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Effect of Various Artificial Roughness Parameters on Heat Transfer and Friction Characteristics for Flow inside Rectangular Ducts of Solar Air Heater

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Abstract: Heat transfer and friction correlations are developed for fully developed turbulent flow in rectangular ducts having repeated integral wire ribbed roughness on the absorber plate. Computer program is developed using 'C++' programming language to determine the effect of various wire roughness parameters on heat transfer and friction for flow inside rectangular ducts with roughened absorber. The study covers wide range of different parameters of wire ribbed roughness such as relative roughness pitch (p/e) from 10 to 40, relative roughness height (e/D_h) from 0.01 to 0.04 and angle of attack of flow (α) from 20° to 90°. Flow parameter Reynolds number (Re) is varied from 5,000 to 30,000 and duct aspect ratio (W/B) is varied from 2 to 10. Program can be used for iterative work to identify the optimum parameters of roughness and duct to enhance thermal performance.

Keywords: Heat Transfer; Friction factor; Wire ribbed absorber; Stanton number; Nusselt number; Efficiency index.

I. INTRODUCTION

Solar air heaters form the major component of solar energy utilization system, which absorbs the incoming solar radiation, converting it into thermal energy at the absorbing surface, and transferring the energy to a fluid flowing through the collector. The efficiency of solar air heaters has been found to be low because of low convective heat transfer coefficient between absorber plate and the flowing air, which increases the absorber plate temperature, leading to higher heat losses to the environment.

Several methods including the use of fins, ribbed roughness and packed beds in the ducts, have been proposed for the enhancement of thermal performance of these collectors. Use of roughness in the form of repeated ribs has been found to be a convenient method for enhancing the heat transfer to fluid flowing in the duct.

This is an investigation made to improve the performance of solar air heaters by using wire ribbed roughness on the absorber plate. Solar air heaters with wire roughness on the absorber perform better than the plane ones under the same operating conditions. The parameters influencing the heat transfer characteristics include Reynolds number (Re), angle of attack of flow (α), relative roughness pitch (p/e), relative roughness height (e/D_h) and the aspect ratio (W/B) of the air heater duct. The range of parameters for this study has been decided on the basis of practical considerations of the system and operating conditions.

Thus, even a marginal increase in the efficiency of air heaters can reduce the cost of pumping power and effective area required to collect the fixed quantum of heat energy. In this study, wire ribs of various parameters are used in the computer program and performance of such systems has been investigated. This software program developed using C++ language can be used for iterative work to find the best optimum design.

The purpose of using wire ribbed artificial roughness on the absorber surface is to make the flow turbulent adjacent to the wall that is laminar sub layer region. Many investigations have been made to select a range of roughness elements, which reduces friction losses and at the same time enhances the heat transfer. Abdul-Malik et al. [1] conducted an experimental investigation of the effect of geometrical parameters of V-shaped ribs on heat transfer and flow characteristics of rectangular duct of solar air heater with absorber plate having V-shaped ribs on its underside.

The range of parameters for this study has been decided on the basis of practical considerations of the system and operating conditions. The investigation has covered various parameters such as Reynolds number, relative roughness height, angle of attack of flow and relative roughness pitch. Results have also been compared with those of smooth duct under similar flow conditions to determine the enhancement in heat transfer coefficient and friction factor. They also developed expressions for heat transfer and friction factor for fully developed turbulent flow in a solar air heater duct. They found increase in both Nusselt number and friction factor with increase in roughness height. J.C. Han [2] carried out an experimental study of fully developed turbulent air flow in square ducts with two opposite rib roughened walls and observed that the average friction factor was 2.1 to 6 times that for four sided 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. R.L. Webb et al. [3] conducted a comparative study between the roughened tubes and smooth tubes in design of heat exchangers. This study is conducted mainly to achieve enhancement of heat transfers capacity and to reduce the friction factor. R.L. Webb et al. [4] developed heat transfer and friction correlations for turbulent flow in tubes having repeated rib roughness. The correlations are verified with experimental data taken with relative roughness height 0.01 to 0.04 and relative roughness pitch 10 to 40 and Prandtl number 0.71 to 37.6. N. Sheriff et al. [5] investigated experimentally the heat transfer and friction characteristics of a surface with discrete roughness. It is shown that pumping power required to force the fluid for same heat transfer surface and fluid temperature difference, will be minimum when $(f_r/f_s) < (St_r/St_s)^3$. This shows any increase in the friction factor increases the heat transfer characteristics of roughened surface resulting in a more efficient heat transfer surface. E.M. Sparrow et al. [6] conducted experiments to determine the heat transfer, pressure drop and flow field responses to the rounding of the peaks of a corrugated wall duct. J.C. Han et al. [7] investigated effects of heat transfer and friction for rib roughened surfaces.

