# Analysis of The Performance of Air Heater Using Arc Shaped Turbulators on The Absorber Plate

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***Abstract-*** *Heat energy plays most important role in the field of power generation which involves the heat transfer in domestic as well as industrial purposes. The heat transfer coefficient between heat transferring surface and air is low which leads to lower thermal efficiency of the system. Therefore it is important to increase heat transfer coefficient between heat transferring surface and air. Artificial roughness applied on the absorber plate is the most acclaimed method to improve thermal performance of air heaters at the cost of low to moderate friction penalty. Experimental analysis admissible to rough surface geometries spread that the improvement in heat transfer is accompanied by considerable rise in drain power. This work covers the types of technique used in enhance heat transfer coefficient in field of heat transfer and review a variety of papers dealing with artificial roughness on heat transfer field. Heat absorbers are the special kind of heat exchangers that transform heat flux energy to internal energy of the transport medium. The major component of heater system is the absorber plate. The heat energy is flowing from the circulatory fluid either directly to the air or space conditioning equipment or to a thermal energy storage tank from which can be drawn for use.*

***Keywords***- Artificial roughness, Air heater, Roughness geometry, thermal performance.

## I. INTRODUCTION

Heater is the devices that can be used to transfer heat from solid to fluid (liquid or gas) at different temperature. Electrical heater is devices which exchange of energy take place between heating element to air or liquid at different temperature. A heat exchanger utilized a fact that, whenever there is a difference in temperature flow of energy occurs, heat will flow from higher temperature heat reservoir to lower temperature heat reservoir. The heated surface provides a necessary temperature difference and thus force of energy to flow between them. Heat exchanger are used in different processes raging from conversion, utilization and heat recovery of thermal energy in various industrial, commercial and domestic applications. This include power production, processes, chemical and food reservation, waste heat recovery, manufacturing industry and air conditioning, refrigeration and space application. Examples of heat exchanger that can be found in all homes are heat radiators, the coil in your refrigerator and air conditioner and hot water tank.

The performance of conventional air heater can be substantially improved by number of enhancement technique. The research effort has been devoted to develop an apparatus and performing experiments to define the conditions under which an enhancement technique will improve heat transfer. The goal of enhanced heat transfer is to accommodate high heat fluxes. This reduces air heater size which generally leads to less capital cost. Another advantage is the reduction of temperature driving forces. The heat transfer enhancement enables air heater to work at less speed but still achieve the same or even higher heat transfer coefficient. This means the reduction of pressure drop, corresponding to less operating cost may be achieved. All these advantages have made heat transfer enhancement technology attractive in air heater applications. For air heaters the turbulators wire geometry insert technology, additional exchanger can often avoided and thus significant cost saving can be achieved. Heat transfer rate can be improved by including a disturbance in air flow by different enhancement techniques (breaking the thermal boundary layer). In this project, a review of heat transfer enhancement tool i.e. inserting turbulated wire geometry.

### 1.1 Problem Statement

The problems with smooth surface absorber plate air heaters are excessive time taken to transfer heat to air, which leads to increase electrical energy use and also serve damage to the system due to excessive heating .

For improving heat transfer rate and improving heat transfer coefficient, turbulence created by roughened surface very effective. As a result different turbulence geometry is proposed.

## II. LITERATURE REVIEW

Several investigators have been carried out to study the effect of artificial roughness on the heat transfer and friction factor in solar air heaters. Some of them are discussed below.

**Abdul-Malik Ebrahim Momin et.al**experimentally investigated the effect of geometrical parameters of V-shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of solar air heater with absorber plate having V-shaped ribs on its underside.

In general, Nusselt number increases whereas the friction factor decreases with an increase of Reynolds number. The values of Nusselt number and friction factor are substantially higher as compared to those obtained for smooth absorber plates. This is due to distinct change in the fluid flow characteristics as a result of roughness that causes flow separations, reattachments and the generation of secondary flows. It was observed that the rate of increase of Nusselt number with an increase in Reynolds number is lower than the rate of increase of friction factor; this appears due to the fact that at relatively higher values of relative roughness height, the re-attachment of free shear layer might not occur and the rate of heat transfer enhancement will not be proportional to that of friction factor.

