughness pitch (p/e) used is 10,20,30 and 40. The heat transfer and friction factor values obtained are compared with those of smooth duct under similar flow conditions. Investigation shows. The maximum enhancement of average Nusselt number is found to be 1.12 times for transverse wire geometry, 1.17 times for single arc wire geometry and 1.22 times for double arc wire geometry compared to that of smooth plate. The maximum enhancement of average friction factor is found to be 5.61 times for transvers wires geometry, 5.89 times for single arc wire geometry and 6.09 times for double arc wire geometry. The comparison of transverse, single arc, double arc, and smooth absorber plate shows that the Nusselt number , friction factor and efficiency index is highest for double arc shaped roughness followed by single arc, transverse and least for smooth absorber plate.

Keywords: Solar energy, Solar air heater, Artificial roughness, heat transfer enhancement, Nusselt number, Friction factor.

1. Introduction

One of the most promising and economical method of increasing the heat transfer rate is providing roughness on the absorber plate, which creates turbulence in the flowing air. Artificial roughness is basically a passive heat transfer enhancement technique by which thermo hydraulic performance of a solar air heater can be improved. It has been found that the artificial roughness applied on the heat transferring surface breaks the viscous sub-layer, which reduces thermal resistance and promotes turbulence in a region close to artificially roughened surface in order to attain higher heat transfer coefficient, the laminar sub-layer formed in the vicinity of the absorber plate is broken and the flow at the heat-transferring surface is made turbulent by introducing artificial roughness on the surface. 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 must be created only in the region very close to the heat transferring surface, so that the power requirement may be lessened. This can be done by keeping the height of the roughness elements to be small in comparison with the duct dimensions. Experimental setup is fabricated and tests are conducted to determine the effect of geometrical parameters of arc shaped roughness on heat transfer and friction characteristics

Abdul-Malik Ebrahim Momin, J. S. Saini, S. C. Solanki [1] experimentally investigated the effect of geometrical parameters of V-shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of olar air heater with absorber plate having V-shaped ribs on its underside. It was found that for relative roughness height of 0.034 and for angle of attack of 60°, the V-shaped ribs enhance the values of Nusselt number by 1.14 and 2.30 times over inclined ribs and smooth plate case at Reynolds number of 17034. It means that the V-shaped ribs have definite advantage over the inclined ribs for similar operating conditions. J.C. Han [2] carried out an experimental study of fully developed turbulent air flow in square ducts with two opposite rib roughened walls was performed to determine the effects of the rib pitch to height and rib height to equivalent diameter rations on friction factor and heat transfer coefficients. Reynolds number is varied from 7,000 to 90,000. Results of roughened wall were compared with those of smooth wall and observed that the average friction factor was 2.1 to 6 times that for four sides smooth duct. The Stanton number of the ribbed side is about 1.5 to 2.2 times that of the four-sided smooth duct when relative roughness pitch varies from 40 to 10. N. Sheriff and P. Gumley [3] investigated experimentally the heat transfer and friction characteristics of a surface with discrete roughness. They used metal wires as roughness elements of different sizes varying in a wide range of diameter. In this study they kept relative roughness pitch in the range 10:1. In this study it is shown that pumping power required to force the fluid for same heat transfer surface and fluid temperature difference, will be minimum when (fr / f s) < (Nur/Nus)^3 .This shows any increase in the friction factor increases the heat transfer characteristics of roughened surface resulting in a more efficient heat transfer surface. Santosh B. Bopche and Madhukar S. Tandale [4] investigated on heat transfer coefficient and friction factor by using artificial roughness in the form of specially prepared inverted U shaped turbulators on the absorber surface of an air heater duct. The roughened wall is uniformly heated while the remaining three walls are insulated. These boundary conditions correspond closely to those found in solar air heaters. The experiments encompassed the Reynolds number range from 3800 to 18000; ratio of turbulator height to duct hydraulic mean diameter is varied from, e/D_h = 0.0186 to 0.03986 (D_h = 37.63 mm and e = 0.7 to 1.5 mm) and turbulator pitch to height ratio is varied from, p/e = 6.67 to 57.14 (p = 10 to 40 mm). The angle of attack of flow on turbulators, α= 90° kept constant during the whole experimentation. At Reynolds number, Re = 3800, the maximum enhancement in Nusselt number and friction factor are of the order of 2.388 and 2.50, respectively. The maximum enhancement in Nusselt number and friction factor values compared to smooth duct (in this medium 'Re' flow range of investigation at pitch of about 10 mm and turbulator tip height of 1.5 mm) are of the order of 2.82 and 3.72, respectively. The turbulences generated only in the viscous sub-layer region of boundary layer results in better thermo-hydraulic performance i.e. maximum heat transfer enhancement at affordable friction penalty.

M. M. Sahu and J. L. Bhagoria [5] carried out investigation to study the heat transfer coefficient by using 90° broken transverse ribs on absorber plate of a solar air heater; the roughened wall being heated while the remaining three walls are insulated. They found that roughened absorber plates increase the heat transfer coefficient 1.25–1.4 times as compared to smooth rectangular duct under similar operating conditions at higher Reynolds number. Rajkumar Ahirwar and A .R. Jaurker[6] experimentally investigated the heat transfer characteristics in solar air heater duct having three surfaces are smooth and top surface which absorb and transfer the heat is roughened by artificial roughness in V- shaped. The experimental parameters involve Reynolds number (Re) ranges from 2000 to 14000, Relative roughness pitch (p/e) 6 to12, for a fixed parameter relative roughness height (e/D_h) 0.036 and angle of attack of flow (α) 60°. The presence of artificial roughness in the form of ribs gives higher heat transfer rate as compared to smooth duct. The maximum enhancement of heat transfer as compared to smooth is found on the relative roughness pitch (p/e) 10 in the range of present parameters under investigation. Hence the use of artificial roughness in the form of V- shaped is recommended in all practical purposes in the field of heat transfer. Apurba Layek J.S. Saini S.C. Solanki [7] investigated heat and fluid flow characteristics of fully developed turbulent flow in a rectangular duct having repeated integral transverse chamfered rib-groove roughness on one broad wall. Experiments encompassed the Reynolds number range of 3000–21,000; relative roughness pitch of 4.5–10, chamfer angle of 5–30,relative groove position of 0.3–0.6 and relative roughness height of 0.022–0.04. As compared to smooth surface the roughened surface can yields a maximum of about 3.24-fold and 3.78-fold increase in the Nusselt number and friction factor, respectively in the range of parameters investigated. The maximum heat transfer enhancement occurs for the relative roughness pitch of 6 and relative groove position of 0.4.

R.P. Saini and Jitendra Verma [8] developed heat transfer and friction factor correlations for a duct having dimple shape artificial roughness on the underside of the absorber surface. The investigation has covered the range of Reynolds number (Re) from 2000 to 12,000, relative roughness height (e/D_h) from 0.018 to 0.037 and relative pitch (p/e) from 8 to 12. The maximum value of Nusselt number has been found corresponds to relative roughness height (e/D_h) of 0.0379 and relative pitch (p/e) of 10. While minimum value of friction factor has been found correspond to relative roughness height (e/D_h) of 0.0289 and relative pitch (p/e) of 10. It is therefore, roughness parameters of the geometry can be selected by considering the net heat gain and corresponding power required to propel air through the duct. S.V. Karmare, A.N. Tikekar [9] experimentally investigated the heat transfer to the airflow in the rectangular duct of an aspect ratio 10:1. The top wall surface is made rough with metal ribs of circular cross section in staggered manner to form defined grid. The roughened wall is uniformly heated and the other walls are insulated. This geometry of duct closely corresponds to that used in solar air heaters. The main findings of their study are, The presence of metal grit ribs on collector surface of the duct yields up to two-fold enhancement in the Nusselt number and three-fold enhancement in the friction factor, for the range of Re (4000–17,000).