Correlation for friction factor and heat transfer was developed to account for rib shape, spacing and angle of attack. Ribs at 45^o angle of attack are found to have superior heat transfer performance at a given friction power when compared to ribs at a 90^o angle of attack of flow. D.L. Gee et al. [8] contributed experimental information for single phase forced convection in circular tube containing a two dimensional rib roughness. In their investigation they have used a parameter 'efficiency index' η, which can be defined as the ratio of enhancement factor of heat transfer coefficient to that of friction coefficient. $\eta = (St_r/St_s)/(f_r/f_s)$. M.J. Lewis [9] carried out an elementary analysis for predicting the momentum and heat transfer characteristics of a hydraulically rough surface. In this work analytical model for flow over rough surface is developed. Santosh B. Bopche et al. [10] conducted experimental investigations on heat transfer and frictional characteristics of a turbulators roughened solar air heater duct. In their study they used artificial roughness in the form of specially prepared inverted U-shaped turbulators on the absorber surface of air heater duct. This roughened wall is uniformly heated while the other three walls are insulated. They showed that roughened duct enhances the heat transfer and friction factor by 2.82 and 3.72 times than the smooth duct respectively.

R. Kamali et al. [11] conducted study on the importance of rib shape effects on the local heat transfer and flow friction characteristics of square ducts with ribbed internal surfaces. They developed program to study the turbulent heat transfer and friction in a square duct with various-shaped ribs mounted on one wall. The results show that features of the inter-rib distribution of the heat transfer coefficient are strongly affected by the rib shape and trapezoidal ribs with decreasing height in the flow direction provide higher heat transfer enhancement and pressure drop than other shapes. P. Bharadwaj et al. [12] experimentally determined pressure drop and heat transfer characteristics of flow of water in a spirally grooved tube with twisted tape insert. Compared to smooth tube, the heat transfer enhancement due to spiral grooves is further augmented by inserting twisted tapes. Rajendra Karwa et al. [13] presented results of an experimental study of heat transfer and friction in a rectangular section duct with perforated baffles. The baffled wall of the duct is uniformly heated while the remaining three walls are insulated. These boundary conditions correspond closely to those found in solar air heaters. Their study shows an enhancement of 79-169% in Nusselt number over the smooth duct for the fully perforated baffles and 133-274% for the half perforated baffles while the friction factor for the fully perforated baffles is 2.98-8.02 times of that for the smooth duct and is 4.42-17.5 times for the half perforated baffles. Sharad Kumar et al. [14] investigated the performance of a solar air heater duct provided with artificial roughness in the form of thin circular wire in arc shaped geometry has been analyzed using Computational Fluid Dynamics (CFD). The effect of arc shaped geometry on heat transfer coefficient, friction factor and performance enhancement was investigated covering the wide range of roughness parameters such as relative roughness height, relative roughness angle and working parameter such as Reynolds number. Different turbulent models have been used for the analysis and their results are compared. The overall enhancement ratio has been calculated in order to discuss the overall effect of the roughness and working parameters. A maximum value of overall enhancement ratio has been found to be as 1.7 for the range of parameters investigated. M.K. Gupta et al. [15] conducted a parametric study of artificial roughness geometry of expanded metal mesh type in the absorber plate of solar air heater duct and compared with smooth duct. The performance evaluation in terms of energy augmentation ratio, effective energy augmentation ratio and exergy augmentation ratio has been carried out for various values of Reynolds number and roughness parameters of expanded metal mesh roughness geometry in the absorber plate of solar air heater duct. M.E. Taslim et al. [16] have shown that the enhancement in heat transfer coefficients for airflow in a channel roughened with angled ribs is on the average higher than that roughened with 90° ribs of the same geometry. Secondary flows generated by the angled ribs are believed to be responsible for these higher heat transfer coefficients. Heat transfer coefficients and friction factors are compared with

