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Review on Thermo-hydraulic Performance of Solar Air Heater having Artificial Roughness on Absorber Plate

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Abstract - *: Second law analysis shows that solar energy can be efficiently used for process heating in industries, drying agriculture products etc. The poor thermal efficiency of solar air heaters is generally due to the low heat transfer coefficient between absorber plate and fluid flowing through the duct. The low heat transfer coefficient is due to existence of thin boundary layer known as the viscous sub-layer which cause to make the insulate situation between heated surface and heat convicting fluid. In the last few years, there has been a growing interest in using roughness underside of the absorber plate which is a very effective method for improving the heat transfer coefficient. In particularly, using the roughness cause to break the viscous sub-layer results into the high thermal performance of the solar air heaters. However, the major drawback of this approach is accompanied by higher friction factor that causes excessive centrifugal fan power for propelling the forced air through the duct. In view a designer needs to carefully examine shape and orientation of roughness elements in order to choose the best fit roughness geometry for intended application. Furthermore it is essential to understand how flow field is affected by particular roughness geometry so that direction of future researches could be conceived. So as to clarify the useful findings an attempt has been made to review roughness geometries employed in solar air heaters. Different roughness geometries have been compared on the basis of heat transfer enrichment and thermo-hydraulic performance to draw attention towards their usefulness.Thus there is scope of further enhancement of thermo-hydraulic performance.*

Keywords—:Thermo-hydraulic performance, Nusselt's number, Friction factor, Relative roughness height, Relative roughness pitch.

1. INTRODUCTION

1.1Artifical Roughness

Artificial roughness is basically a heat transfer enhancement technique by which thermo hydraulic performance of a solar air heater can be improved. Artificial roughness may be in the form of continuous ribs, baffles, dimple shape or protrusions etc. Energy for creating turbulence in flow passage 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 power requirement may be reduced.

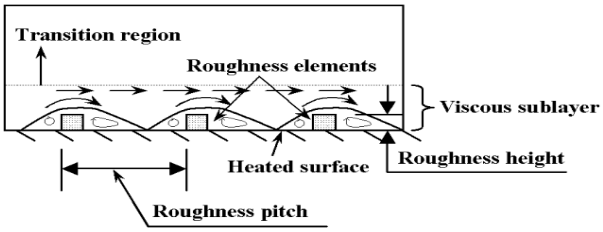


Fig.1. Effect of roughness elements on flow field

Thermo-hydraulic performance (THP): This parameter facilitated the simultaneous consideration of thermal and hydraulic performance and is given by Webb et al [2]

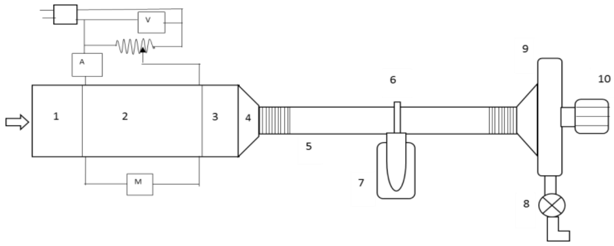
$$THP = \frac{(Nur/Nus)}{(FFr/FFs)^{1/3}}$$

The value of thermohydraulic performance parameter greater than 1 is considered to be goood one .This parameter is generally used to compare the performance of different roughness arrangements.

1.2Experimental approach

Performance estimation of artificially roughened solar air heater requires generation of experimental data on heat transfer and friction for specified roughness geometry. To ensure the reliability of experimental data it is essential to perform experimentation on validated experimental setup under standard test conditions.

Figure 1.2 shows the schematic diagram of indoor experimental setup to generate data on heat transfer and friction in artificially roughened solar air heater as per the guidelines of ASHRAE standard 93-77



1 - Entry Section 5 – Piping 9 – Air Blower
2 - Test Section 6 – Orifice Plat 10 – Prime Mover

3 – Exit Section 7 – Differential Manometer

4 – Plenum 8 - Control Valve

M - Micro manometer, A & V – Ammeter & Voltmeter

Figure 1.2: Schematic diagram of experimental setup

The main elements of the system comprise of insulated rectangular duct with metallic artificially roughened absorber plate. Uniform heat flux is supplied over the top surface of the plate by means of electrical heater and the bottom surface is modified by providing artificial roughness elements. The duct is connected to a circular pipe which accommodates the flow measurement device and flow control valve. The other end of pipe is connected to the suction side of centrifugal blower which exhales the air to the surrounding thus forming an open loop system. The temperature of absorber plate at different locations and inlet and exit temperatures of air are measured with the aid of thermocouples. Pressure drop across the test section of the duct is measured to find the friction losses during the flow over roughened test section.