A. R. Jaurker, J. S. Saini, B. K. Gandhi [10], experimentally investigated the heat transfer and friction characteristics of rib-grooved artificial roughness on one broad heated wall of a large aspect ratio duct shows that Nusselt number can be further enhanced beyond that of ribbed duct while keeping the friction factor enhancement low. As compared to the smooth duct, the presence of rib grooved artificial roughness yields Nusselt number up to 2.7 times while the friction factor rises up to 3.6 times in the range of parameters investigated. The maximum heat transfer occurs for a relative roughness pitch of about 6.0, and it decreases either side of relative roughness pitch and similar trend is observed for friction factor. The optimum condition for heat transfer occurs at a groove position to pitch ratio of 0.4, while on the either side of this ratio, both Nusselt number and friction factor decreases. A. M. Lanjewar, J. L. Bhagoria and R. M. Sarviya [11] investigated the heat transfer enhancement by using W-shape ribs on absorber plate of solar air heater, the roughened wall being heated while the remaining three walls insulated. Artificially roughened solar air duct having an angle of attack of 60° gives best performance in the range of parameters investigated. W-shaped absorber plate with 60° rib angle brings heat transfer enhancement of about 1.32-1.92 times as compared to smooth plate in the range of parameters investigated. Varun, R. P. Saini, S. K. Singal [12] experimentally

investigated heat transfer and friction characteristics by using a combination of inclined as well as transverse ribs on the absorber plate of a solar air heater. The experimental investigation encompassed the Reynolds number (Re) ranges from 2000 to 14,000, relative roughness pitch (p/e) 3–8 and relative roughness height (e/D$_h$) 0.030. The thermal performance of roughened solar air heater is influenced by the roughness parameters and the best performance has been found for the roughness parameter that yield maximum heat transfer coefficient. The geometry having relative roughness pitch of 8 have the maximum thermal efficiency.

Anil Kumar, R. P. Saini, J. S. Saini [13] carried out an experimental investigation to study the effect of artificial roughness, in the form of multi v-shaped ribs with gap on one heated wall of a rectangular solar air heater duct and they found that the value of Nusselt number and friction factor increase with an increase of angle of attack and are maximum at an angle of attack of 60°.A symmetrical gap equal to the rib height in both legs of a Multi v-shaped rib results in a substantial improvement in the thermo-hydraulic performance. Manish Kumar Chauhan, Varun, Sachin Chaudhary [14] made an attempt to enhance the thermal as well as thermo hydraulic performance of solar air heater by providing roughness elements. An experimental investigation has been carried out on M-shaped ribs having circular cross section on absorber plate. 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. The artificial roughened solar air heater also causes 6-7 times enhancement in friction factor. The enhancements in the Nusselt number, friction factor and thermal efficiency have been observed strong function of the relative roughness height (e/D$_h$). S. S. Pawar, Dr. D. A. Hindolia, Dr. J. L. Bhagoria [18], Experimental investigation has been carried out to study heat transfer enhancement by using diamond shape as a rib on absorber plate. The experimental values of the thermal efficiency of the roughened absorber plate tested have been compared with that of the smooth plate and it is found that a plate having roughness Pitch 15 mm gives the highest efficiency of 83.2%.i.e 2.01 times that of the smooth plate
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2. Experimental details.

The schematic diagram of the test setup is shown in figure 1.The setup consists of a duct having artificially roughened absorber plate, heater, pipe line, centrifugal blower and instrumentations for measuring mass flow rate of air, pressure drop, temperature and voltage for heating the absorber plate. The most important part of the setup is the duct which was fabricated from plywood panels.
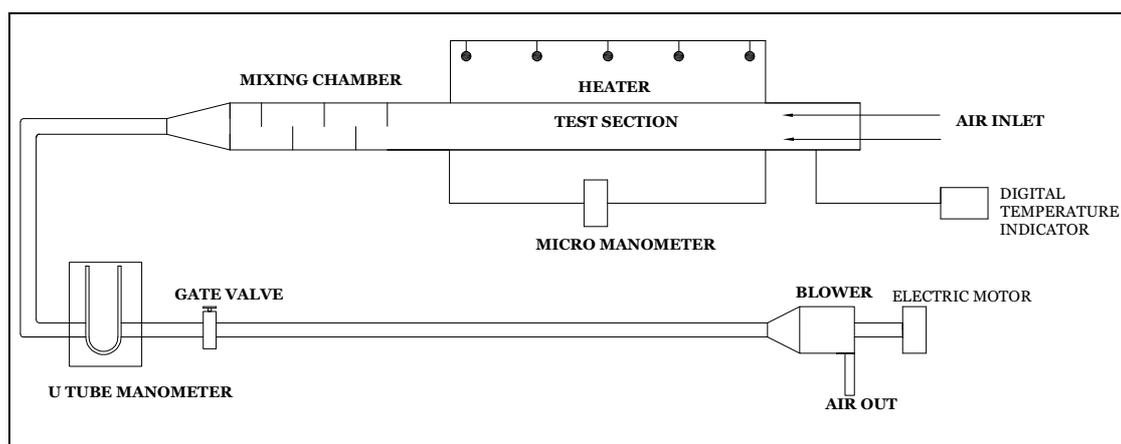

Figure 1: Experimental Setup

The duct is 2600 mm long, 150 mm wide and 30 mm deep. The test section is 1200 mm long and has a cross section of 150 mm width ×30 mm depth. The entry and exit lengths were 800 mm and 600 mm, respectively. It may be noted that ASHRAE Standard [93–77] recommends minimum entry and exit length of

$5\sqrt{WH}$ and $2.5\sqrt{WH}$, respectively. The sidewalls of duct were also made of thick plywood panels having smooth surface. A 2 mm thick Aluminium sheet of 1200mm× 150mm size is used as an absorber plate. The lower surface of the plate is provided with artificial roughness with varying relative pitch (p/e). A uniform heat flux up to a maximum of 1500W/m^2 to the absorber plate is provided by a solar simulator. It consist of five electric bulbs of 100 Watt each. These bulbs are equally spaced and kept on the absorber plate to provide uniform heating and to simulate the solar radiation flux. The power supply to the heater is controlled through an AC variac. To ensure all the heat flux is transferred to the duct and also to minimize the heat loss the duct is insulated on the other three sides except absorber plate side. An orifice meter connected with U- tube manometer is used to measure the air flow rate in the duct. A micro-manometer is used to measure the pressure drop across the test section. To measure the temperature of air and absorber plate at various locations calibrated chromel-alumel thermocouples are used.

2.1 Instrumentations.

A. Temperature measurement: Calibrated chrome-alumel thermocouples with digital micro-Volt-millimeter, indicating output in degree centigrade with an accuracy of 0.1 °C were used to measure air temperature and the temperatures of the absorber plate at different locations. The thermocouples were calibrated with a standard thermometer. Absorber Plate temperature are recorded by pasting eight thermocouples on the top surface of the plate. The location of thermocouples on the absorber plate is shown in figure below. Ten thermocouples are inserted inside the duct at different location as shown in figure to measure air temperature inside the duct. Two separate thermocouples are used to measure inlet and outlet temperature of air. Figure 2 shows the positioning of thermocouples on the absorber plate.
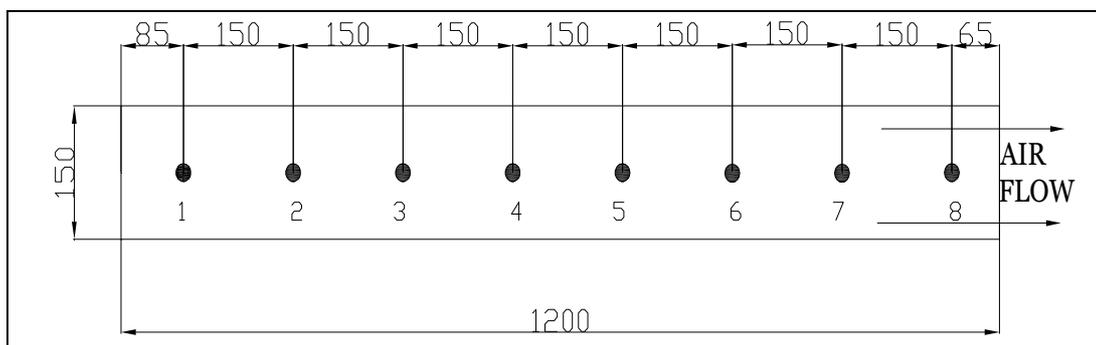