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those of 90° and 45° and discrete angled ribs. Test results show that 90° ribs represent the lowest thermal performance compared to other angled ribs.

II. ANALYSIS OF FLUID FLOW AND HEAT TRANSFER

Absorber plate is the only surface in solar air heaters which absorbs the incoming solar radiation and converts it into thermal energy and transfers the energy to the flowing fluid. Solar air heaters usually have ducts with wide aspect ratio. In this study duct with aspect ratio ranging from 1 to 10 is selected for analysis. To increase heat transfer in solar air heater, ribbed roughness are considered.

Parameters selected include the height of wire (e), pitch of the roughness element (p), Reynolds number (Re), relative roughness pitch and height (p/e & e/D_h). Prandtl number represents the thermo physical property of the fluid.

A. Fluid Flow Analysis

J.C. Han [2] developed an expression for the friction factor by considering the case of present solar air heater duct of rectangular cross section with W>>B.

$$f_s = \tau_s/[(½)\rho V_s^2] \quad (1)$$

The above relation gives friction factor for smooth ducts for fully developed turbulent flow. Similarly friction factor for a four-sided rough duct for fully developed turbulent flow can be given by,

$$f_r = \tau_r/[(½)\rho V_r^2] \quad (2)$$

The average friction factor for this case is be expressed as,

$$f_{av} = \tau_{av}/[(½)\rho V^2] \quad (3)$$

In above equation average shear stress can be related to shear stresses produced by three smooth and one rough side wall will be given by,

$$\{(W+2B)\tau_s+W\tau_r\}L = \{(W+2B)\tau_s+W\tau_r\}L \quad (4)$$

The average shear stress can be related as,

$$\{(2W+2B)\ \tau_{av}\}L = \{(W+2B)\ \tau_s+ W\ \tau_r\}L \quad (5)$$

On simplifying,

$$(2W+2B)\ \tau_{av} = \{(W+2B)\ \tau_s+ W\ \tau_r\} \quad (6)$$

$$\text{Assuming, } ½\ \rho V_s^2 = ½\ \rho V_r^2 = ½\ \rho V^2 \quad (7)$$

$$\text{Simplifying, } V_s^2 = V_r^2 = V^2 \quad (8)$$

Rearranging above equation,

$$(2W+2B)\ f_{av} = \{(W+2B)\ f_s+ W\ f_r\} \quad (9)$$

$$f_{av} = \{(W+2B)\ f_s+ W\ f_r\} / (2W+2B) \quad (10)$$

Further simplifying,

$$f_{av} = \{[(W/B)+2]f_s+(W/B)f_r\}/2[(W/B)+1) \quad (11)$$

From the above equation the expression for friction factor for the present solar air heater duct can be obtained. The friction factor (f_s) can be calculated from the well-known Blasius equation for Reynolds number (Re) ranging from

$$5000 \text{ to } 30{,}000 \text{ by, } f_s = 0.079\ Re^{-0.25} \quad (12)$$

The friction factor for a four-sided rough duct (fr) can be calculated using the friction similarity law. Considering first, the friction factors for geometrically similar roughness, the basic assumptions used are the velocity defect law and law of the similarity mentioned earlier.
Assuming a region of overlap, the velocity defect law and the law of the wall are combined to give the following expressions for turbulence dominated part of wall region.