**J.C. Hanet.al** 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. An experimental facility was constructed to determine the temperature variation and the increase in friction factor using uniformly heated plate with pin fins has been carried out. The collector had a rectangular shape with large width to gap ratio and one surface subjected to a uniform heat flux and other parallel plates perfectly insulated. Use of fins increases the collector efficiency. The collector with the fins collected 11% more energy than the collector without fins. But this was accompanied by significant increase in pressure drop.

**N. Sheriff et.al** 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 $(fr / fs) < (Nur/Nus)^3$ .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. Bopcheet.al** 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 heat transfer and friction factor data obtained is compared with the data obtained from smooth duct under similar geometrical and flow conditions. Following conclusions are drawn.

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 et.al**larried 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. Following conclusions are drawn.

In the entire range of Reynolds number it is found that the Nusselt number increases, attains a maximum for roughness pitch of 20 mm and decreases with an increase of roughness pitch. Based on experimentation it is found that the maximum thermal efficiency of roughened solar air heater to of the order of (51–83.5%) depending upon the flow conditions. 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 et.al** 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 results show that the heat transfer increases with increasing the Reynolds number for smooth and roughened absorber plates, because of the higher turbulences. In the entire range of Reynolds number it is found that the Nusselt number increases, with increasing Reynolds number. Nusselt number increases, with increasing relative roughness pitch up to 10. It attains a maximum value at for relative roughness pitch of 10 and then decreases with an increase of relative roughness pitch beyond 10.

## III. OBJECTIVE

1. In this work it is intended to investigate experimentally the effect of wire geometry (transverse wires) of thin circular wire used as artificial roughness on heat transfer coefficient and friction factor in air ducts.
2. Also it is planned to investigate experimentally the effect of thin wires (arc shaped) with multiple arcs as artificial roughness on heat transfer coefficient and friction factor in air ducts.
3. Effect of relative roughness pitch (p/e) 10 to 40, for a fixed parameter relative roughness height (e/Dh) 0.034 and Reynolds number (Re) range from 3000 to 8000 for above roughness shapes are planned to be determined.
4. Finally different roughness configurations will be compared with smooth surface.

## IV. CONCEPT OF ARTIFICIAL ROUGHNESS

One of the most promising and economical method of increasing the heat transfer rate is providing the artificial 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 surfaces 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 thevicinity 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 producing such turbulence has to come from the pumping devices and the extra power is required to flow air through the empty space provided. 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 providing the height of the roughness elements to be minimum in comparison with the passage dimensions.

The key dimensionless geometrical parameters that are used to characterize roughness are:

1. Relative roughness pitch (p/e):
   Relative roughness pitch (p/e) is defined as the ratio of gap between the two consecutive ribs to the height of the ribs.
2. Relative roughness height (e/d):
   Relative roughness height (e/d) is the ratio of rib height to equivalent diameter of the air passage.
3. Angle of attack:
   Angle of attack is inclination of rib with direction of air flow in passage.
4. Shape of roughness element:
   The roughness elements can be multi-dimensional ribs or multi-dimensional distinct elements, inclined ribs or V-shaped continuous or discontinuous ribs with gap or without gap. The roughness elements can also be C shaped wire or dimple or empty passage or compound rib grooved. The common shape of ribs is square but different shapes like circular, semi-circular and chamfered have also been considered to investigate thermo hydraulic performance.
5. Aspect ratio:
   It is ratio of duct width to duct height. This factor also plays a very important role in enhancing thermo-hydraulic characteristics.