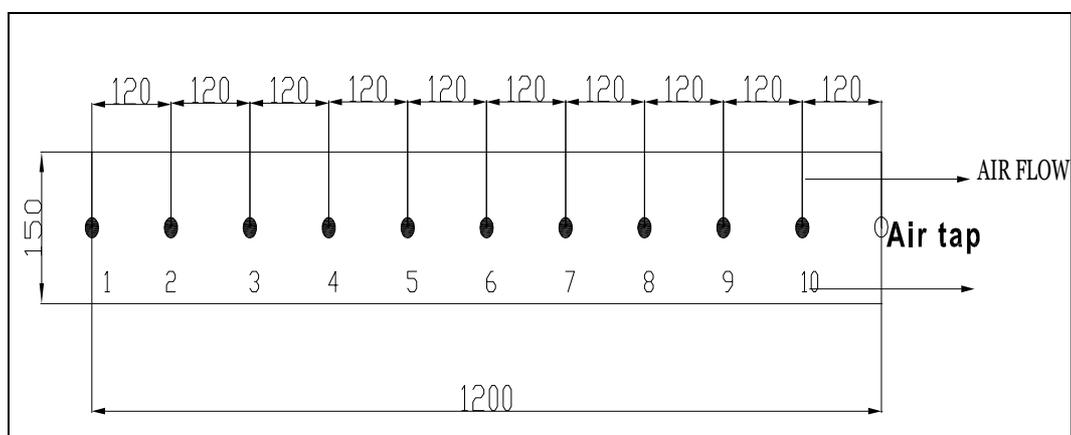**Figure 2**: Positioning of thermocouples on absorber plate

Figure. 3. Positioning of thermocouples inside air duct

B. Air flow measurement: The mass flow rate of air was measured by means of a flange type orifice meter calibrated by using pitot tube. The control valves were provided in the line to control the flow. The air is sucked through the rectangular duct by means of a blower. Mixing chamber consisting of baffles are provided for proper mixing of air and the temperature is measured at the exit.

C. Pressure Drop measurement inside the duct: Micro-manometer is used for measuring the pressure drop in the test section. Diesel having specific gravity 0.8373 is used in the manometer to increase the accuracy further.

2.2 Roughness geometries and range of parameters

The roughness geometries on the absorber plate which is investigated under the present investigation is shown in figure 4 (a), 4(b), 4(c). The range of parameters covered under the present investigation are given in the table 1.
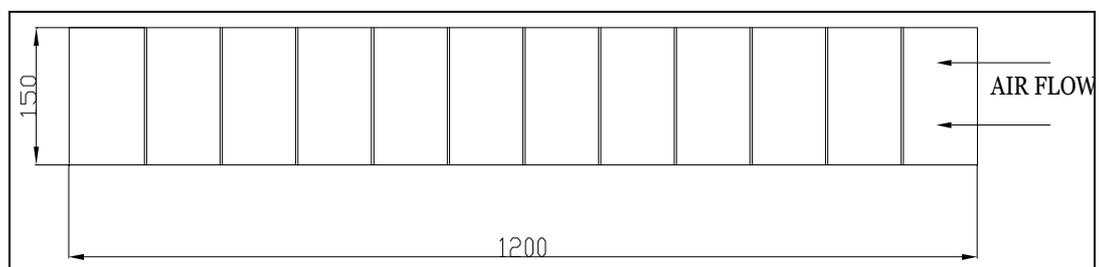

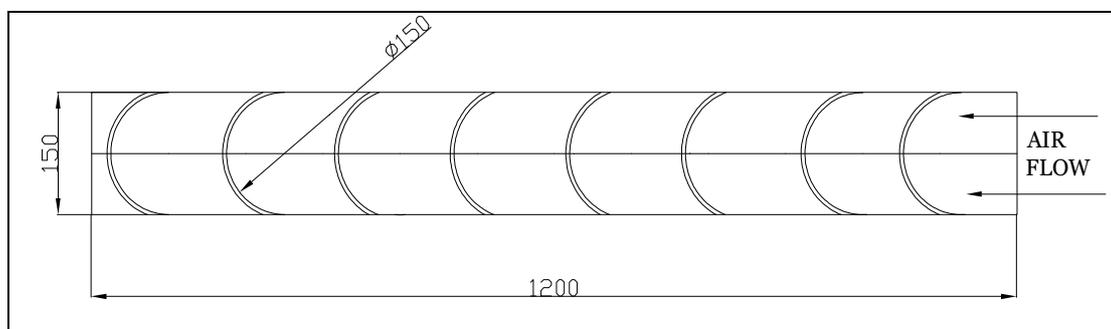**Figure. 4(a).** Schematic diagram of transverswire roughness geometry on the absorber plate.

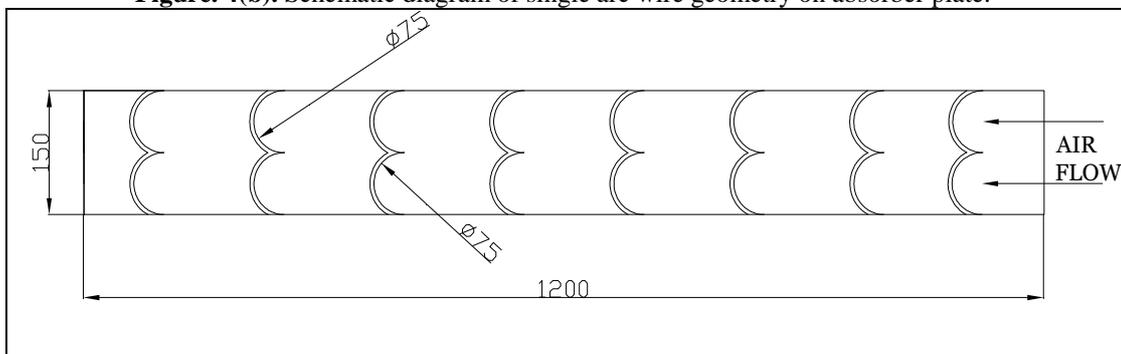**Figure. 4(b).** Schematic diagram of single arc wire geometry on absorber plate.

Figure. 4(c). Schematic diagram of double arc wire roughness geometry on the absorber plate.

Table 1: Range of parameters.

SI No	Parameters	Range
1.	Reynolds number(Re)	3000-8000
2.	Duct aspect ratio (W/B)	5
3.	Test-section length, L (mm)	1200
4.	Roughness height, wire diameter, e (mm)	1.7
5.	Hydraulic diameter of duct, D (mm)	50
6.	Relative roughness height, e/D	0.034
7.	Relative roughness pitch, p/e	10 to 40

2.3. Experimental Procedure

The test runs to collect the relevant heat transfer data were conducted under steady state conditions. For different airflow rates, the system was allowed to attain a steady state before the data were recorded. All the components of the setup and instrument are examined carefully for their proper operation. Artificially roughened absorber plate is installed and the test section is assembled. Heater is used for supplying energy for heating for one hour to the test section. After one hour the blower is switched on. No leakage in the joints of the duct and pressure tapping's is ensured. Control valve is used to adjust the mass flow rate. All the readings are taken once the mass flow rate is fixed. Tests for eleven values of mass flow rate of air is conducted in order to cover the entire range of Reynolds number.. The following parameters are measured during experimentation:

- Temperature of the absorber plate and air at inlet and outlet of the test section.
- Pressure drop across the test section.
- Pressure difference across the orifice meter.
- Voltage and current supply to the heater.

2.4. Experimental Data Reduction

The test runs to collect the relevant heat transfer data were conducted under steady state conditions. Eleven values of flow rate were used for each set of test. After each change of flow rate, the system was allowed to reach steady state before the data were recorded. The experimental data includes thermocouple readings and air mass flow rates. This data have been reduced to obtain the average plate temperature, average air temperature, mass flow rate (m), velocity of air (V), heat supplied to the air (q), and heat transfer coefficient (h). The thermo physical properties of the air have been taken at the average plate fluid temperature.