$$u^+ = 2.5 \ln (y/e) + R_M(e^+) \quad (13)$$

According to law of wall similarity and similarity law covering a wide range of e/D_h as,

$$R_M(e^+) = (2/f_r)^{½} + 2.5 \ln (2e/D_h) + 3.75 \quad (14)$$

Where e+ is the roughness Reynolds number (Re) given

$$\text{by, } e^+ = (e/D_h)\ Re(f_r/2)^{½} \quad (15)$$

The momentum transfer roughness function (R) is a boundary condition, which represents the dimensionless velocity (u^+) at a distance (e) from the wall. This is a general function determine empirically for each type of roughness present on the surface. The roughness function for sand grain roughness has the constant value of 8.48 when e+ is greater than roughness 70 that is in completely rough regime $R_M(e^+)$ does not depend on e^+. Based on friction similarity law D.L. Gee et al. [8] found a successful friction correlation for turbulent flow through tubes with repeated rib-roughness, by taking in to account the geometrically non similar roughness parameter (p/e), the correlation can be given by following equation

$$\text{for}(p/e)=10,\ e+=35.\ R_M(e^+)=0.95(p/e)^{0.53} \quad (16)$$

The above equation is used to correlate the data of present study, accounting for the angle of attack of flow (α) using Power law dependency as given by D.L. Gee et al. [8] as,

$$R_M(e^+)=[(2/f_r)^{½}+2.5\ln(2e/D_h)+3.75](\alpha/50)^{0.16} \quad (17)$$

Angle of attack of flow (α) is made non-dimensional parameter by dividing by 50°, as optimum condition prevailed from the experimental investigations of D.L. Gee et al. [8] and the correlation used by them is very well

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correlated. The present work used 50^o to make the angle of attack of flow non-dimensional in the expressions as the experimental work indicated optimum performance at 50^o. Thus, from the above equations,

$$0.95(p/e)^{0.53}=[(2/f_r)^{½}+2.5\ln(2e/D_h)+3.75](\alpha/50)^{0.16} \quad (18)$$

Further equation for friction factor for roughened duct can be obtained as

$$f_r=2/[0.95(p/e)^{0.53}(\alpha/50)^{-0.16}+2.5\ln(D_h/2e)-3.75]^2 \quad (19)$$

In this way by substituting the values of (f_s) and (f_r) in equation (11), we can calculate the average friction factor for the present solar air heater duct.

B. Heat Transfer Analysis

The prediction method for heat transfer will be very much similar to that of friction factor as described above if the heat and momentum transfer analogy is assured valid. J.C. Han [2] developed an expression for the Stanton number for fully developed turbulent flow in four sided smooth ducts and four sided roughened ducts, can be defined by equations (21) and (22).

$$St_s = q_s / [GC_p(T_w - T)_s] \quad (20)$$

Where, G = ρV (Product of density of air in kg/m^3 and average velocity of air m/s)

$$St_r = q_r / [GC_p(T_w - T) \quad (21)$$

Similarly for one sided roughened and three sided smooth duct Stanton number can be express as,

$$St_{av} = q_{av} / [GC_p(T_w - T)_{av}] \quad (22)$$

If the analogous assumption for heat transfer follows the same form as was made for friction factor then,

$$\{(W+2B)q_s+Wq_r\}L = \{(W+2B)q_s+Wq_r\}L \quad (23)$$

The average heat flux can be related as,

$$\{(2W+2B)\ q_{av}\}L = \{(W+2B)\ q_s+ W\ q_r\}L \quad (24)$$

On simplifying,

$$(2W+2B)\ q_{av} = \{(W+2B)\ q_s+ W\ q_r\} \quad (25)$$

Assuming,

$$GC_p(T_w - T)_s = GC_p(T_w - T)_r = GC_p(T_w - T)_{av} \quad (26)$$

Simplifying, $(T_w - T)_s = (T_w - T)_r = (T_w - T)_{av}$ (27)