1.3 Data Reduction

Steady-state values of the plate and air temperature in the duct at various locations were used to determine the values of useful parameters. Mass flow rate, m, velocity of air v, heat supplied to the air Q and heat transfer coefficient h, where calculated by usingFollowing expressions

Pressure difference $\Delta Po = \rho * g * \Delta h$

Mass flow rate $m = C_d * A_O * \left(\frac{2\rho\Delta P_O}{1-\beta^4}\right)^{0.5}$

Velocity of Air $v = \frac{m}{\rho WH}$

Hydraulic diameter $D_h = \frac{4WH}{2(W+H)}$

Reynolds Number $Re = \frac{VD_h}{\vartheta}$

Heat gained by air$Q_a = m * C_p * \left(T_{o_{avg}} - T_{i_{avg}}\right)$

Heat transfer coefficient $h = \frac{Q_a}{A_p * \left(T_{P_{avg}} - T_{F_{avg}}\right)}$

Where the temperature $T_{P_{avg}}$and $T_{F_{avg}}$ are average values of

the absorber plate and fluid (air) temperature, respectively the average value of plate temperature $T_{P_{avg}}$was determined

from the detailed temperature profile of the absorbing plate. It was found that the temperature of the absorber plate varies predominantly in the flow direction only and is linear. The variations in the direction normal to the flow were found to be negligible. The plate temperature was determined at different locations. The average plate temperature $T_{P_{avg}}$ was

determined by taking average temperature of all thermocouples located at different locations on plate the air temperature variations were found to be linear. Since the temperature variations of the absorber plate in the direction normal to the flow were found to be negligible in that direction, the air temperature variations in that direction can be assumed to be negligible. The temperature variation in the direction normal to the absorber plate could not be measured because of very small duct depth and were assumed to be negligible. The air temperature T_f was

determined as an average of the temperatures measured at few location of the duct cross-section along the flow direction at in late and out late. The convective heat transfer coefficient was then used to obtain the average Nusselt number from the following expression:

Nusselt Number $Nu = \frac{hD_h}{k}$

Friction factor $f = \frac{2\Delta P_T D_h}{4\rho L V^2}$

The friction factor was determined from the measured values of pressure drop ΔP_T across the test section length, L. The

validity of the test setup was verified by conducting experiments for smooth duct under similar conditions. The values of Nusselt number and friction factor determined for smooth duct were compared with the values obtained from correlations of Dittus–Boetler equation for the Nusselt number and modified Blasius equation for the friction factor. Nusselt number for a smooth rectangular duct is given by Dittus–Boetler equation.

$$Nu = 0.023Re^{0.8}Pr^{0.4}$$

Friction factor for a smooth rectangular duct is given by Modified Blasius equation as

$$Fr = 0.085Re^{-0.25}$$

Thermo-hydraulic performance (THP)

$$THP = \frac{Nu_r}{Nu_s} \Big/ \left(\frac{f_r}{f_s}\right)^{\frac{1}{3}}$$

2. VARIOUS ROUGHNESS GEOMETRIES USED IN SOLAR AIR HEATERS

Artificial roughness in a solar air heater covers wide variety of roughness geometries for examines heat transfer and friction characteristics. General arrangements of different types of roughness geometries reported by various investigators have been discussed in following sub-sections.

2.1 Artificial Roughness in the Form Ribs/Wire

Prasad and Saini[3,4]reported an experimental investigation of fully developed turbulent flow in a solar air heater duct having small diameter protrusion wires fixed on absorber plate as shown in Fig.2.1. Nusselt number and friction factor correlations were developed by using experimental data. An enhancement in Nusselt number and friction factor was observed over plane duct of the order of 2.38 and 4.25 times respectively corresponding to relative roughness height of 0.033 and relative roughness pitch of 10.