## V. EXPERIMENTAL DETAILS

The experimental setup is shown in Figure 1. The setup consists of smooth entrance section, roughened entrance section, test section, an exit section, mixing chamber, a flow meter and an air blower

### 5.1. Experimental Apparatus

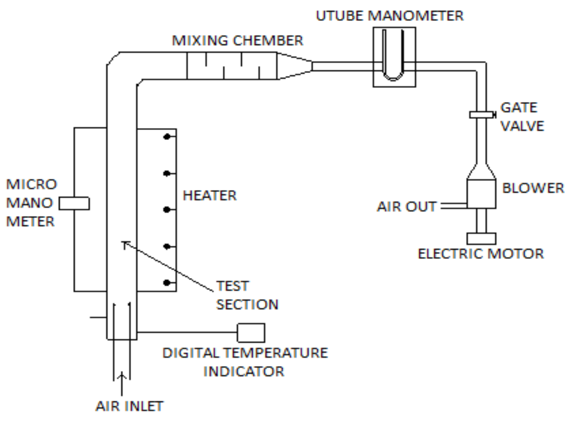


**Figure 5.1**Experimental Setup

### 5.1.1Air duct

Duct 2600mm x 150mm x 30mm
Test Section 1200mm x 150mm x 30mm
Entry & Exit Length 800mm & 600mm

**5.1.2 Absorber plate**

Aluminum 2mm thick 1200mm x 150mm

**5.1.3 Heater**

Five electric bulbs each of 100 watt

**5.2 Instrumentations**

**5.2.1Temperature measurement**

Calibrated chrome-alumel thermocouples with digital micro-Volt-millimeter, indicating output in degree centigrade with an accuracy of 0.1 °C

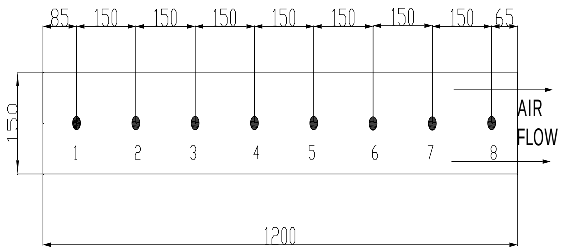


**Figure5.2**Position of thermocouples on absorber plate

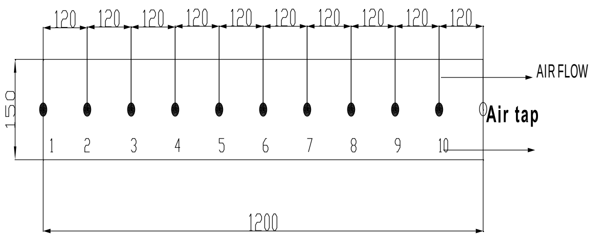


**Figure 5.3**Position of thermocouple inside air duct

**5.2.2 Air Flow Measurement**

The mass flow rate of air was measured by means of a flange type orifice meter calibrated by usingPitot tube. The control valves were provided in the line to control the flow.

**5.2.3 Pressure Drop measurement inside the duct**

Micro-manometer is used for measuring the pressure drop in the test section. The least count of micro- The micro manometer consists of a relatively large reservoir of fluid the meniscus of the liquid in glass tube is maintained at a fixed prescribed mark by sliding the tube up or down. This movement yields the pressure difference across the two pressures tapping (across the test length). The pressure tapping fixed at outlet of the test length is connected to the reservoir of the micro-manometer. Diesel having specific gravity 0.8373 is used in the manometer to increase the accuracy further.

**5.3 Roughness geometries used in the study**

**5.3.1 Transverse ribs in the form of small diameter wires**

Artificial roughness in the form of transvers ribs were used on the underside of absorber plate for flow separation and for creating turbulence. Experimentation was conducted to predict the effect of various roughness parameters such as relative roughness pitch, (p/e) on the heat transfer and friction factor for various Reynolds number (Re) for a fixed relative roughness height (e/D).