Rearranging above equation,

$$(2W+2B)\ St_{av} = \{(W+2B)\ St_s+ W\ St_r\} \quad (28)$$

$$St_{av} = \{(W+2B)\ St_s+ W\ St_r\} / (2W+2B) \quad (29)$$

Further simplifying,

$$St_{av} = \{[(W/B)+2]St_s+(W/B)St_r\}/2[(W/B)+1] \quad (30)$$

The Stanton number (St_s) can be calculated from the well-known Dittus-Boelter equation.

$$St_s = 0.023/(Re^{0.2}\ Pr^{0.6}) \quad (31)$$

Expression for the Stanton number of roughened duct can be obtained from the following analysis. The heat transfer similarity law has been developed to correlate the heat transfer data for fully developed turbulent flow in tubes with rough sand coated walls and this similarity method is extended to correlate heat transfer data for turbulent flow in rib roughened tubes by D.L. Gee et al. [8]

It is assumed that the same method can be applied for flow in a four sided ribbed duct using the heat and momentum transfer analogy, giving a dimensionless temperature profile normal to the ribbed wall as

$$T^+ = 2.5\ \ln(y/e) + G_H(e^+,Pr) \quad (32)$$

Where by definition, $T^+ = [(T_w - T)\ \rho C_p u]/q_r$ (33)

Where $u = (\tau/\rho)^{1/2}$ i.e friction velocity in m/s. Integration of above equation over the flow channel cross-section

produces, $T_{av}^+ = (2/f_r)^{½} - R_M(e^+) + G_H(e^+,Pr)$ (34)

Where the dimensionless average temperature T_{av}^+ profile

can be expressed by, $T_{av}^+ = (f_r/2)^{½} / St_r$ (35)

Substituting (36) in (35) yields,

$$G_H(e^+,Pr)=R_M(e^+)+\{[(f_r/2St_r)-1]/(f_r/2)^{½}\} \quad (36)$$

Assuming that the heat transfer roughness function $G_H(e^+,Pr)$ is independent of flow channel cross-section and it is only dependent on roughness geometry. The $G_H(e^+,Pr)$ measured in tube flow by D.L. Gee et al. [8] can be can be employed in duct flow for similar geometry that is,

$$G_H(e^+,Pr) = 4.5\,(e^+)^{0.28}\ Pr^{0.57} \quad (37)$$

The above equation is valid for $e^+ \geq 25$. Now to account for the angle of attack of flow (α) in the heat transfer roughness function can be modified by the power law dependency as given by D.L. Gee et al. [8].

$$G_H(e^+,Pr) = \{R_M(e^+)+[(f_r/2St_r)-1]/(f_r/2)^{½}\}\ (\alpha/50)^j \quad (38)$$

Where, j = 0.37 for $\alpha \leq 50^o$ or j = - 0.15 for $\alpha > 50^o$.
Combining the above equations (38) and (39) we get,

$$St_r = (f_r/2)/\{1+(f_r/2)^{½}[4.5\,(e^+)^{0.28}Pr^{0.57}(\alpha/50)^{-j}- R_M(e^+)]\} \quad (39)$$

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In this way by substituting the values of (St_s) and (St_r) in equation (30), we can calculate the average Stanton number for the present solar air heater duct. Similarly Nusselt number for smooth and roughened duct can be calculated by expressions (40) and (41).

$$Nu_s = St_s \, Re \, Pr \qquad (40)$$

$$Nu_r = St_r \, Re \, Pr \qquad (41)$$

The average Nusselt number for one sided roughened and three side smooth duct can be given as,

$$Nu_{av} = St_{av} \, Re \, Pr \qquad (42)$$

Equations obtained are used to predict the values of average friction factor and Stanton number of the duct and these values are compared with those for smooth duct. An efficiency index (η) is used as an optimization parameter and is defined as ratio of enhancement factor of heat transfer coefficient to that of friction coefficient.

$$\eta = (St_{av}/St_s) \,/\, (f_{av}/f_s) \qquad (43)$$

III.COMPUTER AIDED ANALYSIS

The following correlations are developed for heat transfer coefficient and friction factor for both smooth and roughened duct.