The mean air temperature or average flow temperature (T_f) is the simple arithmetic mean of the measured values of air at ten different locations of the duct.

$$T_f = \frac{(T1 + T2 + \cdots + T10)}{10}$$

(1)

The mean plate temperature, (Tp) is the average temperature of absorber plate at Eight different locations.

$$T_p = \frac{T11 + T12 \ldots T18}{8}$$

(2)

The mass flow rate is calculated as follows,

$$m = \rho c_d \, a_o \sqrt{\frac{2gH}{1 - (a_o/a_1)^2}}$$

(3)

The velocity of air in the duct is calculated as follows,

$$V = \frac{m}{\rho W B}$$

(4)

Useful heat gain to the air flowing in the duct is calculated as follows,

$$q = m c_p (t_o - t_i)$$

(5)

Convective heat transfer coefficient for the test section is calculated as follows

$$h = \frac{q}{A_c \left(t_p - t_f \right)}$$

(6)

The Nusselt number is calculated from the following expression

$$Nu_r = \frac{h D_h}{k}$$

(7)

The Reynolds number is calculate as follows

$$R_e = \frac{V D_h}{v}$$

(8)

The friction factor was determined from the following equations

$$f_r = \frac{D_h \, \Delta p}{2LV^2 \rho}$$

(9)

3. Results and Discussions

3.1. Plate with transverse wires as artificial roughness.

The heat transfer characteristics of solar air duct roughened with transverse wires, computed on the basis of experimental data collected for various flow and roughness parameters, have been discussed below. The results have been compared with those obtained in case of smooth ducts operating under similar operating conditions to discuss the enhancement in Nusselt number.

Figure. 5 shows the Nusselt number as a function of Reynolds number for relative roughness pitch (p/e) values of 10, 20, 30, 40 and for a fixed value of relative roughness height (e/D_h) of 0.034. It can be seen from the figures that for given value of roughness parameters, the Nusselt number monotonously increases with increase of Reynold number. It is seen from figure.5 that Nusselt number has been found maximum corresponding to relative roughness pitch, (p/e) of 10 which may be discussed on similar line as investigated by other investigators. Due to separation of the flow at the transverse wires, reattachment of free shear layer occurs for (p/e) of 10 and maximum heat transfer coefficient occurs in the vicinity of the reattachment region. For relative roughness pitch (p/e) of 20, 30, 40 reattachment may not occur hence the values of heat transfer coefficient corresponds to these values of (p/e) are found to be minimum. At very low Reynolds number (less than 5000), it can be seen from figure.5 that the values of Nusselt number for smooth duct is nearly equal to that of roughened duct may be attributed to the fact that laminar sub layer thickness increases as the flow is retarded by roughness elements in Reynolds number region.
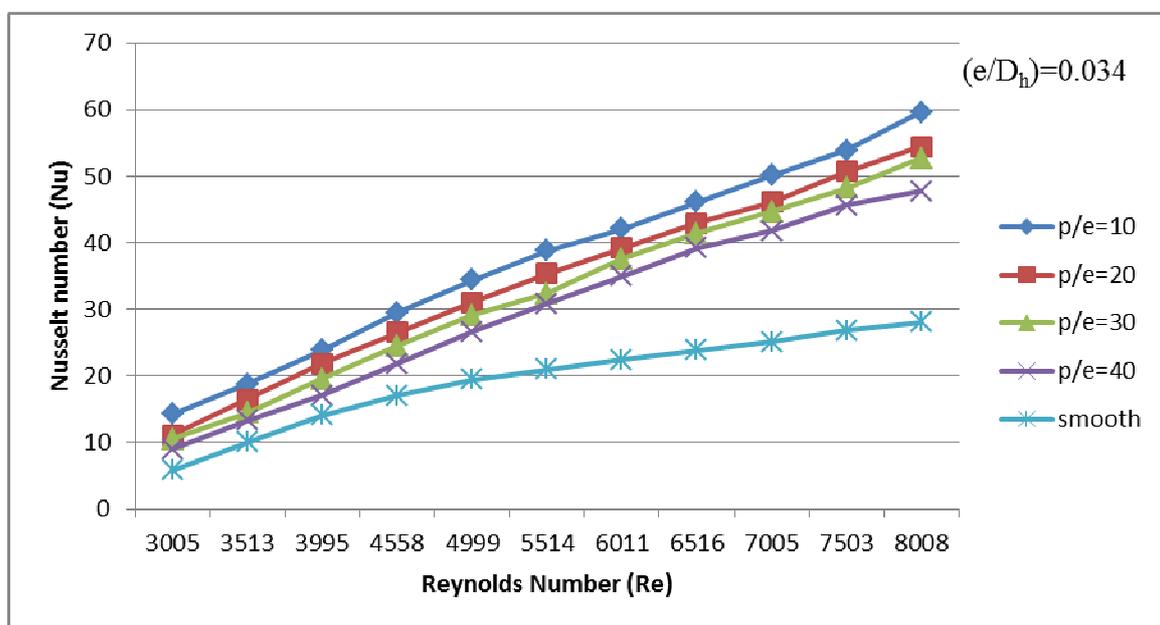

Figure. 5. Variation of Nusselt number with Reynolds number for different values of (p/e) and for a fixed value of (e/D_h) for Transverse geometry.

Figure 6 shows the variation of friction factor with Reynolds number for various relative roughness pitch (p/e) values of 10, 20, 30, 40 and for a fixed value of relative roughness height (e/D) of 0.034 from the graph it can be seen that the friction factor decreases with an increase of Reynolds number in all the cases due to the suppression of viscous sub-layer with increase in Reynolds number. Smooth plate has also been plotted in this figure for the purpose of comparison. It is observed that for a relative roughness pitch of 10 has maximum friction factor and that of 40 has minimum value
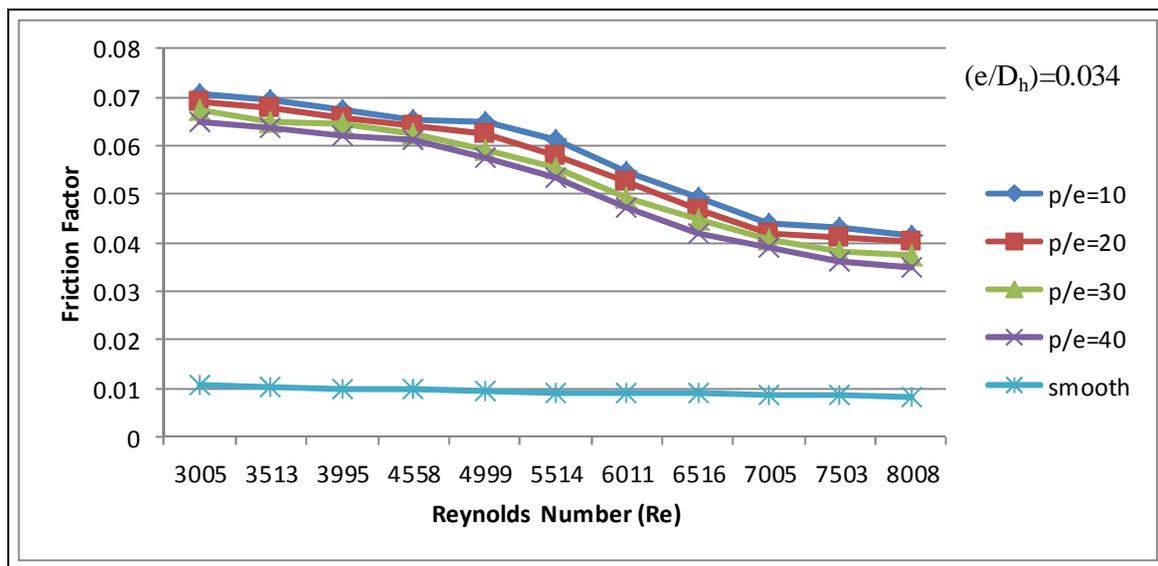

Figure. 6. Variation of Friction factor with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Transverse geometry.