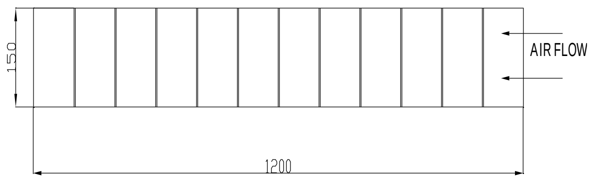


**Figure 5.4** Schematic diagram of transverse wire roughness geometry on absorber plate

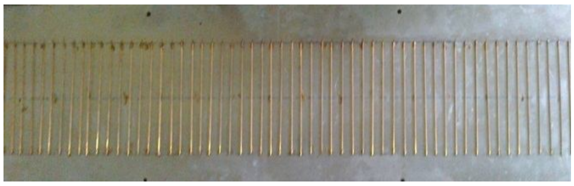

**Figure 5.5** Photograph of transverse wires roughness geometry (p/e=10) on absorber plate

**5.3.2 Single arc shape roughness**

Further the roughness in the form of Arc shapes were attached to underside of the absorber plate. The diameter of the arc was 150mm. Roughness parameters such as relative roughness pitch, (p/e) was varied for different values of Reynolds number and the data was recorded.

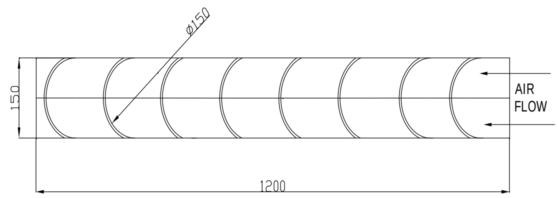


**Figure 5.6** Schematic diagram of single arc wire roughness geometry on absorber plate

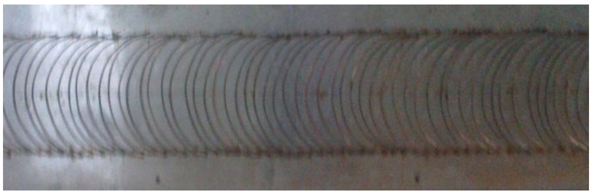

**Figure5.7** Photograph of single arc wire roughness geometry (p/e=10) on absorber plate

### 5.3.3 Double Arc shape Roughness

Double Arc shaped roughness was used to determine the heat transfer and friction. the diameter of each arcs are 75 mm, roughness parameters such as relative roughness pitch, (p/e) was varied for different values of Reynolds number and the data was recorded.

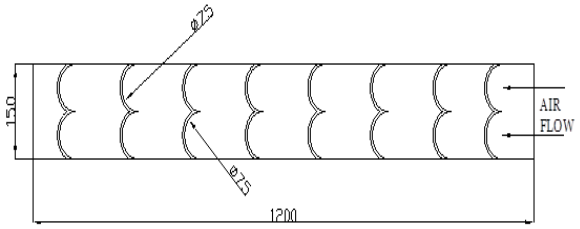


**Figure5.8** Schematic diagram of double arc wire roughness geometry on absorber plate\

**Figure5.9** Photograph of double arc wire roughness geometry (p/e=10) on absorber plate.

## VI. EXPERIMENTAL PROCEDURE

The test runs to collect relevant heat transfer and flow friction 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 are conducted in order to cover the entire range of Reynolds number. The following parameters are measured during experimentation:

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

The range of parameters considered under the experimental study is as given in Table.

**TABLE 6.1** 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 | 1.7 |
| 5. | Hydraulic diameter of duct | 50 |
| 6. | Relative roughness height, e/D | 0.034 |
| 7. | Relative roughness pitch, p/e | 10 to 40 |

### 6.1 Mean air and plate temperatures

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

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

Where,

T1-T10 is the temperature of air at various locations of air duct.

The mean plate temperature, (Tp) is the averagetemperature of absorber plate at eight differentlocations.

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

Where,
T11-T18 is the temperature of plate at various location of absorber plate.

### 6.2 Mass flow rate

The mass flow rate is calculated as follows,

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

Where,
ρ, Density of air, kg/m3
$c_d$, Coefficient of discharge, 0.62
$a_o$, area of orifice, m2
$a_1$, area of pipe, m2
$H = h_m(\rho_w|\rho)$
$h_m$, is manometer reading

**6.3 Velocity of air**

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

$$V = \frac{m}{\rho WB}$$

Where,
ρ, Density of air, kg/m3
W, width of duct, m
B, height of duct, m

**6.4 Heat transfer rate**

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

$$q = mc_p(t_o - t_i)$$

Where,
m, mass flow rate, kg/s
$c_p$, Specific heat of air, J/Kg k
$t_o$, temperature of air at outlet,°C
$t_i$, temperature of air at inlet, °C