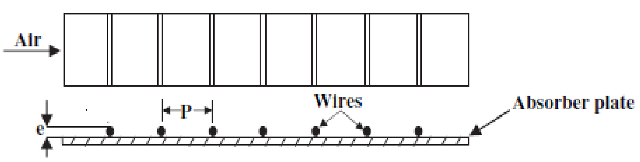


Fig. 2.1Roughened absorber plate fixed with transverse continuous wires

2.2 Inclined and V-shaped or staggered ribs

Gupta et al. [8] established optimum design parameters under actual climatic conditions for roughened solar air heaters for varying relative roughness height (e/D) and for a relative roughness pitch (P/e) of 10 at an angle of attack (a) of 60 deg. Geometry of roughened absorber plate is shown in Fig. 2.2. An enhancement of heat transfer and friction factor was obtained of the order of 1.8 and2.7 times respectively. Maximum heat transfer coefficient and friction factor values were obtained at an angle of attack of 60degrespectively in the range of carry out research on parameters.

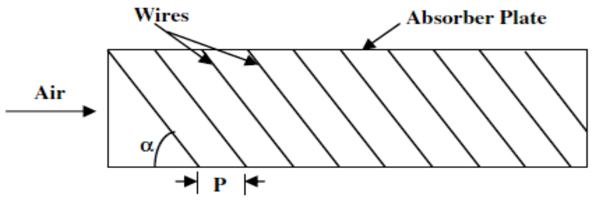


Fig. 2.2 Roughened absorber plate with inclined ribs.

2.3 V-shaped continuous Wire ribs

Momin et al. [11] inspected effect of geometrical parameters on heat transfer and fluid flow characteristics of rectangular duct of solar air heater having V-shaped ribbed roughness on the absorber plate as shown in Fig. This experimental investigation covered a Reynolds number range of 2500–18,000, relative roughness height (e/D) of 0.02–0.034 and angle of attack (α) of 30–90degfora fixed relative roughness pitch (P/e) of 10. It was reported that V-shape ribs with an angle of attack (α) of 60degenhanced Nusselt number by 1.14 and 2.30 times and friction factor by 2.30 and 2.83times over inclined ribs and smooth plate respectively.

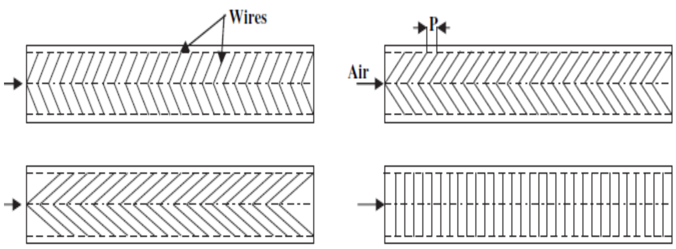


Fig. 2.3 V-shape and transverse roughness elements on absorber plate.

2.4 Grid shaped wire ribs

Karmare and Tikekar [14] developed heat transfer coefficient and friction factor correlation for artificially roughened duct with metal grit ribs as shown in Fig. 2.4. Effect of range of system parameters of grit geometry on heat transfer coefficient and friction Factor was carry out research on for Reynolds number range of 4000–17,000. It is reported that plate having roughness parameters L/S = 1.72,e/D = 0.044 and P/e = 17.5 resulted optimum performance and as compared to smooth duct yields up to two-fold enhancement in Nusselt number and three-fold enhancement in friction factor.

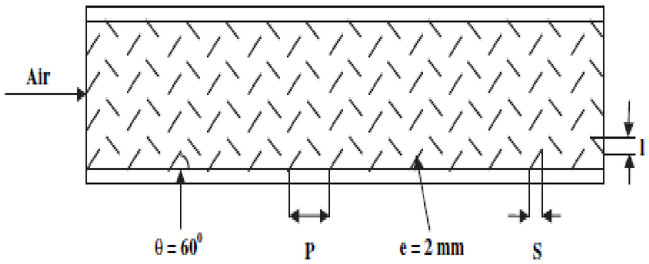


Fig. 2.4 Roughness geometry in rectangular channel as grit shape ribs Gap in an inclined continuous rib

Aharwal et al. [15] inspected effect of artificial roughness by using an inclined non-continuous rib arrangement in a rectangular duct shown in Fig. 2.5. Maximum enhancement in Nusselt number and friction factor as compared to smooth duct was observed to be2.59 and 2.87 times respectively.