The Efficiency index of roughness plate with varying Reynolds number is shown in the figure 7. From the graph it can be seen that the efficiency increases with the increase of Reynolds number up to certain extent but after that it starts to stabilize.
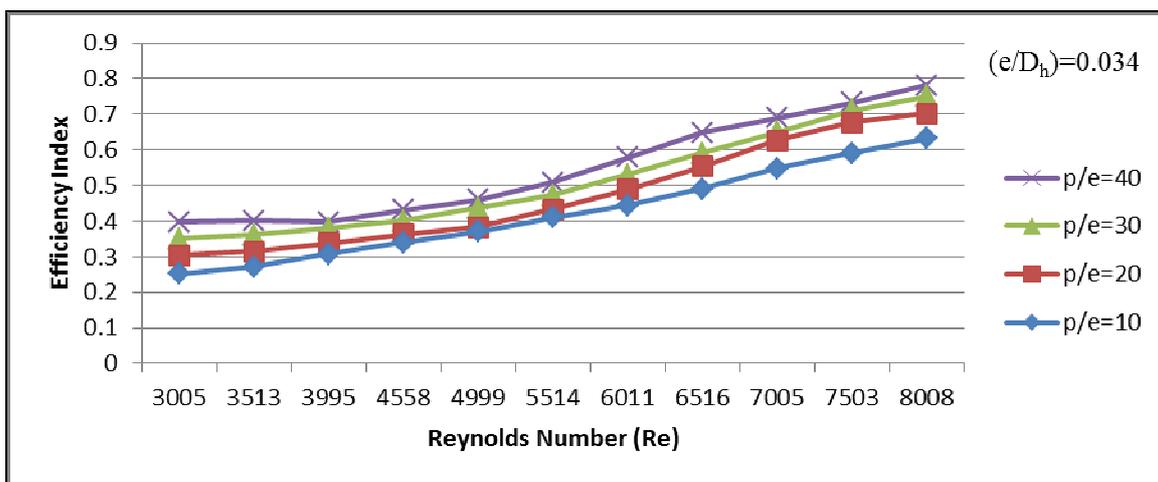

Figure. 7. Variation of Efficiency Index with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Transverse geometry.

3.2. Plate with single arc as artificial roughness

Figure 8. Shows the variation of Nusselt number with Reynolds number for different values of (p/e) for 10,20,30,40 and for a fixed value of Relative roughness height of (e/D_h) of 0.034. From the figure it is seen that the Nusselt number increases with increase in Reynolds number. Form the figure the Nusselt number is found to be maximum for relative roughness pitch (P/e) value of 10. This can be explained on the basis of flow separation concept. Due to flow separation at the arc shaped geometry, reattachment of free shear layer occurs for (p/e) 10 and maximum heat transfer coefficients occurs in the vicinity of the reattachment region. The heat transfer coefficients corresponding to the values of relative roughness pitch (p/e) of 20, 30, 40 are found to be minimum due to no reattachment of free shear layer. The Nusselt number for that of the smooth plate is also given for the comparison purpose, from the figure it can be seen at low Reynolds number (less than 5000) the values of Nusselt number for smooth plate is nearly equal to that of roughened duct due to the fact that laminar sub layer thickness increases as the flow is retarded by roughness elements in Reynolds number region.
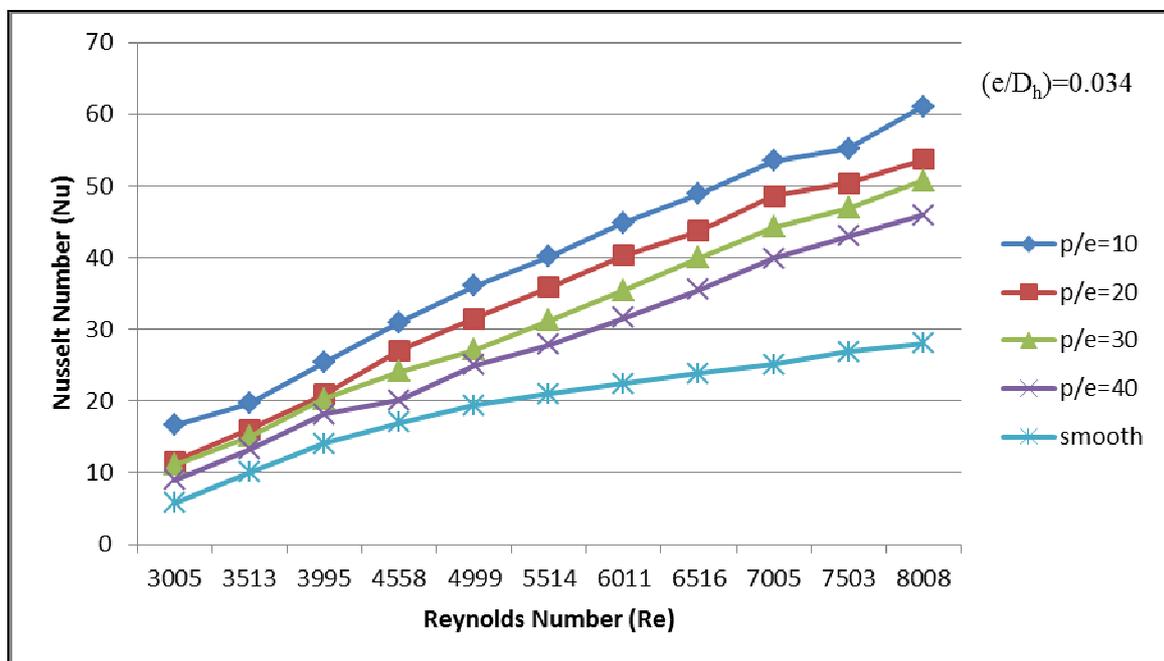

Figure.8. Variation of Nusselt number with Reynolds number for different values of (p/e) and for a fixed value of (e/D_h) for Single arc geometry.

Figure 9. Shows the variation of Friction factor with Reynolds number for various relative roughness pitch (p/e) values of 10, 20, 30, 40 and for a fixed value of relative roughness height (e/D_h) of 0.034 for sing arc geometry. It can be seen from the graph that the friction factor decreases with an increase of Reynolds number in all the cases. For the purpose of comparison friction factor of smooth plate is also presented in the plot. It can be observed that for a relative roughness pitch of 10 has maximum friction factor and that of 40 has minimum value.
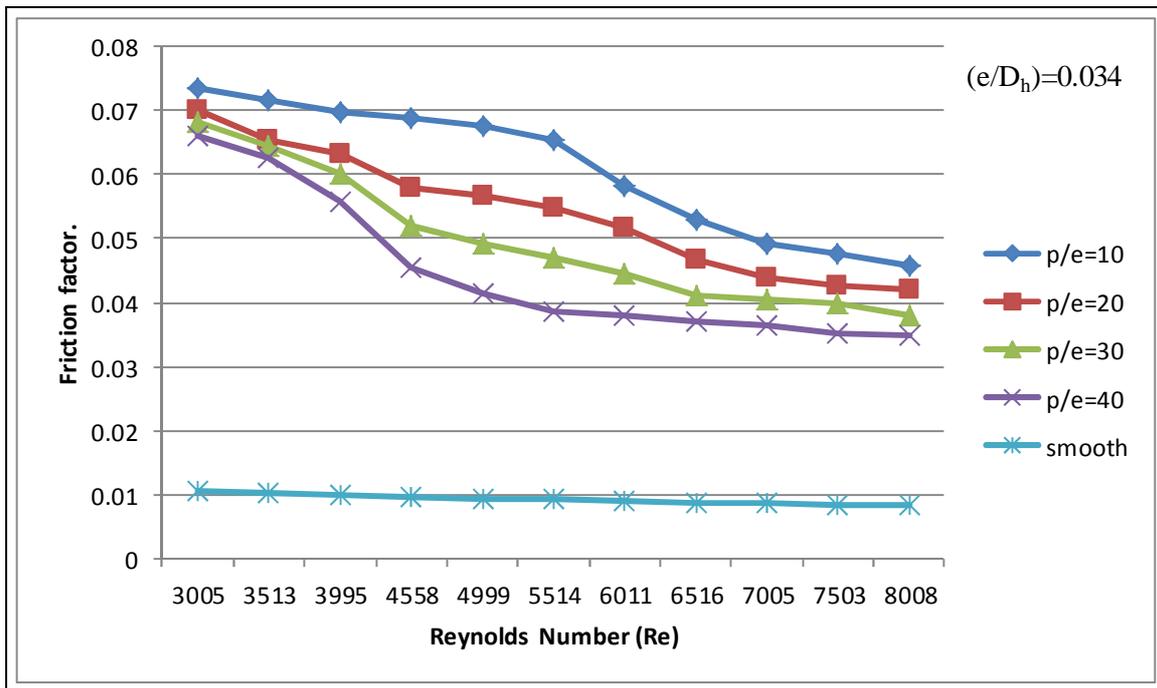
Figure. 9. Variation of Friction factor with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Single arc geometry.