**6.5 Convective heat transfer co efficient**

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

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

Where,
$A_c$, area of absorber plate, m2
q, heat transfer rate
$t_p$, average temperature of absorber plate,
$t_f$, average temperature of fluid, °C

**6.6 Nusselt number**

The Nusselt number is calculated from the following expression,

$$Nu_r = \frac{hD_h}{k}$$

Where,

h, Convective heat transfer coefficient, (W/m2k)
k, thermal conductive of air, W/m K
Dh= Hydraulic diameter $= \frac{4WH}{2(W+H)}$

**6.7 Reynolds number**

The Reynolds number is calculate as follows,

$$R_e = \frac{VD_h}{\nu}$$

Where,

V, velocity of air, m/s
ν= kinematic viscosity,m2/s

**6.8 Friction factor**

The friction factor was determined from the following equations,

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

Where,

$\Delta p$, Pressure drop
L= test length, 1200mm

## VII. RESULTS AND DISCUSSION

**7.1 Comparison of Transverse, Single arc, Double arc roughness geometries**

Figure 7.19 shows the variation in Nusselt number as a function of Reynolds number for different roughness geometries.. 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.

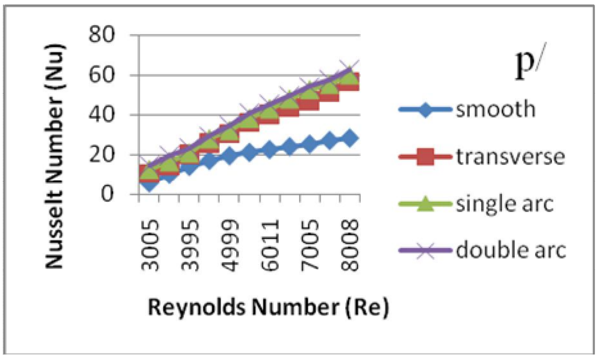


**Figure7.1**Comparison of Nusselt number with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10)

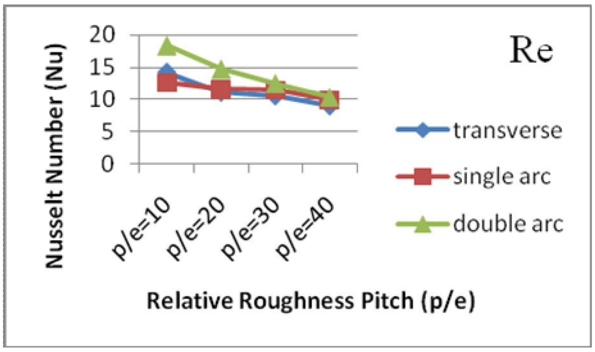


**Figure7.2**Comparison of Nusselt number with Relative Roughness pitch (P/e) for different roughness geometries for fixed value of Reynolds number

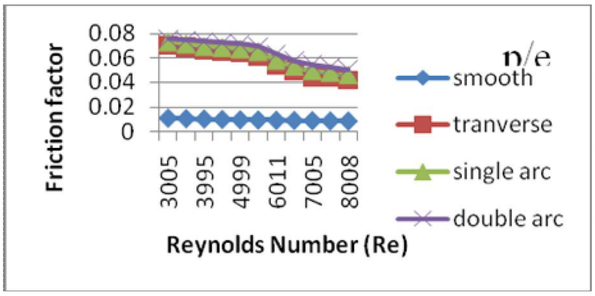


**Figure7.3**Comparison of Friction factor with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10)

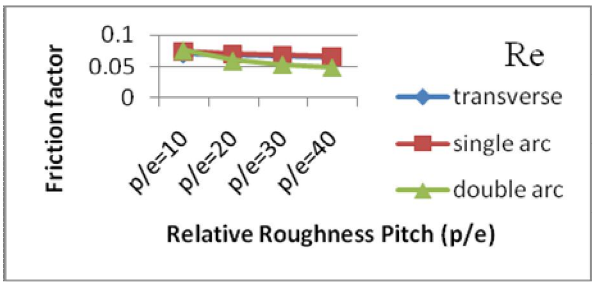


**Figure7.4**Comparison of Friction factor with Relative Roughness pitch (P/e) for different roughness geometries for fixed value of Reynolds number