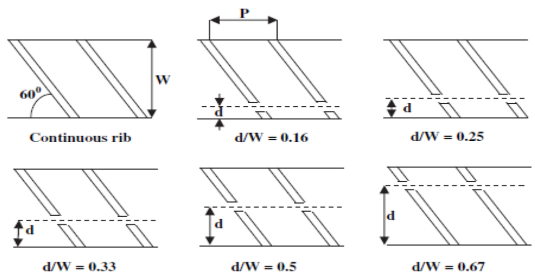


Fig. 2.5 Roughness geometry as inclined non-continuous arrangement ribs.

2.5 Combination of inclined and transverse ribs

Saini and Saini [17]carry out research on effect of arc shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of solar air heater as shown in Fig. 2.6 This experimental investigation covered a Reynolds number range of 2000–17,000, relative roughness height (e/D) of 0.0213–0.0422 and relative angle of attack of flow(a/90) of 0.3333–0.6666 for a fixed relative roughness pitch (P/e) of10. Maximum development in Nusselt number and friction factor as compared to smooth duct was observed to be 3.6 and 1.75 times respectively.

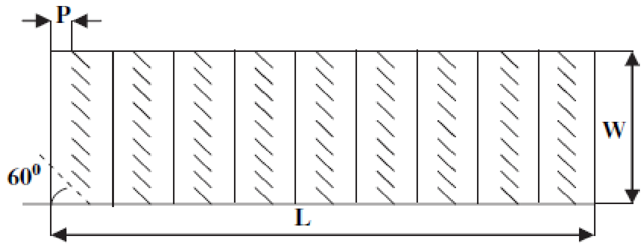


Fig. 2.6 Roughness geometry as a combination of inclined and transverse ribs.

2.6 Wedge shaped ribs

Bhagoria et al. [21]carry out research on air heater rectangular duct roughened by wedge shaped transverse integral ribs as shown inFig.2.7.for Reynolds number range of 3000–18,000. Range of relative roughness height (e/D), relative roughness pitch (P/e) and rib wedge angle was 0.015–0.033, 60.17_1.0264 <p/e< 12.12 and8–15degrespectively. Authors reported an enhancement in Nusselt number and friction factor of the order of 2.4 and 5.3 times respectively as compared to smooth duct.

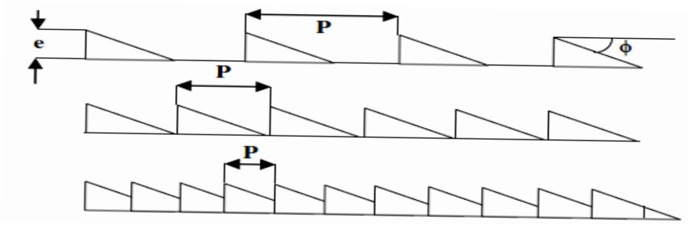


Fig. 2.7 Absorber plate having transverse wedge shaped rib roughness.

2.7 Combination of different integral rib roughness elements

Jaurker et al. [29] reported an experimental investigation on heat and fluid flow characteristics for fully developed turbulent flow in a rectangular duct having repeated integral transverse rib groove roughness as shown in Fig.2.8 for Reynolds number range of 3000–21,000. Ranges of relative roughness height (e/D), relative roughness pitch (P/e) and groove position to pitch ratio (g/P) was0.0181–0.0363, 4.5–10.0 and 0.3–0.7 respectively.

Enhancement of Nusselt number of the order of 2.75 times of the smooth duct and 1.57 times of ribbed duct with similar rib height and rib spacing was observed. Whereas ribbed duct with similar rib height and rib spacing provides Nusselt number values of the order of 1.7 times that of smooth duct for range of parameters. On the other hand friction factor increases in the order of 3.61 times that of smooth duct and 1.17 times that of ribbed duct. Whereas a ribbed duct with similar rib height and rib spacing results in friction factor value of the order of 3 times that of the smooth duct.

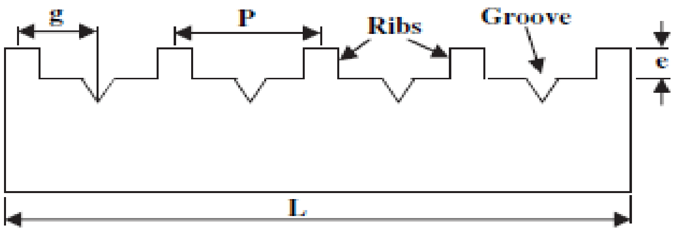


Fig. 2.8 Absorber plate having rib-grooved artificial roughness.