Figure 10. Shows the variation of Efficiency Index with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Single arc geometry. From the graph it can be seen that the efficiency increases as the Reynolds number increases for all the relative roughness pitch (P/e) values. The maximum efficiency is found to be at (p/e) values of 40 and minimum for (p/e) value of 10.
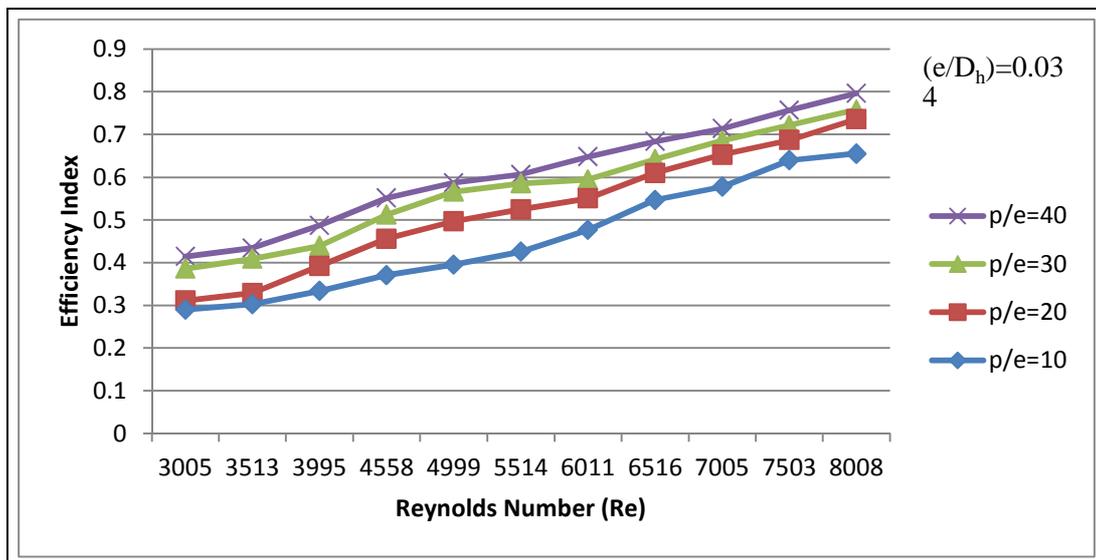

Figure. 10. Variation of Efficiency Index with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Single arc geometry.

3.3. Plate with double arc as artificial roughness

Figure 11. shows the variation of Nusselt number with Reynolds number for different values of (p/e) and for a fixed value of (e/D$_h$) for Double arc geometry. It has been observed that Nusselt number increases with an increase of Reynolds number for all values of relatives roughness pitch (p/e) as compared to smooth duct. This is due to the distinct change in the fluid characteristics as a result of roughness that causes flow separations, reattachments and the generation of secondary flows. It is observed that at lower Reynolds (lower than 5000) the values of Nusselt number for smooth plate are nearly equal to that of the roughened plate with double arc. The figure. 4.14. shows the variation of Nusselt number with Relative Roughness pitch (P/e) for different values of Reynolds number for Double arc geometry. It is seen that for a constant value of Reynolds number the value of Nusselt number decreases with increasing in Relative roughness pitch (p/e).
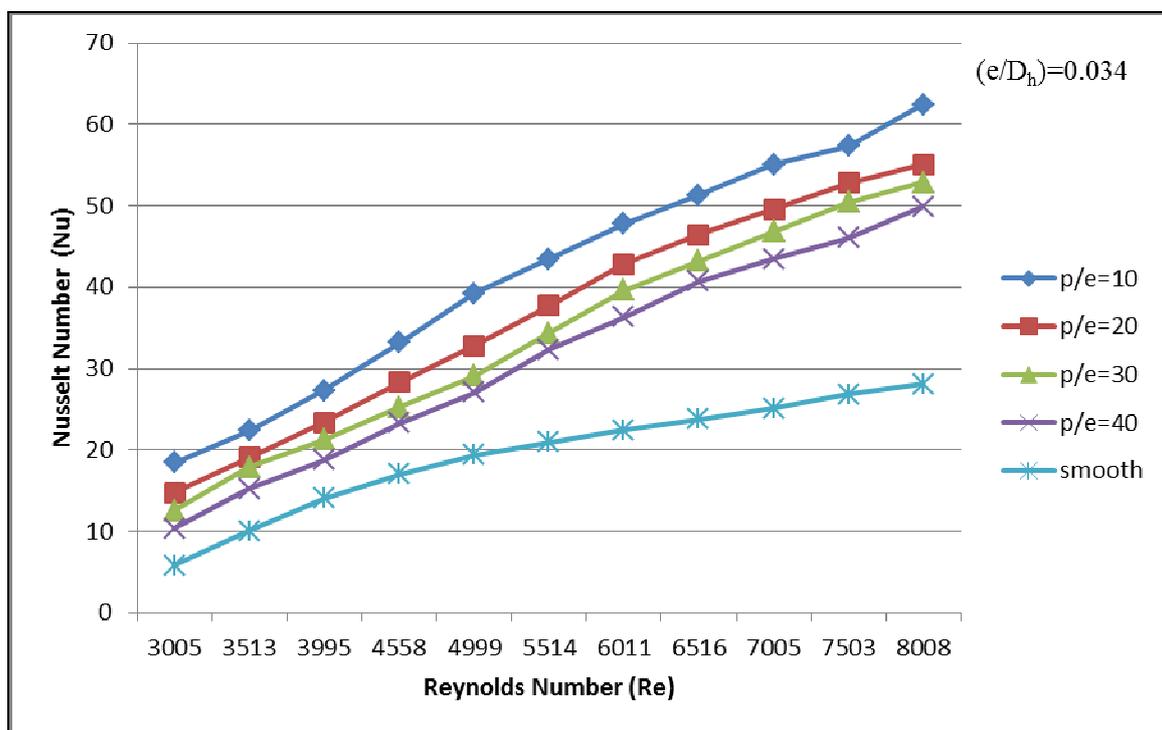

Figure. 11. Variation of Nusselt number with Reynolds number for different values of (p/e) and for a fixed value of (e/D$_h$) for Double arc geometry.

Figure 12. Variation of Friction factor with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Double arc geometry. The friction factor decreases with increase in Reynolds number for all the values of (p/e), this is explained on the basis of flow separation as explained above in the earlier cases. The maximum friction factor is observed for relative roughness pitch of 10 and minimum for 40.
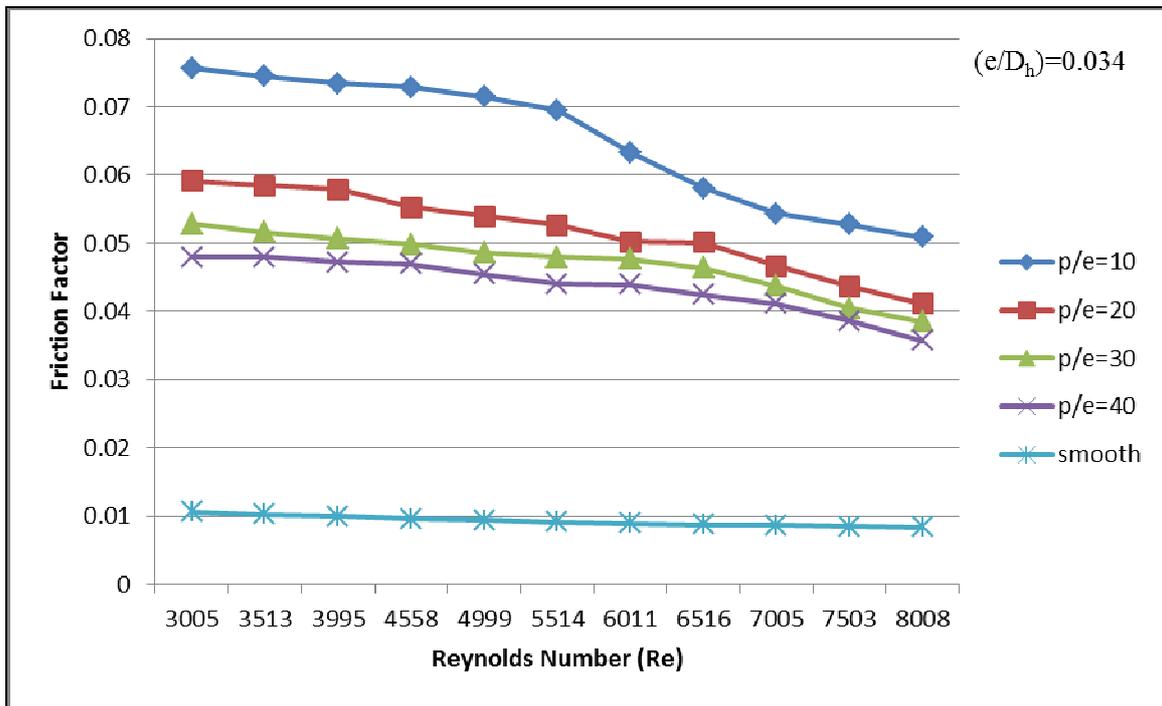
Figure. 12 Variation of Friction factor with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Double arc geometry.