Topside of the duct is roughened with artificial roughness and the remaining three sides are smooth and insulated. In this investigation small diameter wires are considered as the roughness elements.

a) $f_s = 0.079 \, Re^{-0.25}$

b) $f_r = 2/[0.95(p/e)^{0.53}(\alpha/50)^{-0.16} + 2.5\ln(D_h/2e) - 3.75]^2$

c) $f_{av} = \{[(W/B)+2]\, f_s + (W/B)\, f_r\} \,/\, 2[(W/B)+1)$

d) $e^+ = (e/D_h)\, Re(f_r/2)^{½}$

e) $R_M(e^+) = [(2/f_r)^{½} + 2.5\ln(2e/D_h) + 3.75](\alpha/50)^{0.16}$

f) $St_s = 0.023/(Re^{0.2}\, Pr^{0.6})$

g) $St_r = (f_r/2)/\{1+(f_r/2)^{½}[4.5 \, (e^+)^{0.28} Pr^{0.57} (\alpha/50)^{-j} - R_M(e^+)]\}$

h) $St_{av} = \{[(W/B)+2]St_s + (W/B)St_r\}/2[(W/B)+1)$

i) $Nu_s = St_s \, Re \, Pr$

j) $Nu_r = St_r \, Re \, Pr$

k) $Nu_{av} = St_{av} \, Re \, Pr$

l) $\eta = (St_{av}/St_s) \,/\, (f_{av}/f_s)$

The investigations are carried out for Reynolds number (Re) range of 5000-30000, angle of attack of flow (α) of 20° - 90°, relative roughness pitch (p/e) of 10-40, relative roughness height (e/D_h) of 0.01-0.04 and aspect ratio (W/B) of 1-10 for a fixed value of Prandtl number (Pr) of 0.71. The effect of different parameters on the friction factor, Stanton number and efficiency index are analyzed. Flow chart is shown in Fig. 1.

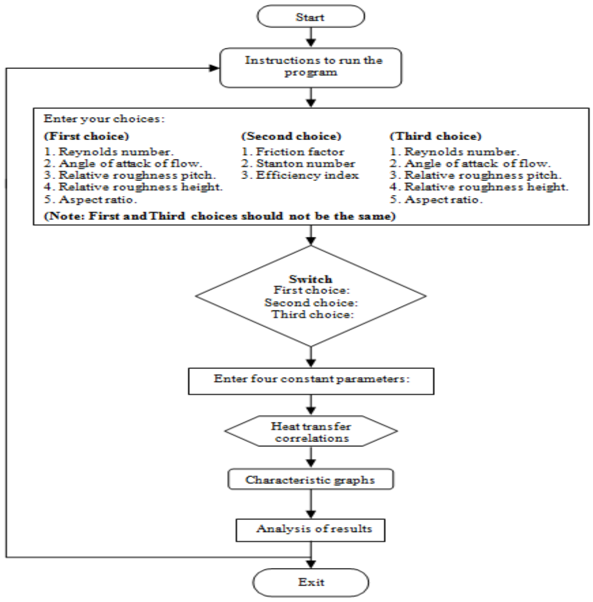
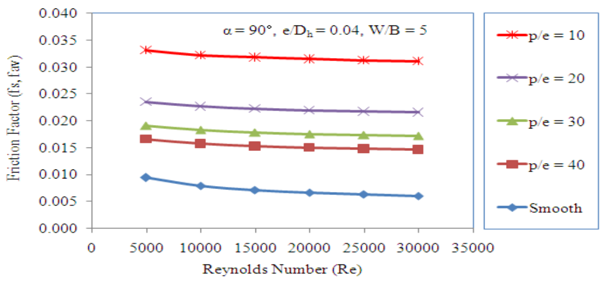
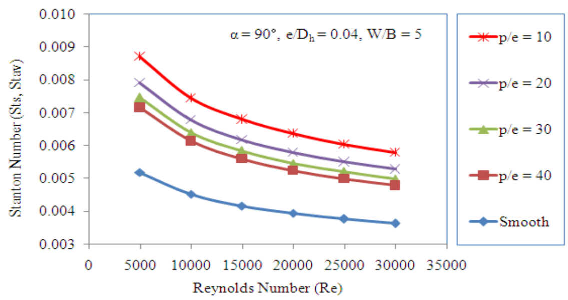