2.8 Chamfered rib-groove combination

Layek et al. [23] carry out research on heat transfer and friction characteristics of repeated integral transverse chamfered rib-groove roughness as shown in Fig.2.9 for a Reynolds number range of3000–21,000, relative roughness pitch of 4.5–10, chamfer angle of5–30deg, relative groove position of 0.3–0.6 and relative roughness height of 0.022–0.04. Authors reported that Nusselt number and friction factor increased by 3.24 times and 3.78 times respectively as compare to smooth duct. Maximum enhancement of Nusselt number and friction factor was obtained corresponding to relative groove position of 0.4.

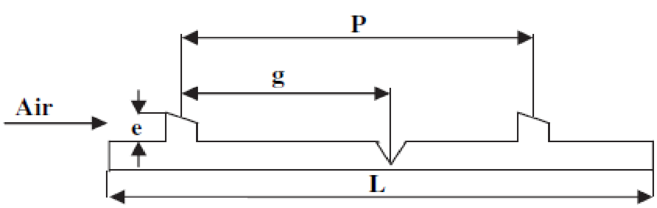


Fig. 2.9 Absorber plate having chamfered rib-grooved artificial roughness.

2.9Protruded Dimple shaped geometry

Saini and Verma [30] carry out research on heat transfer and friction characteristics of dimple shaped artificial roughness geometry shown in Fig.2.10for Reynolds number 2000–12,000. Range of relative roughness height (e/D) and relative pitch (p/e) was 0.018–0.037 and 8–12 respectively. Authors reported that Nusselt number and friction factor increased by 1.8 and 1.4 times respectively as compared to smooth duct.

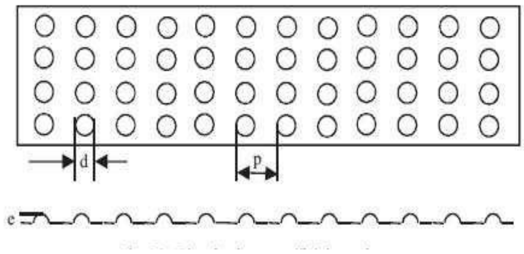


Fig. 2.10 Dimple shaped artificial roughness

2.10 Staggered protruded dimple roughness

Bhushan et al [28]carry out research on staggered dimple roughness in place of transverse dimple roughness. Range of parameter carry out research on were relative short way length (S/e) as 18.75-37.50, relative long way length (L/e) as 25.00-37.50, relative print diameter(d/D) as 0.147-0.367 , relative roughness height as 0.03, aspect ratio as 10 and Reynolds number from 4000-20,000. Under given condition maximum enhancement of Nusselt number and friction factor was 3.8 and 2.2 times respectively in comparison to smooth duct. Maximum improvement in heat transfer coefficient was reported for relative short way length (S/e) of 31.25, relative long way length (L/e) of 31.25 and relative print diameter (d/D) of 0.294, Roughness geometry shown in fig.2.11.

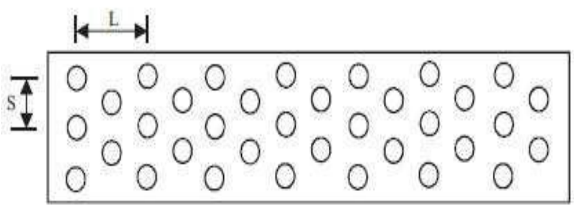


Fig. 2.11 Staggered dimple roughness

2.11 Arc shaped protruded dimple roughness

Yadav et al [31] employed Arc shaped dimple roughness. Research parameter were Reynolds number range from 3600-18,000,(p/e) as 12 to 24. (e/D) as 0.015-0.03 and arc angle of protrusion arrangement as 45-75 . Maximum enhancement of Nusselt number and friction factor was found to be 2.89 and 2.93 times respectively of smooth duct for range of parameter investigated. Maximum enhancement of heat transfer and friction factor occurred for relative roughness height of 0.03, relative roughness pitch of 12 and for arc angle value of 60 deg. Roughness geometry shown in fig.2.12