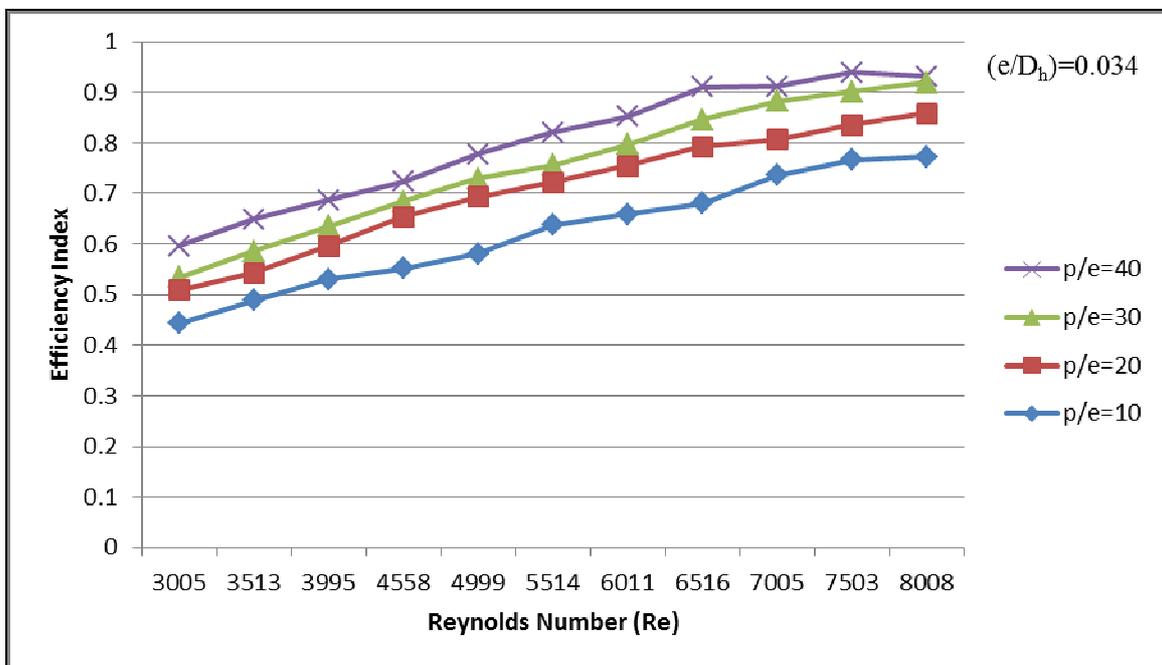The efficiency Index of roughness plate with double arc geometry is plotted in the figure 13.

Figure. 13 Variation of Efficiency Index with Reynolds number for different values of (P/e) and for a fixed values of (e/D$_h$) for Double arc geometry

3.4. Comparison of Transverse, single arc, double arc roughness geometries.

Figure. 14 shows the variation in Nusselt number as a function of Reynolds number for different roughness geometries. Considerable jump in the Nusselt number can be seen in all the cases as compared to the Smooth surface. Plot reveals that the value of Nusselt number is highest in case of Double arc roughness and lowest in case of Transverse ribs for all the values of Reynolds number. On the basis of this observation one can choose preferably double arc shape roughness to achieve higher heat transfer rates. Since Transverse ribs has only one secondary flow cell while single arc got two secondary flow cells and double arc roughness can generate multiple secondary flow cells and therefore double arc roughness shows maximum heat transfer augmentation.

Figure. 14 Comparison of Nusselt number with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10)

Figure 15. Shows the comparison of Friction factor with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10). From the plot it can be observed that the friction factor decreases as the Reynolds number is increased for all the roughness geometries. The friction factor is found to be highest for Double arc roughened plate and minimum for the transvers roughened plate. The friction factor of smooth plate is also give for the comparison.
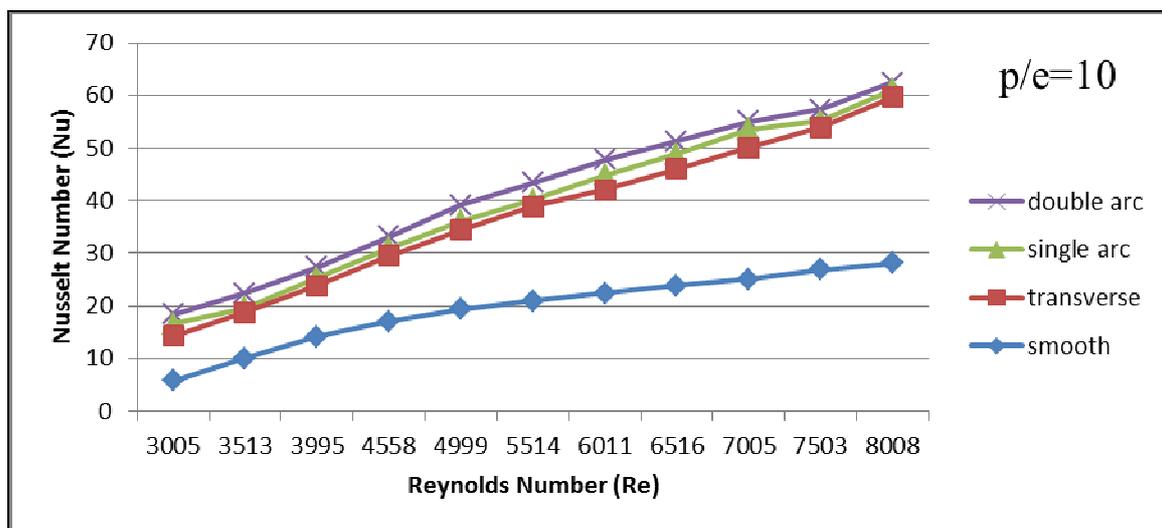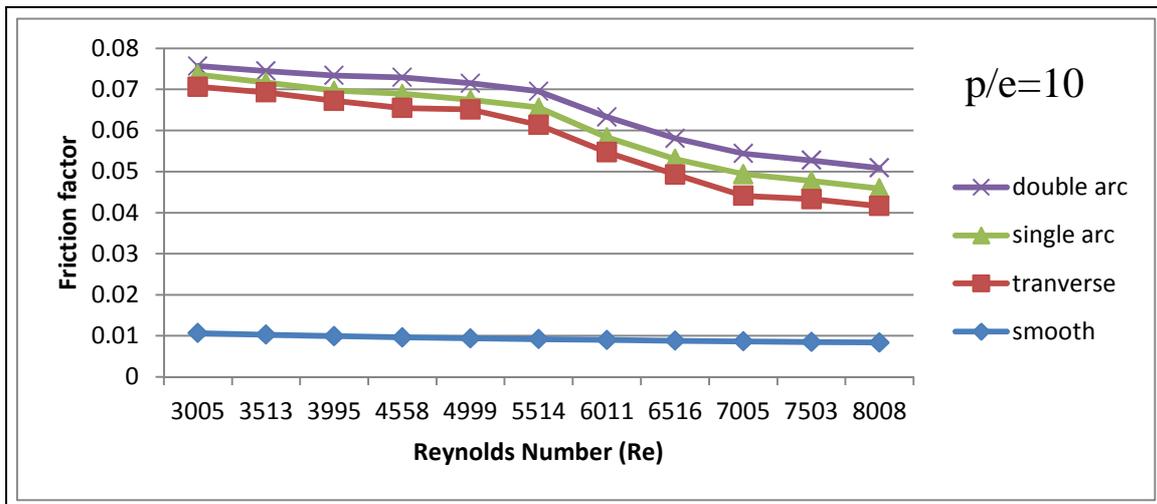
Figure. 15 Comparison of Friction factor with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10)