Fig. 1 Flow chart of the computer program

a)



b)



c)

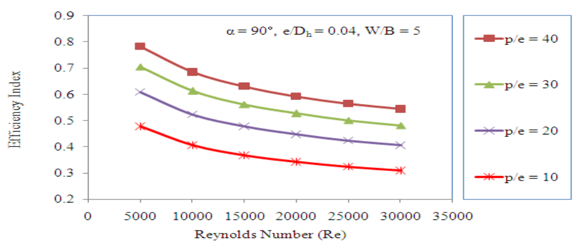


Fig. 2 Effect of Reynolds number on a) friction Factor b) Stanton number c) efficiency index for different relative roughness pitch

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Effect of Reynolds number on friction factor, Stanton number and efficiency index for different values of relative roughness pitch is shown in Fig. 2. It is observed that friction factor and Stanton number for both smooth and roughened ducts decreases as Reynolds number increases. Also, efficiency index decreases as Reynolds number increases. Decrease in friction factor is due to a distinct change in the fluid flow characteristics as a result of roughness that causes flow separation, reattachments and the generation of secondary flows. Stanton number depicts the heat transfer rate. Decrease in Stanton numbers are mainly due to increased heat losses due to turbulence. Similarly comparative values for a smooth and roughened duct clearly show that heat transfer characteristics are better for roughened duct. Also, when compared to smooth duct the roughened one has two to three times increase in value of friction factor mainly due to artificial roughness elements. In general any increase in Reynolds number increases turbulence and hence not desirable for a better performance of air heater duct. It is also observed that friction factor and Stanton number decreases with increase in the value of relative roughness pitch. The decrease in friction is more prominent when compared to Stanton number. Therefore, with increase in the value of relative roughness pitch though both friction factors and Stanton numbers decreases and efficiency index increases. The value of relative roughness pitch 40 produces the highest value of efficiency index as it corresponds to a lower value of friction factor.

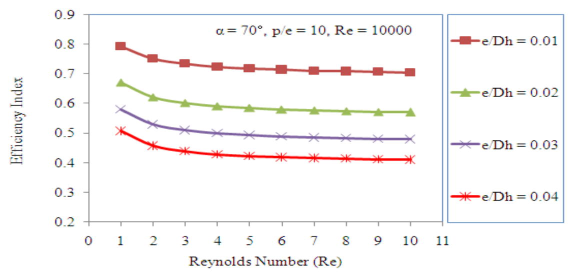
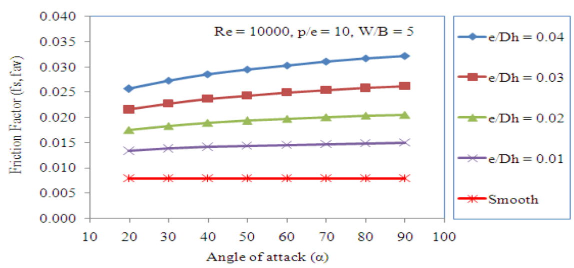


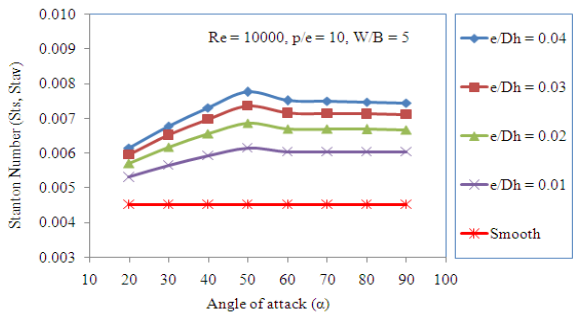
Fig. 3 Effect of aspect ratio on efficiency index for different relative roughness height