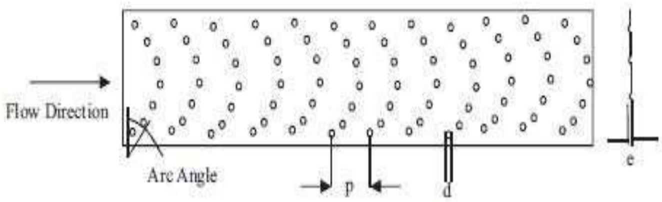


Fig. 2.12 Dimple roughness in arc maner

3. Summary of experimental investigation performed by several researcher

Investigator	Roughness Geometry	Parameters Used	Remark
Prasad and Saini	Transverse Small diameter protrusion wire	e/D: 0.020–0.033 p/e: 10–20 Re:3000-18000	Relative roughness height e/D: 0.033 Relative roughness pitch p/e: 10 Nusselt number (Nu) and friction factor(FF) for smooth duct & for roughness duct Nur & FFr resp. Nur=2.38Nu FFr=4.25FF THP :1.46931
Gupta et al.	V-Shaped/Inclined Wire Ribs Inclined wire ribs	e/D: 0.02–0.053 Re: 5000–30,000 α: 30–90_ p/e: 7.5–10	Angle of attack 60 deg Nur=1.8Nu FFr=2.7FF THP :1.2926
Momin et al.	V-shaped continuous Wire ribs	e/D: 0.02–0.034 p/e: 10 α:30–90_deg	Angle of attack 60 deg Nur=2.3Nu FFr=2.83FF

		Re: 2500–18,000	THP :1.626
Karmare and Tikekar	Grid shaped wire rib	e/Dh: 0.035–0.044 p/e: 12.5–36 l/s: 1.72–1	p/e=17.5 e/d=.044 , l/s= 1.72, Nur=2Nu FFr=3FF THP :1.3861
Aharwal et al	Gap in an inclined continuous	p/e: 10 e & b: 2mm e/Dh: 0.0377 W/H: 5.87 Re: 3000–18,000 d/W: 0.167–0.5 α: 60 deg	Angle of attack 60 deg Nur=1.8Nu FFr=2.7FF THP :1.822
Saini and saini	Combination of inclined and transverse ribs	Re: 2000–17000 p/e: 10 W/H: 12 e/D: 0.0213–0.0422 α/90: 0.333–0.666	Angle of attack 60 deg Nur=3.6Nu FFr=1.75FF THP :2.987
Bhagoria et al	Wedge shaped ribs	e/D: 0.015–0.033 p/e:12.5-60.17 α: 8, 10, 12, 15 Re: 3000–18,000	Nur=2.4Nu FFr=5.3FF THP :1.3765
Jaurkar et al	Rib-Groove combination	e/D: 0.0181–0.0363 p/e: 4.5–10 Re: 3000–21,000 g/p: 0.3–0.7	Nur=2.75Nu FFr=1.57FF THP :1.7924

Layek et al.	Chamfered rib-groove combination	Re: 3000–21,000 e/Dh: 0.022–0.04 P/e: 4.5–10 g/P: 0.3–0.6	Relative groove position of 0.4 Nur=3.24Nu FFr=3.78FF THP :2.0791
Sainiet al	Transverse protruded Dimple Roughness	e/D: 0.018–0.037 And p/e : 8–12 Re: 2000-12000	Nur=1.8Nu FFr=1.4FF THP :1.6
Bhushan et al	Staggered protruded Dimple Roughness	S/e :18.75-37.50 L/e :25.00-37.50 d/D:0.147-0.367 Re:4000-20000 Aspect ratio 10	s/e:31.25 l/e:31.25 d/D:0.294 Nur=3.8Nu FFr=2.2FF THP :2.921
Yadav et al	Arc shaped protruded Dimple Roughness	p/e) as 12 to 24. (e/D) as 0.015-0.03, Arc angle of protrusion 45-75 deg. Re:3600-18000	e/d:0.03 p/e:12 Arc angle 60deg Nur=2.89Nu FFr=2.93FF THP :2.0196

4. CONCLUSION

This paper reported review on thermo-hydraulic performance (THP) of different artificially roughened surface with different geometries of different shapes, sizes and orientation by various investigators Use of artificial roughened surface is found to be the good technique to enhance the heat transfer rate with little penalty. Maximum increment in Nusselt number and Friction Factor due to various roughened surfaces over smooth duct have been summarized and listed in Table. The least and maximum values of THP are found to be 1.2 and 2.9 respectively. Combination of inclined and transverse ribs and staggered protruded Dimple Roughness shows highest THP. Thus there is scope of further enhancement of THP by increase in Nusselt number with less penalty of friction factor. For that optimum shape, size and orientations of roughness element need to be done.

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