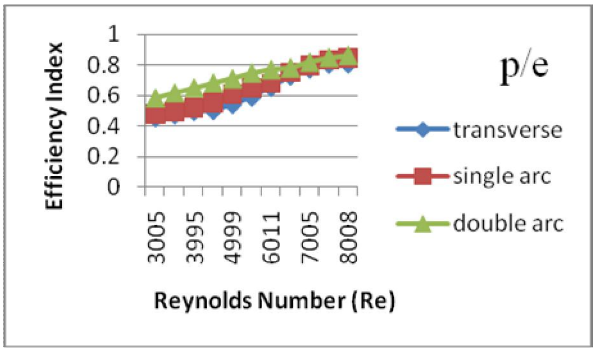


**Figure 7.5**Comparison of Efficiency Index with Reynolds number for different roughness geometries for fixed value of Relative roughness pitch (p/e=10)

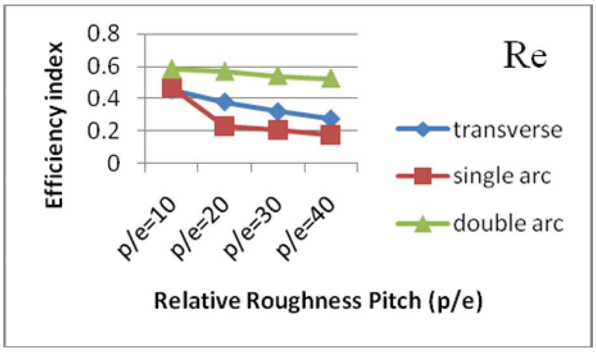


**Figure 7.6**Comparison Efficiency Index with Relative Roughness pitch (P/e) for different roughness geometries for fixed value of Reynolds number

## VIII. CONCLUSIONS

The experimental work with investigation of different wire geometry (transfers, single arc, double arc) on the absorber plate of air heater. Obtained results are compared to determine improved heat transfer coefficient with smooth duct.

The following conclusions have been drawn from this investigation.

1) Forflowing fluid with, moderate rise in fluid friction, use of artificial roughness surface with different type of roughness geometries is found to be most effective.
2) As Nusselt number increases and friction factor decreases with increasing Raynold number in all conditions under entire range of Raynold number, it is concluded that Nusselt number and friction factor are comparatively high compared smooth absorber plate because roughness causes flow separation, reattachment and generation of secondary flows.
3) Nusselt number increases with increase in Raynold number for all the three roughness geometries

(transverse, single arc, double arc). The maximum and minimum values of Nusselt number are found corresponding to relative roughness pitch (p/e) of 10 and (p/e) of 40 respectively. The friction factor is inversely proportional to Raynold number for all relative roughness pitch (p/e) values, it is found to be maximum and minimum for relative roughness pitch (p/e) of 10 (p/e) of 40 respectively.

4) The Nusselt number reduces if the value of relative roughness pitch (p/e) is increased from 10 to 40 for all the Reynolds number, it is found to be maximum and minimum for (p/e) value of 10 and for Reynolds number of 8000 and for (p/e) value of 40 and for Reynolds number of 3000 respectively. The friction factor also inversely proportional to the relative roughness pitch (p/e) values for all the Reynolds number, it is found to be highest and lowest for (p/e) value of 10 and for Reynolds number of 3000 and (p/e) value of 40 and for Reynolds number of 8000 respectively.
5) The Nusselt number,friction factor and efficiency index is maximum and minimum fordouble arc shape roughness geometry followed by single arc, transverse and for smooth absorber plate respectively.
6) The smooth plate and roughened plate have very minor variation of heat transfer for smaller than 5000 Raynold number value.
7) It can be concluded that the maximum is the double arc roughness absorber plate than other shapes.

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