Figure. 16. shows the comparison of Efficiency Index with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10), from the graph it can be seen that the efficiency is highest for Double arc roughened plate and lowest for Transvers roughened plate. More secondary flows are generated in the Double arc roughened plate than in the Single arc roughened plate and lowest in case of Transvers plate hence the efficiency is highest for Double arc roughened plate.
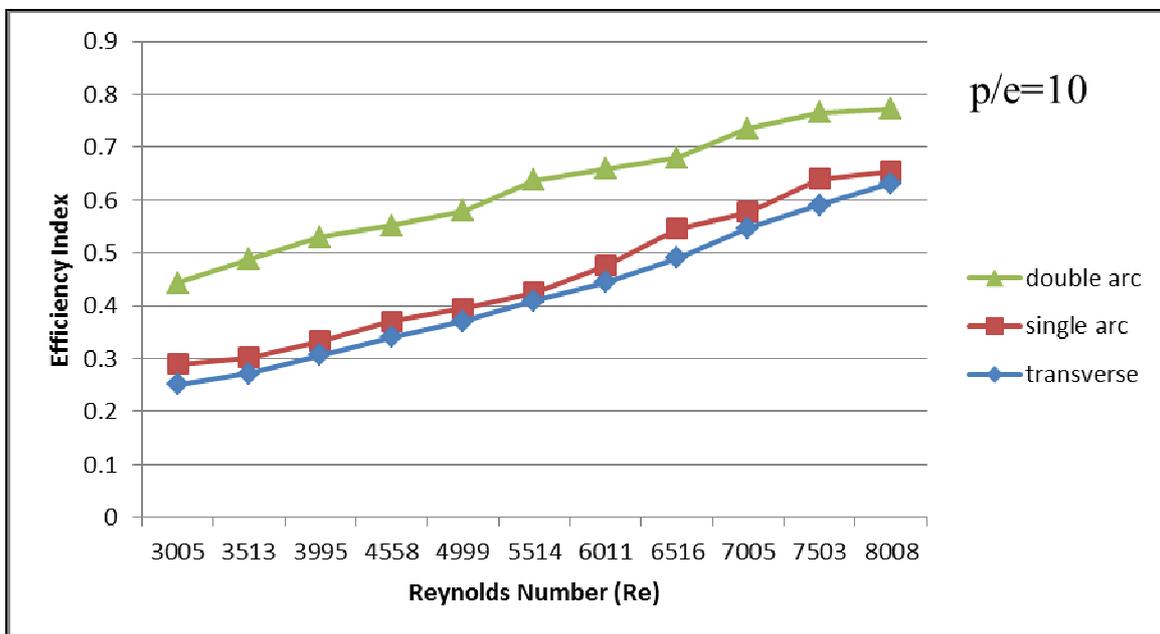

Figure 16. Comparison of Efficiency Index with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10)

4. Conclusion

The present work is undertaken with the objective of experimental investigation of transverse, single arc, double arc shaped roughness geometries created artificially on the absorber plate of solar air heater. Results obtained have been compared with those of smooth duct under similar flow conditions to determine the enhancement in heat transfer coefficient. The following conclusions have been drawn from this investigation.

- Use of artificial roughened surface with different type of roughness geometries is found to be the most effective technique to enhance the heat transfer rates from the heated surface to flowing fluid at the cost of moderate rise in fluid friction.
- In the entire range of Reynolds Number it is found that Nusselt number increases and friction factor decreases with an increase of Reynolds number in all the cases, Nusselt number and friction factor are significantly higher as compared to those obtained for smooth absorber plates. This is due to the distinct change in the fluid flow characteristics as a result of roughness that causes flow separation, reattachment and the generation of secondary flows.
- For all the three roughness geometries (transverse, single arc, double arc) it was found that the Nusselt number increases as Reynold number increases, the maximum value of Nusselt number is found corresponding to relative roughness pitch (p/e) of 10, while the minimum value of Nusselt number corresponds to relative roughness pitch (p/e) of 40. The friction factor decrease as the Reynolds number is increased for all the relative roughness pitch (p/e) value, it is found to be maximum for relative roughness pitch (p/e) of 10 and minimum for (p/e) of 40.
- The Nusselt number decrease as the value of relative roughness pitch (p/e) is increased from 10 to 40 for all the Reynolds number, it is found to be highest for (p/e) value of 10 and for Reynolds number of 8000 and found to be minimum for (p/e) value of 40 and for Reynolds number of 3000. The friction factor also decreases as the relative roughness pitch (p/e) values are increased for all the Reynolds number, it is found to be highest for (p/e) value of 10 and for Reynolds number of 3000 and lowest for (p/e) value of 40 and for Reynolds number of 8000.
- The efficiency index increases as the Reynolds number is increased for all the values of relative roughness pitch (p/e), it was found to be maximum for (p/e) value of 40 and minimum for (p/e) value of 10. Also the efficiency index was found to be increases with increase of (p/e) values and it was found to be maximum for (p/e) of 40 and for higher Reynolds number of 8000, and was found to be minimum for (P/e) value of 10 and for lower Reynolds number of 3000.
- The maximum enhancement of average Nusselt number is found to be 1.12 times for transverse wire geometry, 1.17 times for single arc wire geometry and 1.22 times for double arc wire geometry compared to that of smooth plate.
- The maximum enhancement of average friction factor is found to be 5.61 times for transvers wire geometry, 5.89 times for single arc wire geometry and 6.09 times for double arc wire geometry compared to that of smooth plate.
- The comparison of transvers, single arc, double arc, and smooth absorber plate shows that the Nusselt number, friction factor and efficiency index is highest for double arc shaped roughness followed by single arc, transvers and least for smooth absorber plate.
- It has also been observed that a lower Reynolds number (less than 5000) a smooth plate and an artificially roughened plate have very less variation of heat transfer.
- It can be concluded that the maximum heat transfer is found to be in double arc roughness absorber plate than other shapes.

Nomenclature

A_c,   Area of absorber plate, (m^2)

a_o,   Area of orifice, (m^2)

a_1,   Area of pipe, (m^2)

C_d,   Coefficient of discharge,

c_p,   Specific heat of air, (J/kgk)

D_h,   Hydraulic diameter of duct (m)

e,   Element height (m)

f_r,   Friction factor of roughened plate

f_s,   Friction factor of smooth plate

(e/D_h),   Relative roughness height (dimensionless)

H,   Height of duct, (m)

h,   Convective heat transfer coefficient (W/m^{2}k)

k,   Thermal conductive of air, (W/m k)

L,   Test length (m)

m,   Mass flow rate (kg/s)

Nu_r,   Average Nusselt number of roughned duct   (dimensionless)

Nu_s,   Average Nusselt number of smooth duct (dimensionless)

p/e,   Relative roughness pitch (dimensionless)

Pr,   Prandtl number

q,   Rate of heat transfer to air (W)

Re,   Reynolds number (dimensionless)

t_i,   Temperature of air at inlet (°C)

t_o,   Temperature of air at outlet (°C)

t_f,   Average temperature of fluid (°C)

t_p,   Average temperature of absorber plate (°C)

V,   Velocity of air (m/s)

W,   Width of duct (m)

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