Effect of aspect ratio on efficiency index for different values of relative roughness height is shown in Fig. 3. Results show that there is no significant effect of aspect ratio on efficiency index for different values of relative roughness height. Hence aspect ratio of 5 is considered as for this value the width is much greater than the height.

a)



b)



c)

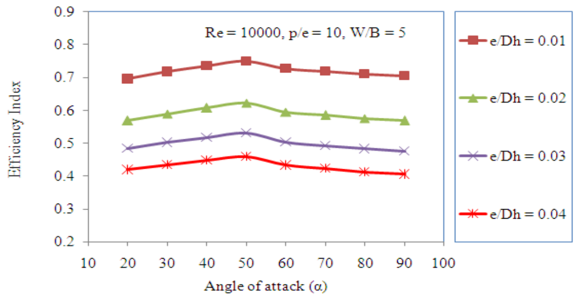


Fig. 4 Effect of angle of attack of flow on a) friction Factor b) Stanton number c) efficiency index for different relative roughness height

From the above discussions it is observed that average friction factor and Stanton number are affected in similar manner by the changes in parameters. When average friction factor increases then, Stanton number also increases but one may dominate over the other. If the Stanton number is dominant, then efficiency index of the duct increases where as if friction factor is dominant, the efficiency index of the duct decreases. Therefore, efficiency index can be defined as the ratio of enhancement factor of heat transfer coefficient to that of friction coefficient. Effect of angle of attack of flow on friction factor, Stanton number and efficiency index for different values of relative roughness height is shown in Fig. 4. Any increase in the value of angle of attack of flow increases frictional resistance for airflow. The value of friction factor for smooth duct is much lower than friction factor for roughened duct, which means that introduction of roughness elements, produces the frictional drag for the flow of air through the duct. Also, it is observed that initially increase in the angle of attack of flow improves the heat transfer characteristics reaching the maximum at 50^o and then further increase in angle of attack of flow reduces the Stanton number. Reason for the occurrence of maximum value at 50^o is yet to be investigated. Also efficiency index is peak at an angle of attack of flow 50^o and for larger angle of attack of flow efficiency index shows a downward tendency. Stanton number for roughened duct at 50^o angle of attack of flow is 20% to 60% more compared to that of smooth duct. Any increase in the value relative roughness height increases the friction

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factor and Stanton number. Also from figures it is clear that efficiency index is higher for lower values of relative roughness height. Results show that increase in value of relative roughness height gives greater increase in friction factor compared to the increase in the Stanton number. Hence it is observed that efficiency index decreases as the relative roughness height increases. The value of relative roughness height 0.01 produces the highest value of efficiency index as it corresponds to a lower value of friction factor. It is observed from the previous discussions that for a solar air heater duct with artificial rib roughness on the absorber plate on one side and with smooth insulated other three sides of the duct the optimum set of parameters are given below.

a) Relative roughness pitch (p/e) = 40
b) Aspect ratio (W/B) = 5
c) Angle of attack of flow (α) = 50^{o}
d) Relative roughness height (e/D_h) = 0.01

IV. CONCLUSIONS

The following are some of the conclusions drawn.

a) The maximum enhancement in Stanton number and friction factor for roughened duct is observed to be 1.13 and 1.19 times that of the smooth duct, respectively.

b) Minimum friction factor and Stanton number occurs for relative roughness pitch of 40, relative roughness height of 0.01. Whereas, efficiency index is maximum for these values.

c) If Stanton number variations are dominant then efficiency of the duct increases whereas if friction factor variations are dominant then efficiency of the duct decreases.

d) Friction factor decreases with decrease in angle of attack of flow. Whereas, Stanton number and efficiency index are maximum at 50° for all the values of relative roughness height and relative roughness pitch. Reason for the maximum values at 50° is yet to be investigated.

e) The empirical expressions for the friction factor and heat transfer can be used for designing and predicting the performance of a solar air collector with the ribbed roughness on the absorber plate.

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